Chapter 6
Developments in Internal Combustion Engines

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Greek Symbols

η_c	Combustion efficiency
η_f	Fuel conversion efficiency
$\eta_{f,b}$	Brake fuel conversion efficiency
$\eta_{f,i,}$	Indicated fuel conversion efficiency
η_m	Mechanical efficiency
η_{th}	Thermal efficiency
$\eta_{th,Carnot}$	Thermal efficiency of the ideal Carnot cycle
$\eta_{th,Otto}$	Thermal efficiency of the ideal heat engine Otto cycle
η_v	Volumetric efficiency
γ	Ratio of specific heats
γ_b	Ratio of specific heats of the burned mixture
φ	Fuel–air equivalence ratio. For $\varphi < 1$ mixture is lean. For $\varphi = 1$, mixture is stoichiometric. For $\varphi > 1$, mixture is rich. Note that φ is the inverse of the often-used air–fuel equivalence ratio, or λ.
$\rho_{a,i}$	Inlet air density
τ	Engine torque

Symbols

BMEP	Brake mean effective pressure
$C_{p,b}$	Constant pressure specific heat of the burned mixture

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$C_{v,b}$	Constant volume specific heat of the burned mixture
f	Residual fraction
f_{final}	Final calculated residual fraction
$f_{final-1}$	Previous iteration residual fraction to final calculated residual fraction
(F/A)	Fuel–air ratio
FMEP	Friction mean effective pressure
h_1	Specific enthalpy at state 1
h_2	Specific enthalpy at state 2
h_3	Specific enthalpy at state 3
h_{3a}	Specific enthalpy at state 3a
h_5	Specific enthalpy at state 5
h_6	Specific enthalpy at state 6
h_e	Specific enthalpy of exhaust mixture
h_i	Specific enthalpy of inlet mixture
$IMEP_g$	Gross indicated mean effective pressure
$IMEP_n$	Net indicated mean effective pressure
m	Mass
m_1	Mass at state 1
m_2	Mass at state 2
m_3	Mass at state 3
m_4	Mass at state 4
m_6	Mass at state 6
m_a	Mass of air
m_f	Mass of fuel
m_r	Residual mass
m_{total}	Total mass
$\dot{m}_a$	Mass flow rate of air
$\dot{m}_f$	Mass flow rate of fuel
MEP	Mean effective pressure
M_b	Molecular weight of the burned mixture
n_R	Number of revolutions per engine cycle
N	Engine speed
$\overline{R}$	Universal gas constant
R	Gas constant
R_5	Gas constant of mixture at state 5
R_6	Gas constant of mixture at state 6
R_e	Gas constant of exhaust mixture
P	Cylinder pressure or power
P_1	Pressure at state 1
P_2	Pressure at state 2
P_3	Pressure at state 3
P_{3a}	Pressure at state 3a
P_4	Pressure at state 4
P_5	Pressure at state 5

P_6	Pressure at state 6
P_7	Pressure at state 7
P_b	Brake power
P_e	Exhaust pressure
P_i	Inlet (initial) pressure
P_{in}	Net indicated power
P_{limit}	Limit pressure
PMEP	Pumping mean effective pressure
$_1Q_2$	Heat transfer of process 1–2
$_6Q_1$	Heat transfer for process 6–1
Q_{HV}	Heating value of fuel
$Q_{HV,f}$	Heating value of fuel
$Q_{HV,i}$	Heating value of specie i
r_c	Compression ratio
s_1	Entropy at state 1
s_2	Entropy at state 2
s_3	Entropy at state 3
s_4	Entropy at state 4
s_5	Entropy at state 5
T_1	Temperature at state 1
T_4	Temperature at state 4
T_5	Temperature at state 5
T_6	Temperature at state 6
$T_{cv,adiabatic}$	Constant volume adiabatic flame temperature
T_e	Exhaust temperature
T_H	Temperature of a source reservoir
T_i	Inlet temperature
T_L	Temperature of a sink reservoir
T_r	Residual fraction temperature
u_1	Specific internal energy at state 1
u_2	Specific internal energy at state 2
u_3	Specific internal energy at state 3
u_{3a}	Specific internal energy at state 3a
u_4	Specific internal energy at state 4
U_1	Internal energy at state 1
U_2	Internal energy at state 2
U_3	Internal energy at state 3
U_4	Internal energy at state 4
v_1	Specific volume at state 1
v_2	Specific volume at state 2
v_3	Specific volume at state 3
v_{3a}	Specific volume at state 3a
v_4	Specific volume at state 4
V	Cylinder volume

V_1	Volume at state 1
V_2	Volume at state 2
V_3	Volume at state 3
V_5	Volume at state 5
V_6	Volume at state 6
V_d	Displaced volume
V_{max}	Maximum cylinder volume
V_{min}	Minimum cylinder volume
W	Thermodynamic work
$_1W_2$	Work for process 1–2
$_2W_3$	Work for process 2–3
$_3W_4$	Work for process 3–4
$_4W_5$	Work for process 4–5
$_5W_6$	Work for process 5–6
$_6W_1$	Work for process 6–1
W_b	Brake work
W_f	Friction work
W_{gross}	Gross work
W_{ig}	Gross indicated work
W_{in}	Net indicated work
W_{ip}	Pump work
W_{net}	Net work
x_i	Mole fraction of specie i
y_i	Mass fraction of specie i

Glossary

Combustion	Rapid oxidation of a fuel–air mixture (reactants) converting reactants to products and in the process releasing thermal energy.
Lean	Air–fuel mixture is such that there is more air than is chemically necessary to oxidize the available fuel.
Products	Species formed as the result of a chemical reaction (in the context of this article species formed as the result of a combustion reaction).
Reactants	Species that are to be involved in a chemical reaction (in the context of this article species that are to be involved in a combustion reaction).
Rich	Air–fuel mixture is such that there is less air than is chemically necessary to oxidize the available fuel.
Stoichiometric	Air–fuel mixture is chemically balanced such that there is the correct amount of air to fully oxidize the available fuel.

Definition of the Subject

An internal combustion (IC) engine is a thermodynamic work conversion device that converts chemical energy (typically delivered to the engine in the form of a liquid or gaseous fuel) to work energy (typically in the form of shaft work issuing from a rotating crankshaft). The internal combustion engine markedly distinguishes itself from other types of power-producing equipment – most notably, the heat engine (e.g., steam engine or steam-cycle plant). In the former, chemical energy is released, via combustion (i.e., rapid oxidation) mechanism, internal the same device that converts the released energy to work energy. In the latter, thermal energy is passed into the device via heat transfer; the device thereby converts the thermal energy to work energy. Because of this subtle difference, the thermodynamic limits of maximum efficiency of the internal combustion engine are constrained differently than the thermodynamic limit of maximum efficiency of the heat engine.

There are several types of internal combustion engines; the two most common being the piston/cylinder reciprocating engine and the gas turbine engine. This article constrains its discussion to just piston/cylinder reciprocating engines. As this article is meant to be brief, readers seeking additional and thorough information are referred to the associated cited articles and the authorities listed in "Books and Reviews".

Introduction

Internal combustion engines are pervasive to our daily activities. They are the primary powerhouse of the transportation industry. They serve as neighborhood and small municipality backup power generators or primary power stations. They provide convenience by powering lawnmowers, leaf blowers, and weed trimmers. They deliver excitement and entertainment in pleasure boats, race cars, and motor bikes. Their scale of usability nearly spans the scales of classical physics, from as small as "micro engines" that fit in the palm of your hand to as large as marine engines that scale three stories and use human-sized doorways for entry into the cylinder block.

The story of the internal combustion engine dates back around 150 years ago, when J.J.E. Lenoir built an engine that combusted coal–air mixtures in a cylinder outfitted with a piston with a two-stroke type fashion (without compression). Soon thereafter, Nicolaus A. Otto and his colleague Eugen Langen expanded on the Lenoir concept, creating an engine that had 11% efficiency compared to Lenoir's 5%. Determined to improve efficiencies of internal combustion engines (which at 11%, were not much better than the steam engine), Otto built the first engine operating on the four-stroke principle which today serves as the primary operating cycle of engines [1]. It is here noted that, although Otto built the first working

four-stroke engine, Alphonse Beau de Rochas described in theory the principles of the four-stroke cycle. Further, Beau de Rochas outlined several conditions to achieve maximum efficiency of the internal combustion engine [1].

From this point, and with the aid of several important developers including Rudolf Diesel, Sir Harry Ricardo, Robert Bosch, and Charles Kettering, the internal combustion engine has become one of the most highly efficient, power dense, cost-effective, easily maintained, and versatile power machinery available to consumers. This article will provide some of the important basic information about internal combustion engines, and indicate some of the more recent developments that continue to make internal combustion engines competitive as the preferred power-producing technology.

The Basics of Internal Combustion Engines

Basic Operating Cycle

As this article concentrates its discussions of IC engines on those of the piston/ cylinder reciprocating type, Fig. 6.1 [2] shows the basic geometrical considerations of the piston/cylinder/crankshaft arrangement, which kinematically is described as a crank-slider mechanism. The chemical energy to work energy conversion occurs inside the cylinder, usually bound on the sides by the cylinder walls, on the top by the cylinder head (which typically houses the gas exchange valves, such as the intake and exhaust valves, and other important hardware such as spark plugs and/or fuel injectors), and on the bottom by the piston which reciprocates within the cylinder. A rigid connecting rod fastens the piston to an eccentric location on the rotating crankshaft. The eccentric placement of the connecting rod converts the reciprocating motion of the piston (i.e., boundary motion work) to the rotating motion of the crankshaft (i.e., shaft work). The eccentric placement of the connecting rod to the crankshaft also dictates the important geometrical parameter of the piston engine called the "stroke." The stroke and "bore," or diameter of the cylinder, create the displaced volume, V_d, of the piston engine. The maximum volume of the cylinder, V_{max}, is attained when the piston is at its bottom-most position; a position referred to as "bottom dead center," or BDC. The minimum volume, or clearance volume, V_c, is attained when the piston is at its top-most position; a position referred to as "top dead center," or TDC. The ratio between V_{max} and V_{min} is called the compression ratio, r_c, and is given as Eq. 6.1:

$$r_c = \frac{V_{max}}{V_{min}} \tag{6.1}$$

The compression ratio, as will be described in section on "Thermodynamic Analysis of Internal Combustion Engines", is a fundamentally critical parameter

Fig. 6.1 Illustration of a piston/cylinder arrangement, as often employed in a reciprocating-piston internal combustion engine (Used with permission from [2])
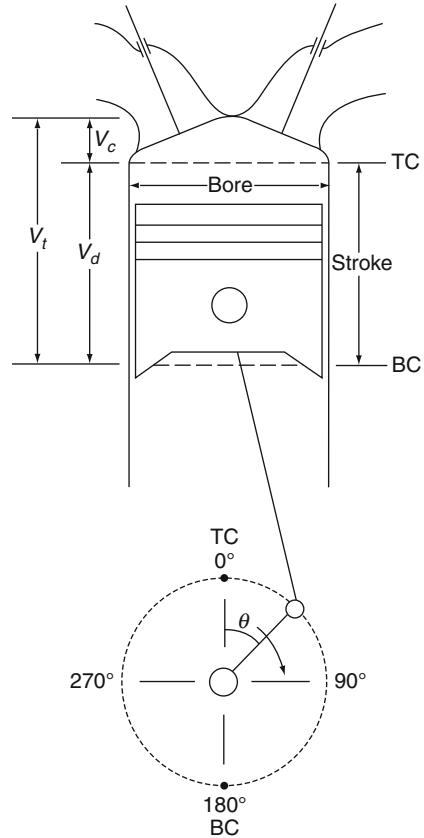
for controlling the efficiency (i.e., the ratio of work energy out to chemical energy in) of an internal combustion piston engine.

There are two major cycles used to exploit the piston engine's conversion of chemical energy to work energy: a "two-stoke" cycle and a "four-stroke" cycle. The earliest prototype engines were of the two-stroke variety (e.g., Lenoir and Otto/ Langen engines) [1]. In pursuit of achieving higher efficiency, Otto (for whom the thermodynamic ideal "Otto Cycle" is named) built the four-stroke version of his engine [1]. Today, four-stroke cycle engines are the dominant form; thus, most of the article will center on the details of the four-stroke cycle.

Four-stroke cycle engines require four strokes of the piston to complete one power-producing cycle, as shown in Fig. 6.2 [3]; the reader is also referred to Fig. 6.4a to aid the discussion. Consider first the cycle starting with the piston at TDC and the intake valve open. The piston moves from TDC to BDC, inducting fresh mixture (conventionally, fuel and air in a gasoline engine and only air in a diesel engine) through the open intake valve in what is called the "intake stroke." At some location near BDC, the intake valve closes, and the piston reverses its motion at BDC. Once the valve closes, the piston/cylinder arrangement creates a closed system. As the piston moves from BDC to TDC, the trapped mixture is
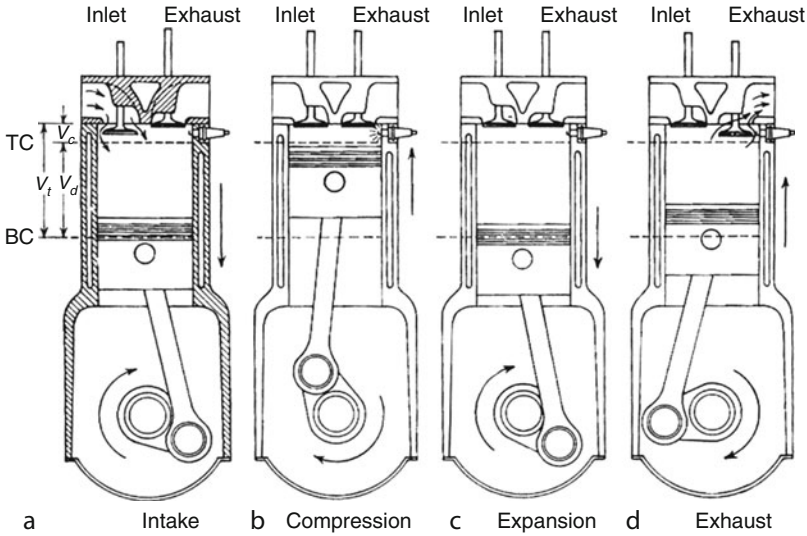
Fig. 6.2 Illustration of the four-stroke operating principle (Used with permission from [3])

compressed in what is called the "compression stroke," increasing the mixture's temperature, pressure, and decreasing its specific volume (i.e., increasing its density). At a point near TDC combustion is expected to commence. In the case of a spark ignition engine (e.g., a conventional gasoline engine), combustion is initiated by the release of spark at a point near (usually advanced of) TDC. In the case of a compression ignition engine (e.g., a conventional diesel engine), combustion is initiated by injecting liquid fuel directly into the cylinder; the compressed air at elevated temperature and pressure atomizes, vaporizes, and mixes with the injected fuel. After a period of time, the high temperature environment causes chemical reaction and start of combustion. Around start of combustion, the piston reaches TDC, reverses direction, and expands the cylinder volume as combustion converts chemical energy into work energy. This stroke takes on many names, including "power stroke," "combustion stroke," and "expansion stroke." As the piston approaches BDC, the exhaust valve opens, allowing the products of combustion to escape the cylinder. At BDC, the piston reverses direction and motions toward TDC with the exhaust valve open, in what is called the "exhaust stroke." Depending on the engine's crankshaft rotational speed – which can vary from as low as 100 rev/min for large marine-application engines to as high as 15,000 rev/min for race car engines – the four-stroke cycle requires as much as about 1.2 s to as little as 8 ms to complete.

Up to this point, the discussion has centered on the processes occurring in a single cylinder. Most engines, however, are composed of many cylinders and take on various forms (e.g., in-line four-cylinder, "V6", "V8", and "W" engine). In such cases, each cylinder undergoes the same processes but usually out of phase. For example, in an in-line four-cylinder engine (i.e., an engine that has four

cylinders oriented sequentially in a single-line bank, an example of which is shown
in Fig. 6.3), the cylinder processes are typically out of phase by 180° (720°/4). The
firing order is usually not linear, however, in order to ensure smooth and continuous
operation. For example, a four-cylinder firing order may go 1-3-2-4; that is, as the
engine crankshaft rotates cylinder 1 produces a power stroke first, followed by
cylinder 3, then cylinder 2, and finally by cylinder 4 (which is then followed again
by cylinder 1). Finally, when describing an engine's displacement, it refers to
a summation of each cylinder's displacement; thus, each cylinder displacement is
on average the total engine displacement divided by the number of cylinders.

How an Engine Makes Power

The reciprocating motion of the piston in a cylinder is the means by which the
chemical energy released during combustion is converted into useful work. Work is
transferred when a force acts through a displacement; in the case of the piston/
cylinder engine, the in-cylinder pressure, P, is interpreted as the force and the
changing cylinder volume, dV, during piston strokes is interpreted as the displacement. Thus, the thermodynamic work of an engine cycle is given by Eq. 6.2:

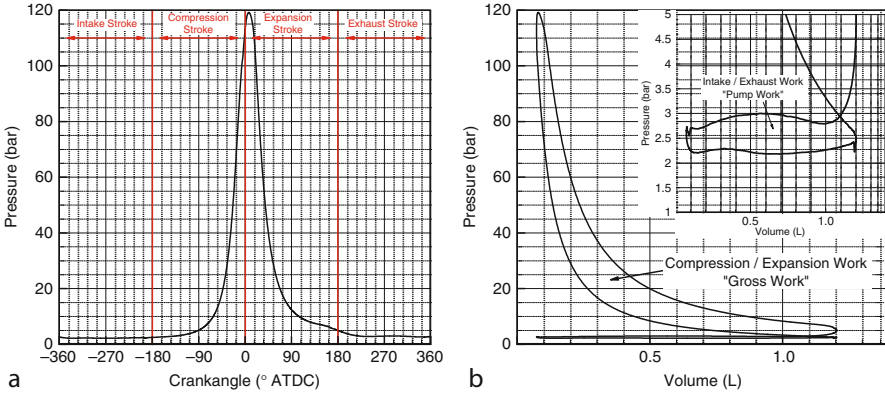$$W = \oint P dV \qquad [\text{kJ/cycle}] \qquad (6.2)$$

Fig. 6.4 (**a**) Pressure as a function of crankshaft rotation, or crankangle, in degrees after top dead center (° ATDC), and (**b**) pressure as a function of cylinder volume, also illustrating the areas of the *P–v* plane that render gross work and pump work (Data from author's laboratory, Texas A&M University)

The work of the cycle given by (Eq. 2) is considered *boundary work*, since it results from the changing boundary of the control system (in this context, the control system is that enclosed by the piston/cylinder arrangement, and the moving boundary is manifested by the moving piston). Conventional engines convert this boundary work of the piston to *shaft work* through the piston connecting rod's eccentrically located connection to the crankshaft. The shaft work issuing from the crankshaft is often best interpreted as *torque, τ*; the work given by Eq. 6.2 is related to the torque of the crankshaft via Eq. 6.3:

$$W = 2\pi n_R \tau \qquad [\text{kJ/cycle}] \qquad (6.3)$$

where n_R is the number of crankshaft revolutions per power cycle (i.e., $n_R = 2$ for a four-stroke cycle and $n_R = 1$ for a two-stroke cycle). The units for τ in Eq. 6.3 are kN-m.

The torque of an engine is routinely measured with a dynamometer; in this way, the torque is considered "brake torque," or the amount of resistance torque the dynamometer must apply to "brake" the engine to a certain speed condition. Via application of Eq. 6.3, the "brake work," W_b, is determined. Only determining brake work, however, reveals no insight into the in-cylinder work processes. The in-cylinder work processes can be calculated using in-cylinder pressure measurement that is precisely coupled to the in-cylinder volume via a crankshaft encoder and detailed knowledge of the cylinder's and crank-slider's geometries. An example "pressure–crankangle" diagram is shown Fig. 6.4a, where crankangle is reported in degrees after top dead center (° ATDC) relative to "combustion TDC." Also shown in Fig. 6.4a are the four strokes of the four-stroke cycle as described in

section "Basic Operating Cycle". In-cylinder pressure is typically measured with a piezo-electric pressure transducer which is able to provide a fast-response indication during the engine cycle [5]. It should be noted that measuring in-cylinder pressure is not a trivial task and great care must be taken to do it properly [6–9]. When using digital equipment (i.e., an analog–digital converter) to electronically record in-cylinder pressure, it becomes necessary to determine the sample rate, which is usually determined by the crankshaft encoder. Varying crankangle resolutions can be used, depending on the level of precision needed of the analysis. For calculating in-cylinder work (described next), a crankangle resolution of up to 10° [6] can be used; for detailed combustion analysis much finer resolution must be used (e.g., about 1° for gasoline engine combustion and 0.25° for diesel engine combustion). The data shown in Fig. 6.4 is recorded every 0.2°. In addition to the crankangle resolution, the engine speed also determines the needed sample rate of the data acquisition system. Finally, it is equally important to know the cylinder volume at each record of pressure when calculating in-cylinder work. This requires precise phasing between the piston's location and the crankshaft encoder; it also requires knowing the precise geometries of the cylinder and crank-slide mechanism. Specifically, the minimum volume (i.e., clearance volume), the piston stroke, and the cylinder bore must be precisely known. It is often best to determine these using precise measuring instruments, rather than rely on manufacturer specifications (which, although have tight tolerances, are nominal values). An example "pressure–volume," or "P–V," diagram is shown in Fig. 6.4b.

Once a precise P–V diagram is determined, the in-cylinder work associated with each process can be determined. The area between the P–V curves, as suggested by Eq. 6.2, represents the in-cylinder work, or "indicated" work (named for the antiquated use of a mechanical stylus-indicator device to record pressure [10]). There are two portions of the typical four-stroke engine cycle, as shown in Fig. 6.4: (1) compression and expansion strokes which in combination result in the "gross work," W_{ig}, and (2) intake and exhaust strokes which in combination result in the "pump work," W_{ip}. It is important to note that gross work and pump work correspond to the respective strokes of the piston, not necessarily the valve events (i.e., not necessarily the closed portion of the cycle versus open portion of the cycle). In combination, i.e., the sum of gross work and pump work result in the "net work," as given by Eq. 6.4:

$$W_{in} = W_{ig} + W_{ip} \qquad (6.4)$$

Note that the subscript "i" on the terms in Eq. 6.4 represents "indicated"; since terms "gross," "pump," and "net" only have relevance from indicated data (i.e., in-cylinder pressure data), it is often dropped as a designator on the work terms.

Pump work, which will be nonzero when intake pressure is different from exhaust pressure (nearly all situations), often decreases net work relative to gross work (i.e., the gas exchange process requires the piston to do work on the gas, or, pump work is negative). There are a few situations, with the use of a turbocharger or

Table 6.1 Summary of various mean effective pressures describing the various work transfers defined for a reciprocating-piston internal combustion engine

Name	Definition
Gross indicated mean effective pressure (gross IMEP)	$IMEP_g = W_{ig}/V_d$
Net indicated mean effective pressure (net IMEP)	$IMEP_n = W_{in}/V_d$
Pump mean effective pressure (PMEP)	$PMEP = W_{ip}/V_d$
Friction mean effective pressure (FMEP)	$FMEP = W_f/V_d$
Brake mean effective pressure (BMEP)	$BMEP = W_b/V_d$

supercharger for example, when intake pressure is greater than exhaust pressure and pump work is positive. In such situations, net work will be greater than gross work.

The difference between net work and brake work, as shown in Eq. 6.5, is the friction work, W_f, of the engine. Friction, of course, always requires work from the system; thus, brake work will always be less than net work. In Eq. 6.5, W_f captures all forms of mechanical friction of the engine, including friction among crank-slider components, in bearings, in valve springs, and in various accessories (e.g., water and oil pumps):

$$W_f = W_{in} - W_b \tag{6.5}$$

Engine researchers typically quantify all of the above-described work parameters on volume-normalized parameters, which in general represent a "mean effective pressure." The general definition for mean effective pressure, MEP, is given as Eq. 6.6:

$$MEP = \frac{W}{V_d} \tag{6.6}$$

The various mean effective pressures and their definitions are summarized in Table 6.1.

One of the major benefits of describing the work of an engine in terms of mean effective pressure is that the "size" of the cylinder is removed from consideration. In other words, it is possible to produce more work from an engine that has a larger displaced volume; however, the mean effective pressure may be lower (relative to a lower displaced volume engine) due to a number of other possible influencing parameters that affect and engine's ability to make power (i.e., fuel conversion and volumetric efficiencies, fuel–air ratio, inlet air density, and fuel heating value). To make these types of assessments, refer to Eq. 6.7, which is developed from the basic definition of power, P (i.e., work per unit time), and respective definitions of other involved parameters as shown by Heywood [11]. An example of the use of Eq. 6.7 to assess factors affecting power in a technology comparison is provided in [12].

$$P = \frac{\eta_f \eta_v \rho_{a,i}(F/A)Q_{HV}V_d N}{n_R} \tag{6.7}$$

where

η_f Fuel conversion efficiency (described in section on "Thermodynamic Analysis of Internal Combustion Engines").

η_v Volumetric efficiency, or the engine's effectiveness at "breathing" air. It is defined as $\eta_v = \frac{m_a}{\rho_{a,i} V_d}$, where m_a is the actual mass of air inducted per cycle (kg) and $\rho_{a,i}$ is the density of the intake air at some reference point (kg/m^3) (usually the intake manifold, but also may be atmospheric air upstream of the engine air filter). Note that V_d has units of (m^3) in Eq. 6.7. It is important to note that volumetric efficiency only quantifies the engine's ability to breath air, i.e., not a fuel–air mixture. Thus, for example, premixing fuel with air prior to induction tends to lower the engine's volumetric efficiency. It is also important to note that volumetric efficiency of the engine depends on the chosen reference point. Thus, if the volumetric efficiency of the whole breathing stream is desired, the reference point is taken as upstream of the intake air filter (for example). If, however, volumetric efficiency of just the intake ports and through the valves is desired, then the reference point is taken as the intake manifold.

$\rho_{a,i}$ As defined above, the density of the intake air at some reference point (kg/m^3).

F/A The mass-based fuel–air ratio of the mixture.

Q_{HV} Heating value of the fuel (typically, lower heating value is used since the products leaving the piston/cylinder system with water as a vapor) (kJ/kg).

N Engine speed (rev/s)

To reveal the form of mean effective pressure, Eq. 6.8 is given as a modified form of Eq. 6.7.

$$\mathrm{MEP} = \eta_f \eta_v \rho_{a,i} (F/A) Q_{HV} \qquad (6.8)$$

Thus, it is apparent from Eq. 6.8 how an engine with a relatively small displacement might have a higher mean effective pressure than an engine with a large displacement, even though the larger-displaced engine may produce more power. It is also clear from Eqs. 6.7 and 6.8 how the performance (i.e., power) of an engine may be improved beyond the "easy" action of increasing displaced volume. One parameter, for example that benefits the power and mean effective pressure of the engine is the fuel conversion efficiency. This very important parameter is discussed in the next section.

Finally, this section concludes by illustrating a typical power/torque/speed curve. Apparent from Eq. 6.3 and the definition of power, there is a relationship among power, torque, and speed of an engine. This relationship for a typical internal combustion engine is shown in Fig. 6.5. There are a few interesting features to point out in this figure. The first, a practical feature, is the identification of "rated torque" and "rated power". Engines are usually "sized" based on the speed at which
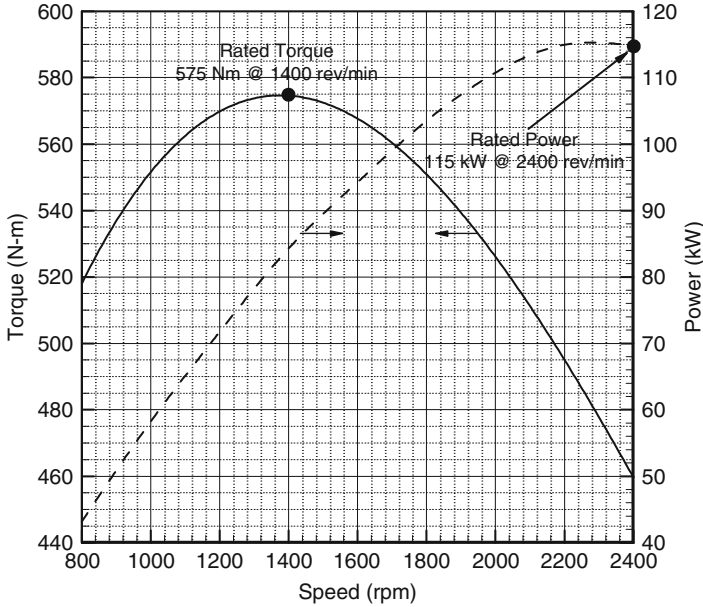
Fig. 6.5 Torque and power as functions of engine speed for a typical internal combustion engine. Data is collected from 4.5 L medium-duty diesel engine with advanced technology such as turbocharging and exhaust gas recirculation (Data from author's laboratory, Texas A&M University)

they develop maximum torque (also known as rated torque) and the speed at which they develop maximum power (also known as rated power). The second feature, which is apparent from the knowledge that rated torque exists, is the seemingly dependent relationship of torque on engine speed. It is clear from the definition of power that it should have a functional relationship on the speed of the engine, which dictates the amount of work per unit time the engine can deliver. But, based on assessment of Eq. 6.3, there is not a direct relationship between in-cylinder work per cycle and the speed of the engine. In fact, in an ideal sense, the torque of the engine should be constant with engine speed (and, if constraining to an ideal engine, higher than the maximum attained torque of the real engine), similar to how torque of an electric motor is nearly constant with motor speed. Thus, the behavior seen in Fig. 6.5 suggests that there exist factors in a real engine that depend on engine speed and also affect the work produced per cycle (thereby affecting the torque of the engine). These factors predominantly consist of heat transfer and friction; both of which have dependencies on the speed at which the engine operates. Heat transfer of course is a time-dependent phenomenon. At low engine speeds there is more time per cycle for thermal energy to be transferred from the cylinder; thus, torque droops at low engine speeds as heat transfer diminishes the energy available for in-cylinder work production. Friction, too, is a speed dependent function due to the mechanical behavior of the engine's interacting components. At high engine speeds, there is

Fig. 6.6 Example of volumetric efficiency, illustrating the many factors that impact the engine's ability to effectively breathe air (Used with permission from [13]). Improvements to the engine's volumetric efficiency can yield substantial improvements to its ability to make power, evident from Eq. 6.7
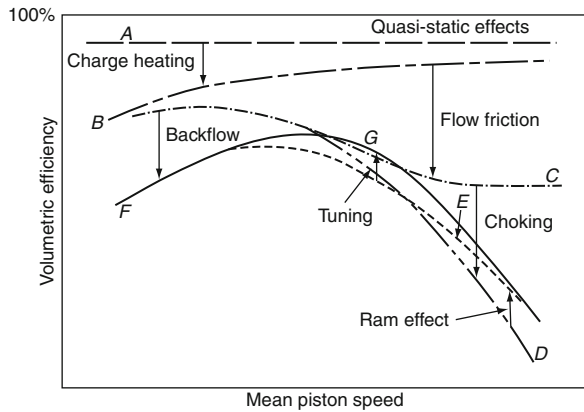
increased friction to be overcome on a per cycle basis; thus torque droops at high engine speeds as increased work energy is required to overcome increased friction.

Relating this discussion to Eq. 6.7, both heat transfer and friction effects tend to decrease η_f. There is, however, another factor of Eq. 6.7 being influenced by heat transfer and friction and thus serving as a major contributor to the behavior of the torque curve shown in Fig. 6.5; this factor is the volumetric efficiency. The maximum amount of work that can be developed per cycle is firstly dependent on the amount of air (or oxidizer) the engine can breathe; the amount of air the engine can breathe ultimately dictates the amount of fuel that can be delivered, which of course serves as the energy carrier to be converted in the cylinder.

A representative volumetric efficiency curve as a function of mean piston speed (which is correlated to engine speed) is shown in Fig. 6.6. There is much important detail in this figure, and the following will describe this in detail; after this discussion, a return to the effect of volumetric efficiency on the torque curve of Fig. 6.5 will be made. First, notice that under ideal situations the volumetric efficiency curve would be 100% and independent of engine speed (like work, trapped mass per cycle ideally has no functional dependency on the number of cycles per unit time, or, speed of the engine). It is noted that, depending on the reference point chosen for quantifying it, volumetric efficiency could be greater than 100% if the engine is, for example, boosted. In such a situation, the chosen reference point is upstream of the boosting device (e.g., turbocharger compressor inlet). Although such a chosen reference point is useful to indicating the quantity of trapped mass in the cylinder, it prevents the quantification of the engine's breathing system outside of the boosting device, and masks opportunities for further improvement to the engine's design.

Returning to the volumetric efficiency curve of Fig. 6.6, and for the specific case of premixed charge engines such as conventional gasoline spark-ignited engines, the premixing of fuel with air prior to induction diminishes the amount of trapped mass of air in the cycle. Such a factor is considered a "quasi-static" effect since it will be present regardless of the speed of the engine. Other quasi-static effects

include factors such as, for example, residual fraction due to manifold pressure differences (between exhaust manifold and intake manifold) and exhaust gas recirculation. Thus, notice secondly that the diminishing effect of quasi-static factors is represented as curve "A" in Fig. 6.6. The next phenomenon to be captured in the volumetric efficiency curve is called "charge heating" and is a heat transfer effect. During normal engine operation, the breathing system of the engine reaches a steady state temperature that is higher than the ambient temperature. Thus, the elevated temperature of the intake port results in heat transfer to the intake air, increasing the air temperature, decreasing its density, and decreasing the amount of mass per unit volume. As described above, heat transfer is a time-dependent phenomenon, therefore its presence is most notable at low engine speeds (low mean piston speeds); notice thirdly in Fig. 6.6 the effect of charge heating on the volumetric efficiency curve, which in combination with quasi-static effects lowers it to the curve labeled "B." Similar to how friction affects work per cycle, friction also affects air flow and has a dependency on mass flow rate (i.e., has a dependency on engine speed). The fourth factor to notice in Fig. 6.6 is the effect of flow friction; its effect in concert with quasi-static and charge heating factors result in the curve labeled "C." Notice at very high speeds, the flow friction effect seems to level off and play a speed-independent role as engine speed increases; this results from the flow attaining choke conditions where, regardless of the pressure drop across the intake system, the mass flow rate remains constant as its velocity reaches the speed of sound. Thus, the fifth factor to notice in Fig. 6.6 is the effect of choking on the volumetric efficiency curve. The net result of all combined effects including choke is the curved labeled "D." Interestingly, some phenomena help to increase the engine's volumetric efficiency; ram effect is one such example. Ram effect occurs at high flow rates where fluid momentum results in continued charging of the cylinder even as the in-cylinder motion of the piston no longer provides the pumping action (i.e., volume increase during intake stroke). The sixth factor to note in Fig. 6.6 is the increase in volumetric efficiency due to ram effect. The combined effects, including ram effect, result in the curve labeled "E." A similar but diminishing gas exchange phenomenon occurs at low engine speeds, known as backflow. Backflow occurs during the valve overlap period (the period during which both exhaust and intake valves are open as exhaust valves close and intake valves open) where in-cylinder motion becomes quiescent as the piston reaches top dead center and exhaust products of combustion backflow into the intake port. The seventh factor to notice in Fig. 6.6 is the effect of backflow on the engine, which tends to pervade at low engine speeds where exhaust momentum is low. The combined effects, including backflow, on volumetric efficiency result in the curve labeled "F." Finally, to end on a positive note, the eighth and last factor to observe in Fig. 6.6 is that of tuning. Tuning is the effort to make use of established resonating sound waves in the intake and exhaust systems to either aid in charging the cylinder (intake tuning) or discharging the cylinder (exhaust tuning). Because this charging or discharging benefits relies upon the timing of compression and rarefaction waves being positioned at precise locations in the intake or exhaust systems, fixed geometry intake and exhaust systems can only be tuned at a narrow

speed range. The benefit of tuning is shown in Fig. 6.6, where for this particular application tuning is designed for the mid-speed range of the engine. Variable tuning systems are becoming available in production, which through the use of creative flow channeling and blocking, can change the "effective" runner length of the flow passage and allow for tuning across broader speed ranges than allowed by fixed geometry runners. With all factors considered, the final volumetric efficiency curve is the solid curve labeled "G."

Now that the details of the volumetric efficiency curve as a function of engine speed, identified as curve "G" in Fig. 6.6, have been discussed, a return to its effect on engine torque will be made. In fact, it is interesting to note the similarities between the final volumetric efficiency curve (curve "G") of Fig. 6.6 and the torque curve of Fig. 6.5. Such a similarity is not a coincidence, as the maximum amount of work attainable per cycle is a strong function of how well the engine breathes air per cycle. Correspondingly, based on the assessment of Eq. 6.7, significant improvements to an engine's ability to make power can be realized through creative improvements to the engine's volumetric efficiency.

Thermodynamic Analysis of Internal Combustion Engines

At its core, an internal combustion engine is a thermodynamic device. That is, it takes one form of energy (i.e., chemical energy) and converts it into another form of energy (i.e., mechanical "shaft work" energy). There are several considerations that should be given to the thermodynamics of the internal combustion engine, in pursuit of improving its ability to make power and, more importantly, to make power *efficiently*. Efficiency is an important thermodynamic concept which quantifies the ability of a work conversion device to convert one form of energy (e.g., chemical energy) into work energy (i.e., mechanical shaft work). In the context of internal combustion engines, *fuel conversion efficiency* is used, as given by Eq. 6.9:

$$\eta_f = \frac{W}{m_f Q_{HV}} \tag{6.9}$$

where m_f is the mass of fuel per cycle. Typical maximum fuel conversion efficiencies for internal combustion engines vary from ca. 20% to 45%, depending on engine design and type of combustion system. Fuel conversion efficiency, like power, can be designated as brake or indicated fuel conversion efficiency. The relationship between brake and indicated fuel conversion efficiency is given as Eq. 6.10:

$$\eta_{f,b} = \eta_{f,i} \eta_m \tag{6.10}$$

where $\eta_{f,b}$ is brake fuel conversion efficiency, $\eta_{f,i}$ is net indicated fuel conversion efficiency, and η_m is mechanical efficiency and given by $\eta_m = \frac{W_b}{W_{in}} = \frac{P_b}{P_{in}} = \frac{\text{BMEP}}{\text{IMEP}_n}$. In words, fuel conversion efficiency is the ratio of work output to fuel energy delivered. There is another efficiency definition that can be considered, called *thermal efficiency*, which is the ratio of work output to fuel energy released. In other words, since the combustion process converts the fuel's chemical energy to thermal energy, the thermal efficiency indicates the conversion of thermal energy to work energy. It is not possible to precisely measure the energy released by the fuel during the combustion process; it is, however, possible to precisely determine the extent of incomplete combustion, or combustion inefficiency. Thus, *combustion efficiency* is introduced to allow for the determination of the engine's thermal efficiency. Combustion inefficiency is defined as the energy of unreacted fuel in the exhaust to fuel energy delivered to the engine. Combustion efficiency, η_c, is thusly given as Eq. 6.11 [14]:

$$\eta_c = 1 - \frac{\sum y_i Q_{HV,i}}{\left[\dot{m}_f \middle/ \dot{m}_f + \dot{m}_a\right] Q_{HV,f}} \tag{6.11}$$

where y_i is an unreacted fuel specie in the exhaust (e.g., carbon monoxide), $Q_{HV,i}$ is the heating value of specie i, $\dot{m}_f$ and $\dot{m}_a$ are the fuel and air mass flow rates, respectively, and $Q_{HV,f}$ is the fuel's heating value. Fuel conversion efficiency, combustion efficiency, and thermal efficiency, η_{th}, are all related by Eq. 6.12:

$$\eta_f = \eta_c \eta_{th} \tag{6.12}$$

Note that η_f of Eq. 6.12 could be either brake or indicated, in which case the corresponding η_{th} will have the same basis.

At this point, it is instructive to question what the maximum possible fuel conversion or thermal efficiency of an internal combustion engine could be. To begin to answer this question, it should be established that, contrary to what is sometimes simply conveyed, an internal combustion engine is not, by rigid definition, a *heat engine* [15–17]. A heat engine is a device that collects thermal energy, via heat transfer mechanism, from a high temperature reservoir and converts the *available portion* of this thermal energy to useful work. The portion of thermal energy that is not available for conversion to useful work (i.e., the entropy of the thermal energy) is rejected to a low temperature sink via heat transfer mechanism. A practical example of a heat engine is the steam engine. It is the steam engine that caused N.L. Sadi Carnot to establish his two postulates on the maximal conditions of the heat engine [18], thus laying the foundation for what would become the second law of thermodynamics and the mathematical axioms for the property entropy. The fully reversible heat engine cycle, i.e., the Carnot cycle, is the most efficient cycle for converting thermal energy into useful work (which, because of its reversible nature, is the most efficient cycle for converting work energy into

Fig. 6.7 (**a**) Pressure–volume and (**b**) temperature–entropy diagrams of the air-standard Otto cycle (Used with permission from [19])
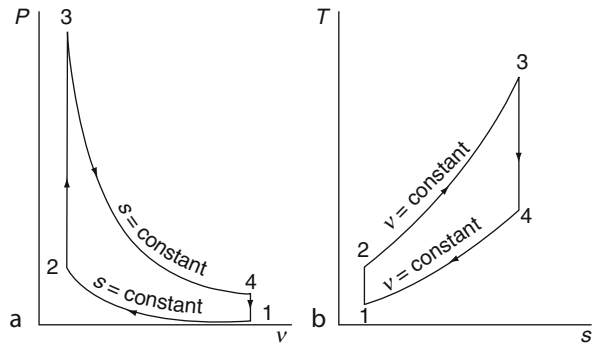

thermal energy). These details are provided because a common misconception is that the maximum efficiency of an internal combustion engine is limited by a so-called Carnot efficiency; this really has no technical basis because the internal combustion engine does not operate on the same principle (i.e., the heat engine principle) around which the Carnot cycle is created.

Perhaps much of the misdiagnosis of considering an internal combustion engine a heat engine comes from the use of ideal heat engine cycles to "model" an internal combustion engine; namely the Otto and Diesel cycles. These cycles do serve an important purpose in that certain fundamental parameters affecting the efficiency of the ideal heat engine cycles (i.e., Otto and Diesel cycles) transcend to also affecting the efficiency of the internal combustion engine. An example of such is compression ratio. Thermodynamic analysis of the Otto Cycle, the four processes of which are shown in Fig. 6.7, on (a) pressure–volume and (b) temperature–entropy diagrams reveal that efficiency of the cycle, $\eta_{\text{th,Otto}}$, is given by Eq. 6.13:

$$\eta_{\text{th,Otto}} = 1 - \frac{1}{r_c^{\gamma-1}} \tag{6.13}$$

where γ is the ratio of specific heats for the working fluid (e.g., $\gamma_{\text{air}} = 1.4$ at 300 K). This result reveals the asymptotic relationship of increasing thermal efficiency with compression ratio, as shown in Fig. 6.8. Similar behavior is observed in internal combustion engines (i.e., increasing efficiency with increasing compression ratio) as the capability to expand the working fluid (i.e., combustion products), and thus the ability to convert thermal energy to work energy, increases with increasing compression ratio. Another fundamental behavior is apparent Fig. 6.8 related to the changing value of γ. Notice that thermal efficiency tends to increase with an increase in γ; similar to the increase in compression ratio, a mixture with a higher γ suggests it requires more energy to increase its temperature during a work-transfer (i.e., constant pressure) process compared to a nonwork (i.e., constant volume) process. The corollary to this is more work energy is transferred out of a mixture with higher γ as the mixture is expanded. In internal combustion engines, leaner fuel/air mixtures (i.e., those with less than stoichiometric, or chemically complete,

**Fig. 6.8** Otto cycle thermal efficiency as a function of compression ratio for several γ values
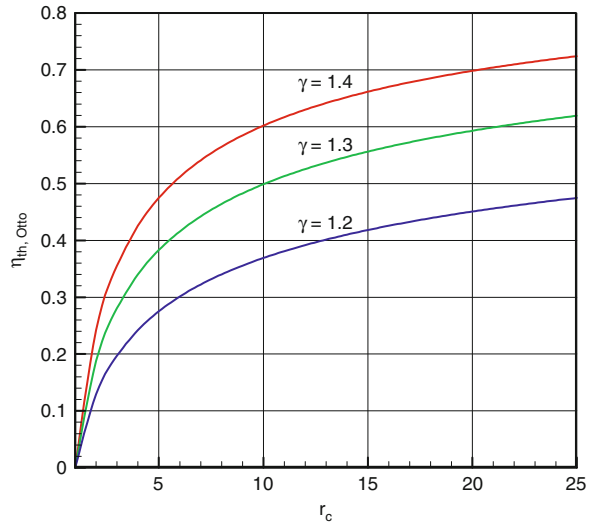
concentrations of fuel in the mixture) have higher γ values, and thus realize an efficiency improvement.

Although the examples of using the Otto cycle to assess the effects of compression ratio and γ value on thermal efficiency are helpful, the Otto cycle efficiency cannot be used to predict maximum attainable efficiency limits of an internal combustion engine. There are several inaccuracies in applying the Otto cycle (or any ideal heat engine cycle) to predict internal combustion engine performance and efficiency. Specifically, the assumptions that go into the development of Eq. 6.13 are as follows (refer to Fig. 6.7):

1. Processes 1–2 and 3–4 are *reversible and adiabatic* (i.e., *isentropic*, or constant entropy) processes.
2. Process 2–3 is a constant volume *heat addition* process, where thermal energy of the system increases due to heat transfer with the surroundings at a temperature of no less than T_3.
3. Process 4–1 is a constant volume *heat rejection* process, where thermal energy of the system decreases due to heat transfer with the surroundings at a temperature no greater than T_1.
4. Properties of the working fluid are constant throughout the cycle.

Of the above four assumptions, all four are impractical (meaning, an actual engine's efficiency will be less than the ideal limit because of real effects, which will be discussed in more detail below). Further, assumptions 2–4 are theoretically improper to impose on an internal combustion engine. Assumptions 2–4 imply a heat engine cycle, which has already been established that an internal combustion engine is not. Instead, an internal combustion engine operates as a collection of processes, all which have ideal limits dictating, in aggregate, the ideal maximum efficiency of the engine. Energy is transferred to the control system (i.e., the

Fig. 6.9 Pressure–volume diagram illustrating the processes assumed in *Fuel–Air Cycle Analysis* of an internal combustion engine. Note that shown cycle is representative of an engine under "throttled" conditions, where intake pressure is less than atmospheric (Used with permission from [21])
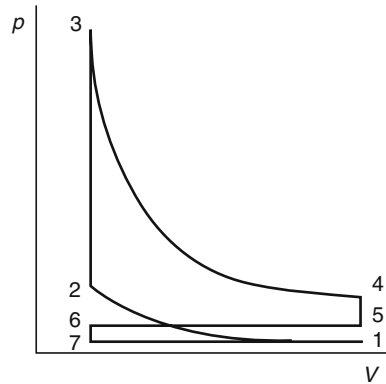
cylinder of the engine) through mass flow rate, as opposed to heat transfer mechanism (as in the heat engine apparatus). This energy, initially in the form of chemical energy, is converted to thermal energy during the combustion process and the expansion of the piston thereby converts the thermal energy to work energy. Additionally, because of the change from reactants to products during the combustion process and the dependence of species' properties on temperature and pressure, it is not appropriate to assume properties of the working substance remain constant during the engine's operation (i.e., assumption no. 4 is not valid).

Instead, maximum ideal efficiency limits of internal combustion engines could be assessed by improving upon the inappropriate assumptions of the Otto heat engine cycle, i.e., allow energy to be transferred into the system via mass flow (i.e., via the intake process) and allow properties of the mixture to change with species, temperature, and pressure. The resulting analysis, referred to as *Fuel–Air Cycle Analysis* [20], employs the following processes (refer to Fig. 6.9) and assumptions in modeling an internal combustion engine cylinder (i.e., the control volume):

Process 6–7–1: Ideal intake process of fuel and air and adiabatic mixing with residual gas from the preceding cycle. Residual gas, or often referred to as "residual fraction," is left-over products of combustion not fully exhausted from the cylinder during the previous cycle's exhaust displacement process.

Process 1–2: Reversible and adiabatic compression of the fuel/air/residual mixture (i.e., the reactants). Chemical species are frozen.

Process 2–3: Complete, adiabatic combustion of reactants to products (the latter of which are assumed to exist in equilibrium concentrations). Combustion, in ideal sense, is modeled as constant volume, constant pressure, or limited pressure. Limited pressure is a combination of constant volume and constant pressure processes. Constant volume combustion occurs until a "limit pressure" is reached, when remainder of combustion occurs at constant pressure.

Process 3–4: Reversible and adiabatic expansion of products, which remain in chemical equilibrium throughout the process.

Process 4 – 5: Reversible and adiabatic "blowdown" of products to exhaust pressure of products remaining in the cylinder. Products remain in fixed composition based on concentrations at State 4.

Process 5–6: Ideal, constant pressure, and adiabatic exhaust displacement of products, remaining in fixed composition.

Ideal gas behavior of the mixture and conserved mass are assumed throughout the analysis. Further, mass transfers only occur during intake and exhaust processes (i.e., no leakage, blow-by, or crevice flow during the cycle). By employing first and second laws and various assumptions to each of the processes, respective states can be thermodynamically fixed, thus allowing determination of work transfers. An item to note about *Fuel–Air Cycle Analysis*, because it accommodates changing species, is that *three* independent properties must be known to fix the states (i.e., two thermodynamic properties, such as temperature and pressure, and species composition). Process 6–7–1 is assumed adiabatic. Further, it is assumed that during the stroke from TDC to BDC, the cylinder pressure is constant and equal to the intake pressure, P_i:

$$P_7 = P_1 = P_i$$

$$_6Q_1 = 0$$

With these two statements, and use of first law, the enthalpy at State 1, h_1, is given as:

$$h_1 = f\left[h_e + R_e T_e\left(\frac{P_i}{P_e} - 1\right) - h_i\right] + h_i$$

where f is residual gas fraction, h_e is enthalpy of the exhaust (and equal to $h_6 = h_5$), R_e is the gas constant of the exhaust (and equal to $R_6 = R_5$), T_e is temperature of the exhaust (and equal to $T_6 = T_5$), P_e is pressure of exhaust (and equal to $P_6 = P_5$), and h_i is the enthalpy of the fresh intake mixture. Residual fraction, as explained above, is the residual mass, m_r (which is equal to m_6, the mass at State 6), divided by the total cylinder mass, m_{total} (which is equal to $m_1 = m_2 = m_3 = m_4$, the mass of the system at States 1, 2, 3, and 4, respectively), and is related to r_c, P_e, T_e, and the pressure and temperature at State 4, P_4 and T_4, respectively, by the following:

$$f = \frac{m_r}{m_{total}} = \frac{m_6}{m_1} = \frac{m_6}{m_4} = \frac{1}{r_c}\frac{P_e T_4}{P_4 T_e}$$

It is clear from the equations for h_1 and f that a priori knowledge of the cycle is needed in order to fix State 1 (note that r_c, P_i, P_e, h_i are chosen based on desired cycle compression ratio, manifold conditions, inlet composition, and inlet temperature, T_i, respectively). Thus, initial values for State 1 are typically assumed for the first iteration; State 4 temperature, pressure, and composition from the initial iteration are

then used to estimate f and h_1 for the next iteration. This iterative process continues until some convergence criterion is met (e.g., close match between f_{final} and $f_{final-1}$). Rather than assume values of f and h_1, it is perhaps more intuitive to assume initial values of f and temperature at State 1, T_1; the following expressions [22] are approximations for establishing initial iteration State 1 properties:

$$f = \left\{ 1 + \frac{T_e}{T_i} \left[r_c \left(\frac{P_i}{P_e} \right) - \left(\frac{P_i}{P_e} \right)^{(\gamma-1)/\gamma} \right] \right\}^{-1}$$

$$T_1 = T_e r_c f \left(\frac{P_i}{P_e} \right)$$

It is also noted that composition of the residual gas must be assumed to fully fix State 1; it is suggested that ideal products of stoichiometric combustion of the fuel–air mixture be used as the residual gas species.

Work analysis of Process 6–7–1 gives the intake work quantity:

$$_6W_1 = P_i(V_1 - V_6)$$

Process 1–2 is assumed reversible and adiabatic (i.e., $_1Q_2 = 0$); from second law, this gives:

$$s_1 = s_2$$

Geometric relationship gives: $v_1 = r_c v_2$

The assumption that chemical composition remains fixed during compression (i.e., Process 1–2) gives chemical composition at State 2; thus, State 2 is fixed. First law analysis of Process 1–2 gives:

$$_1W_2 = U_2 - -U_1 = m(u_2 - u_1)$$

Process 2–3 is the adiabatic combustion process, where reactants become products (products assumed to be in chemical equilibrium) modeled typically in three simple fashions: (1) constant volume combustion, (2) constant pressure combustion, and (3) limited pressure combustion. First law analysis for each of the three cases results in the following, respectively:

1. Adiabatic constant volume combustion

   $u_3 = u_2$

   $v_3 = v_2$

   $_2W_3 = 0$
2. Adiabatic constant pressure combustion

   $h_3 = h_2$

$$P_3 = P_2$$

$$_2W_3 = P_2(V_3 - V_2) = P_3(V_3 - V_2)$$

3. Adiabatic limited pressure combustion

Constant volume portion

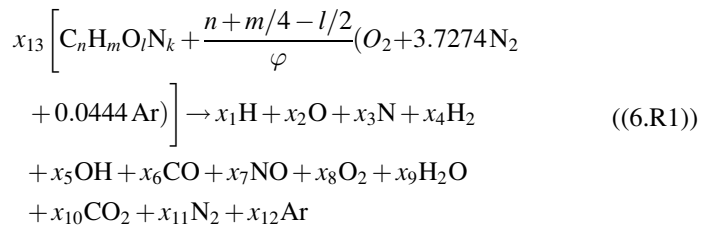$$u_{3a} = u_2$$

$$P_{3a} = P_{\text{limit}}$$

$$v_{3a} = v_2$$

Constant pressure portion

$$h_3 = h_{3a}$$

$$P_{3a} = P_3 = P_{\text{limit}}$$

$$_2W_3 = P_3(V_3 - V_2)$$

At this point, it becomes necessary to describe a way in which product species of combustion can be modeled. It is clear from the above discussion that combustion is modeled as adiabatic and (a) constant volume, (b) constant pressure, or (c) limited pressure. In order to capture a representative combustion reaction, it is necessary to assume that several product species may exist and will exist in equilibrium. Also, it is important to clarify that this approach assumes initial and final equilibrium states in predicting combustion; i.e., the thermodynamic process goes from one equilibrium state (reactants) to a different equilibrium state (products). This is in contrast to using a more detailed approach that models the progression of combustion and relies upon reaction kinetics to predict intermediate species and their nonequilibrium concentrations. See, for example, Foster and Myers [16] for overview of detailed engine modeling and, for example, Westbrook and Dryer [23] for overview of kinetic modeling of combustion. While perhaps hundreds of species could be included as products, Olikara and Borman [24] propose a combustion reaction, given as Reaction (6.R1), that might nearly be general for internal combustion engine purposes:

$$x_{13}\left[C_nH_mO_lN_k + \frac{n+m/4-l/2}{\varphi}(O_2 + 3.7274N_2\right.$$

$$\left. + 0.0444\,\text{Ar})\right] \rightarrow x_1H + x_2O + x_3N + x_4H_2 \qquad ((6.R1))$$

$$+ x_5OH + x_6CO + x_7NO + x_8O_2 + x_9H_2O$$

$$+ x_{10}CO_2 + x_{11}N_2 + x_{12}Ar$$

where x_1 through x_{12} are the mole fractions of the respective product species. x_{13} represents the moles of fuel per mole of products. Solution of the 13 unknowns of course requires 13 independent equations; five of these equations come from the species balance (i.e., balance of C, H, O, N, and Ar species). The remaining eight equations come from the assumption of equilibrium among various product species [24]. As a final note, it is recognized that NO appears as a product species; this represents the equilibrium concentration of NO at the reaction temperature and pressure. Because both the formation and decomposition of NO in the post-flame gas regions are rate limited [25], it is necessary to take a chemical kinetic approach to modeling IC engine exhaust NO concentrations (see, e.g., [26–29]).

Process 3–4 is reversible and adiabatic expansion with the mixture in chemical equilibrium (resulting in different species at State 4 relative to State 3), resulting in the following thermodynamic state (along with geometrical constraint imposed):

$$s_4 = s_3$$

$$v_4 = v_1$$

First law analysis gives the following expression, which is the expansion work of the cycle:

$$_3W_4 = U_4 - U_3 = m(u_4 - u_3)$$

Process 4–5 is the ideal blowdown process (i.e., mixture that remains in the cylinder expands isentropically, filling the cylinder volume voided by the products irreversibly escaping past the open exhaust valve). Species are fixed based on State 4 concentrations. Thus, State 5 is fixed with the following relationships:

$$s_5 = s_4$$

$$P_5 = P_e$$

Although specific volumes change during the blowdown process (due to mass transfer), total volume is constant; thus, work transfer is 0:

$$_4W_5 = 0$$

Finally, Process 5–6 is the ideal exhaust displacement process, i.e., cylinder contents are transferred out of the adiabatic control volume via piston displacement at constant pressure.

At this point, it is useful to describe the attainment of various property values indicated above (e.g., s_1). Because *Fuel–Air Cycle Analysis* seeks to capture the effects of changing properties on the cycle analysis, the approximations for constant-specific heat ideal gas property equations cannot be used. Instead, it becomes

necessary to employ techniques that accommodate changes to mixture properties based on the mixture's temperature, pressure, and species composition. One such technique is to use NASA's Chemical Equilibrium with Applications program [30–34] coupled with the JANAF Thermochemical Tables. By doing so, species at any given temperature and pressure can be predicted (using NASA's Chemical Equilibrium with Applications program) and the corresponding mixture's properties (using JANAF Thermochemical Tables) can be determined. This type of technique, for example, can be programmed into a computer routine.

Based on first law analysis and constant pressure assumption, State 6 is fixed as follows:

$$T_6 = T_5 = T_r$$

$$P_6 = P_5 = P_e$$

The exhaust work is given by the following:

$$_5W_6 = P_e(V_6 - V_5)$$

Based on the above analysis, the net work and gross work of the cycle are given by:

$$W_{net} = {}_6W_1 + {}_1W_2 + {}_2W_3 + {}_3W_4 + {}_5W_6$$

$$W_{gross} = {}_1W_2 + {}_2W_3 + {}_3W_4$$

Pump work is as defined by Eq. 6.4.

Further, various efficiencies can be calculated using Eqs. 6.9, 11, and 6.12. Note that although *Fuel–Air Cycle Analysis* is ideal, there may still be incomplete combustion due to the assumption of species existing in chemical equilibrium. Thus, combustion inefficiency will occur at fuel–air mixtures approaching stoichiometric and rich conditions, as dissociation creates concentrations of CO, H_2, and other partially oxidized species. Upon execution of the above analysis in a computer routine, for example, the theoretical limits of maximum efficiency of an internal combustion engine can be determined. Examples of such ideal efficiencies for constant volume combustion of isooctane/air mixtures at two different EGR levels are given in Fig. 6.10.

Figure 6.10 reveals interesting behavior of two important internal combustion engine parameters on the various efficiencies, i.e., fuel–air equivalence ratio (often designated as φ) and EGR level. First, notice that as φ approaches stoichiometric (i.e., $\varphi = 1$), η_{th} decreases, and then begins to increase as φ goes rich (i.e., $\varphi > 1$). Further, η_{th} increases as EGR level increases. To understand this behavior, it is necessary to understand how mixture properties are affected by (a) effect of varying φ on species, (b) effect of varying EGR level on species, and (c) effects of φ and
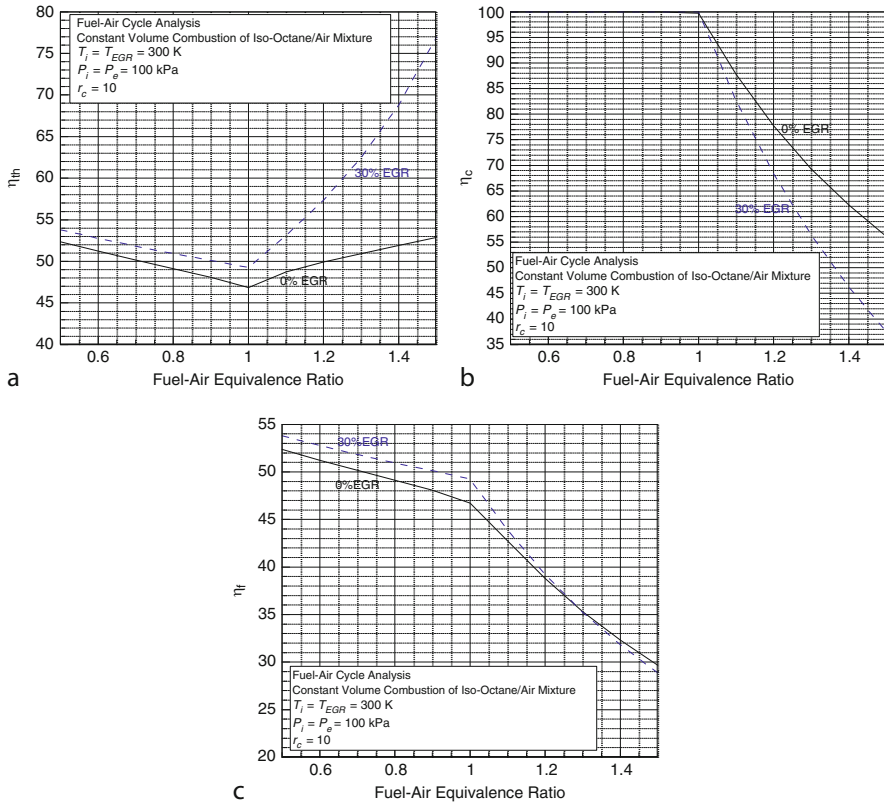
Fig. 6.10 (**a**) Thermal efficiency, (**b**) combustion efficiency, and (**c**) fuel conversion efficiency of an ideal fuel–air cycle analysis of an internal combustion engine assuming constant volume combustion of isooctane (C_8H_{18}) with $T_i = 300$ K, $P_i = P_e = 100$ kPa, $T_{EGR} = 300$ K, and $r_c = 10$

EGR level on temperatures and pressures of the cycle. Ultimately, as fundamentally revealed by Eq. 6.13, a change to mixture properties that results in an increase in γ will cause an increase in efficiency. Figure 6.11 [35] summarizes how burned gas mixture properties of an isooctane/fuel mixture are affected by various parameters such as φ and temperature. The behaviors of the parameters shown in Fig. 6.11 can be related to γ by recognizing that γ can be written as given by Eq. 6.14:

$$\gamma_b = \frac{1}{1 - \frac{\bar{R}}{M_b C_{p,b}}} \tag{6.14}$$

where γ_b is γ for the burned gas mixture, $\bar{R}$ is the universal gas constant (i.e., $\bar{R} \approx$ 8.314 kJ/kg-K), M_b is the molecular weight of the burned gas mixture, and $C_{p,b}$ is the constant pressure specific heat of the burned gas mixture. As an example of the type of analysis that might be done on Fig. 6.11, notice that M_b, (Fig. 6.11a)
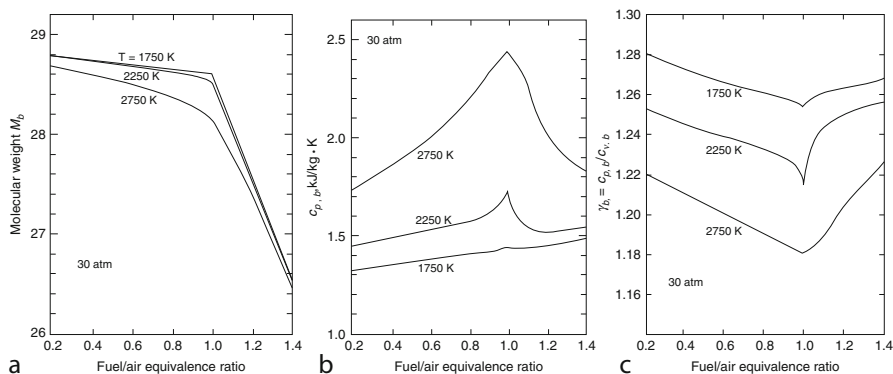
Fig. 6.11 Burned mass (**a**) molecular weight, (**b**) constant pressure specific heat, and (**c**) ratio of specific heats as functions of equivalence ratio of isooctane/air mixtures at various burned mass temperatures and pressure of 30 atm (Used with permission from [35])

decreases as φ increases (for any given temperature of the burned mixture); although concentrations of CO_2 ($M_{CO_2} \approx 44$ g/mol) increase as φ increases, so too do concentrations of H_2O ($M_{H_2O} \approx 18$ g/mol). Since isooctane, being a paraffin, produces more H_2O than CO_2 in the products, the lighter, higher concentration of H_2O dominates the net effect of decreasing M_b with increasing φ. As the mixture approaches $\varphi = 1$ and becomes rich ($\varphi > 1$), M_b decreases substantially as concentrations of partially oxidized species (e.g., CO and H_2) with relatively lighter molecular weights increase. Based on analysis of Eq. 6.14, a decrease in M_b will tend to increase γ_b, which tends to increase η_{th}. Also contributing to γ_b behavior, however, is $C_{p,b}$, which is shown in Fig. 6.11b. Notice that, up to about $\varphi = 1$, $C_{p,b}$ increases with φ. A portion of this increase is due to the increasing concentration of species with higher constant pressure specific heats as φ increases (e.g., At 1,750 K, $(C_p/R)_{H_2O} \approx 5.94$ and $(C_p/R)_{CO_2} \approx 7.15$ whereas $(C_p/R)_{N_2} \approx 4.28$ and $(C_p/R)_{O_2} \approx 4.46$, as shown in Fig. 6.12b); another portion, however, comes from the increase in burned gas temperature which causes an increase in constant pressure specific heat for most species. Specifically, as φ increases, adiabatic flame temperature and thus burned mixture temperature increases up to a peak that occurs slightly rich of stoichiometric (for isooctane; location of peak adiabatic flame temperature will vary depending on the type of fuel), as shown in Fig. 6.12a for isooctane/air–fuel cycle analysis with 0% EGR (note that adiabatic flame temperature, or $T_{CV,adiabatic}$ is equal to State 3 temperature, or T_3 of *Fuel–Air Cycle Analysis* and that T_3 is partially influenced by the changing State 2 temperature, or T_2, as φ varies). The effects of increasing burned gas temperature on the various species' constant pressure specific heats are shown in Fig. 6.12b. Thus as φ increases $C_{p,b}$ increases as species concentrations change (i.e., increased concentrations of CO_2 and H_2O) and burned gas temperature increases. An increase in $C_{p,b}$, like M_b, will cause γ_b to decrease. Thus, for φ less than stoichiometric, there are two competing effects on γ_b; a decreasing M_b and an increasing $C_{p,b}$ as φ increases. The net result, as shown
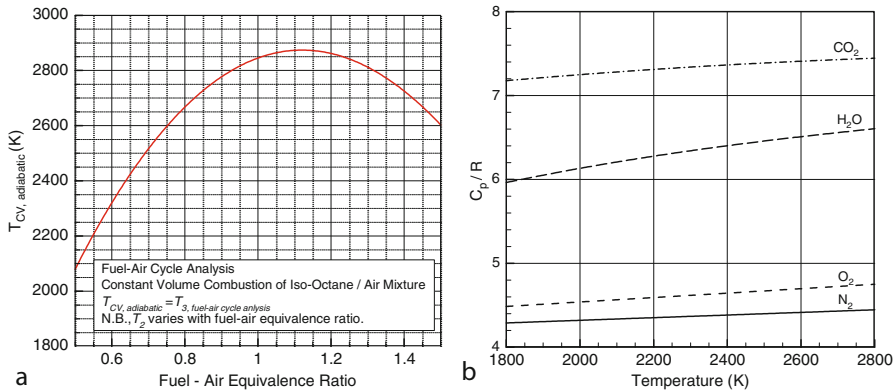
Fig. 6.12 (**a**) Constant volume adiabatic flame temperature as function of fuel–air equivalence ratio of isooctane with initial temperature and pressure as predicted by *Fuel–Air Cycle Analysis* of isooctane/air mixture and 0% EGR, and (**b**) C_P/R for various species typically found in burned mixtures as function of temperature (Data adapted from JANAF *Thermodynamic Tables* [36])

in Fig. 6.11c, is a decrease in γ_b which correspondingly explains the decrease in η_{th} as φ increases up to stoichiometric (see Fig. 6.10a).

The behaviors of M_b and $C_{p,b}$ change as φ increases beyond $\varphi = 1$ (i.e., the mixture becomes rich), causing a corresponding change in γ_b and ultimately in η_{th}. As explained above, M_b decreases more dramatically rich of stoichiometric due to the abundantly increasing concentrations of CO, H_2, and other dissociated species which have lower molecular weights than the fully associated (i.e., CO_2 and H_2O) species. The dramatic decrease in M_b generally dominates any potential increase in $C_{p,b}$, resulting in a net increase in γ_b. Thus, as φ becomes larger than 1, η_{th} increases. Further, at certain temperatures (e.g., 2,750 K) $C_{p,b}$ decreases as φ becomes greater than 1; the higher temperature of the burned mixture causes higher dissociation, which causes increased concentrations of dissociated species (e.g., CO and H_2), which have lower constant pressure specific heats than the fully associated species (e.g., at 2,750 K, $(C_p/R)_{H_2O} \approx 6.6$ whereas $(C_p/R)_{H_2} \approx 4.4$). For such temperatures where $C_{p,b}$ simultaneously decreases along with M_b as φ becomes larger than 1, there is a dramatic increase in γ_b and a corresponding dramatic increase in η_{th}.

Although η_{th} may tend to increase as φ becomes larger than 1, η_c tends to decrease (again, as concentrations of dissociated species such as CO and H_2 increase as φ becomes larger than 1). The result is a general decrease in η_f. Similar analysis can be conducted to understand the effect of EGR on the various efficiencies. It becomes necessary at this point, however, to assess the extent of the utility of such "first law analysis" – i.e., an analysis centered only on energy transfer – in pursuing future developments of internal combustion engines. For example, an engineer may "dream" of a situation where an engine operates rich at a fuel–air equivalence ratio of 1.4 and 30% EGR, yielding a η_{th} of nearly 70%; this certainly outperforms an engine operating lean at a fuel–air equivalence ratio of 0.5

and 30% EGR (see Fig. 6.10a). If only an energy-analysis is conducted to assess this "dream," the idea will immediately be discounted because the corresponding η_f for the engine running at 1.4 fuel–air equivalence ratio and 30% EGR is only ca. 32% (compared to the lean operation which is close to 54%). But, the products of the rich combustion still have useful energy, that is, energy that can be converted to useful work. While it is true the cycle cannot convert such *available* energy to useful work, it is more than a "dream" that some other device could convert it (e.g., a fuel cell or a thermoelectric generator). Thus, it might be conceivable to design an engine that allows it to operate at maximal η_{th} and couple it to another device that, while not having as high of a η_{th}, has a high enough η_{th} to suitably convert the available chemical energy to useful work, so that the net efficiency of both devices is greater than the η_f of the internal combustion engine operating at the lean condition (for example).

Analysis of such "dreams" can only be completed by (a) introducing the concept of *available energy*, or modernly called *exergy*, by employing second law considerations on the analysis and (b) introducing real effects on the engine analysis that include such factors as heat transfer, friction, and other energy "losses" – but not necessarily exergy losses (friction is, of course, an exergy loss). Thus, a brief discussion will follow describing the major benefit of conducting an exergy analysis of an engine system. The idea of available energy – or, that portion of energy which is available to do useful work – is introduced by Gibbs [37] and, in its basic form, suggests that a system with a given amount of energy can transfer that energy out of the system as useful work until the point of equilibrium between the system and its environment (i.e., its surroundings). Using a simplified example, a piston/cylinder arrangement that contains a gas at 2 atm. will, if allowed to interact with its surroundings at 1 atm., expand and transfer work energy until the system pressure is in equilibrium with the surrounding's pressure (i.e., 1 atm.). In a more theoretical sense, it captures the notion that only an orderly flow of energy (e.g., like that established because of a temperature gradient between a system and its surroundings) can be converted into useful work; disorderly energy cannot be converted to useful work. Entropy is the idea of disorder within a system; thus, exergy analysis combines the effect of a system's entropy on its ability to convert its energy into useful work. Because real processes in net increase entropy (i.e., real processes are irreversible and result in entropy generation), systems undergoing real processes destroy exergy; that is, the opportunity to convert energy into useful work is destroyed via some disordering of what had been an orderly flow of energy.

Examples of real effects, or irreversibilities, either in a general thermodynamic system or between a general thermodynamic system and its environment (all of which happen to be present, also, in internal combustion engines) are heat transfer through a finite temperature difference, mixing, unrestrained expansion, combustion, and friction. The general premise of an irreversibility is one which renders a system/environment combination changed as the system undergoes a cycle and returns to its initial state. For example, if a warm body exchanges thermal energy with a cold body (i.e., heat transfer through a finite temperature difference), there is heat transfer, but not work transfer, until the two bodies are in thermal equilibrium.

To return the two bodies to their initial states (i.e., transfer thermal energy from the cold body to the warm body), work energy must be transferred into the system to effect the heat transfer. In the forward process (i.e., the bodies attaining thermal equilibrium), heat transfer occurs without work transfer; in the reverse process (i.e., the bodies returning to their initial states), heat transfer is manifested by a necessary work transfer. A net effect has been made to the surroundings (transfer of work energy) as the system is returned to its initial state; thus, the process is irreversible. If, on the other hand, a device had been positioned between the two bodies of dissimilar temperature that was able to extract the orderly sense of energy motion into useful work (i.e., a heat engine), the irreversibility of the heat transfer through finite temperature difference is diminished. Further, if the heat engine is conceived to be ideal itself, then all the orderly sense of energy motion is converted into useful work; hence, the concept of the Carnot heat engine, or, fully reversible heat engine.

On the one hand, the heat transfer is viewed as a loss; on the other hand, it becomes apparent that there may still be an opportunity to extract useful work from the heat transfer. It is the latter supposition that exergy analysis centers on. To begin a discussion of energy and exergy analysis of an internal combustion engine, a revisit to the assumptions made in the *Fuel–Air Cycle Analysis* is necessary. First, the assumption that the engine is adiabatic is not realistic; all real engines have heat transfer. Thus, it becomes necessary to capture an understanding of the effects of heat transfer on the engine cycle. Second, the assumption of how combustion is modeled is inaccurate. Of course, constant volume combustion in an operating engine is not physically possible due to the finite time required by chemical kinetics to decompose the fuel molecule. Even constant pressure combustion is not realistic because it supposes a constant burn rate with instantaneous starts and ends of combustion. Thus, it becomes necessary to include the effects of "real" combustion on the engine cycle. Third, if it is desired to know the actual torque from the engine (as opposed to what is indicated by the pressure–volume relationship), then a sense of friction must be included in the analysis.

The inclusion of "real effects" in engine cycle analysis is well established. For example, comparisons between actual engine cycles and *Fuel–Air Analysis Cycles* are made by Edson and Taylor [38]. Inclusion of heat transfer, finite combustion rates, and mechanical losses is demonstrated by Strange [39]. The beginnings of detailed combustion modeling, including spatial location of start of combustion and flame propagation, are demonstrated by Patterson and Van Wylen [40]. Such early demonstrations, and the corresponding advancement in digital computers, result in advanced models of heat transfer (e.g., [41–43]), friction [44, 45], and combustion; the latter of which are described in the context of doing a complete engine cycle simulation (e.g., [44, 46–51]). The influences of "real effects" on both energy and exergy perspectives of internal combustion engines are also well established (see, e.g., the reviews by Caton [52] and Rakopoulos and Giakoumis [53]). For example, Fig. 6.13 illustrates both the energy (Fig. 6.13a) and exergy (Fig. 6.13b) distributions of using 0% and 20% adiabatic EGR in a spark ignition engine, as computed by a thermodynamic simulation [54]. Note that when real factors, such as heat transfer and real combustion, are considered, the maximum indicated fuel
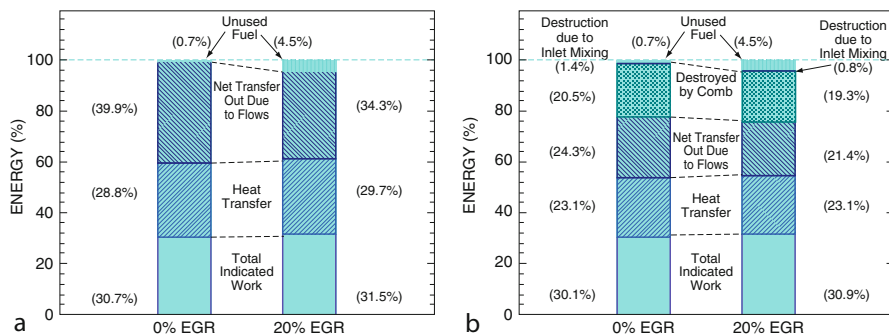
Fig. 6.13 (**a**) Energy distribution and (**b**) exergy (availability) distribution of various transfer and destruction (in the case of exergy) mechanisms for 0% EGR and 20% EGR in a spark ignition engine modeled using a thermodynamic simulation [54] (Used with permission from Professor J. Caton, Texas A&M University)

conversion efficiency is about 30.7% for non-EGR case (0% EGR) and 31.5% for 20% EGR case. Most of the balance of energy is transferred to the coolant (ca. 29% for non-EGR case) due to nonadiabatic control system and out the exhaust (ca. 40% for non-EGR case) due to underutilized conversion of thermal to work energy. Thus, the importance of considering real effects is clear. It is noted that the data shown in Fig. 6.13 do not include the effect of friction; the brake fuel conversion efficiency for the studied conditions of Fig. 6.13 are available in [54]. It is further noted that friction offers no opportunity for conversion to useful work (i.e., energy lost to friction is all destroyed exergy). Finally, it is noted that EGR seems to have a small to negligible effect on the calculated friction of the simulation [54].

Also clear from Fig. 6.13 is the importance of considering the internal combustion engine from an exergy perspective. If considering only the energy transfers of Fig. 6.13a, it may appear that ca. 70% of the energy is "lost," or unavailable for conversion to useful work. While it is correct to say that a fraction of the fuel's energy is underutilized in the engine control system's (i.e., piston/cylinder chamber) conversion to useful work, it is not correct to say that all this energy is lost. As explained above, exergy provides insight into the fraction of energy that is available to do useful work. For example, Fig. 6.13 illustrates that (for non-EGR cases) 28.8% of the fuel's *energy* is transferred out of the control system via heat transfer, whereas only 23.1% of the fuel's *exergy* is transferred out of the control system via heat transfer. That is, for one unit of fuel, 0.288 unit of energy is transferred via heat transfer, but only ca. 80% (0.231/0.288) of that energy is available to do useful work; thus, 0.231 unit of fuel exergy are a missed opportunity for conversion to useful work. This 0.231 unit of missed opportunity could be exploited by configuring an ideal heat engine between, for example, the cylinder walls and the environment (i.e., representative T_H and T_L). Such a conceptualization seems impractical, but exergy is not just passed into the coolant. Exergy is also transferred to the exhaust; ca. 24.3% of the fuel's exergy for the non-EGR case under study in Fig. 6.13b. The high temperature exhaust, with ease of accessibility, offers an

opportunity to exploit the fuel's exergy transferred through the exhaust system. There are several practical technologies that offer an opportunity to extract the fuel's exergy transferred out of the engine control system; these are discussed in more detail in the section "Waste Heat Recovery".

One of the important features of Fig. 6.13b is the exergy destroyed due to combustion. Combustion, in any application, is an irreversible phenomenon and thus will render entropy generation (i.e., exergy destruction). Like friction, the ca. 20% of fuel exergy destroyed due to combustion cannot be recovered for conversion to useful work. The major causes for exergy destruction of a typical hydrocarbon-based combustion process include (1) thermal energy exchange among particles within the system, (2) diffusion among fuel/oxidizer particles, and (3) mixing among product species [55]; of these three, the dominant exergy destruction source is the thermal energy exchange among particles. At the onset of combustion, with the system containing reactants, a metastable equilibrium exists as gradients exist between fuel and oxidizer molecules. As combustion proceeds during the process, natural phenomena cause the system to eliminate such gradients and maximize entropy in the pursuit of attaining a more stable equilibrium (i.e., the products state). The maximization of entropy in reducing the gradients is manifested entirely through entropy generation. Using the dominant source of exergy destruction (i.e., entropy generation) during combustion as an example: internal thermal gradients among particles [56] established during the combustion process create microscopic opportunities to do useful work (i.e., thermal gradients can be converted to useful work via a heat engine). Of course, practical implementations of microscopic heat engines do not exist, thus, the thermal gradients are reduced to zero during the combustion process without any conversion to useful work; heat transfer through a finite temperature difference, as described above, generates entropy (destroys exergy).

The general behavior of exergy destruction due to combustion in internal combustion engines is generally well characterized, as summarized by [57]; generally, an increase in combustion temperature decreases exergy destroyed due to combustion. This statement should not be confused with the situation of internal thermal gradients which, as mentioned above, are a major source of combustion-based exergy destruction. While it is tempting to seek ways to minimize (or even eliminate) combustion irreversibilities (and thus minimize or eliminate combustion-based exergy destruction), it should be recognized that such efforts may decrease the conversion of exergy in other ways. Consider, for example, the use of a lean mixture versus the use of a stoichiometric mixture; the former has higher exergy destruction than the later. While a stoichiometric mixture reduces exergy destruction, its mixture composition also causes lower conversion of thermal to work energy during the expansion process (and, coincidentally, increases exergy transfer via heat transfer). Thus, overall efficiency is lower with a stoichiometric mixture. Again, energy- and exergy-based analyses, such as that presented in Fig. 6.13, are necessary to quantitatively make such assessments. In the case of eliminating combustion irreversibilities, a reversible combustion process [58] can be conceptualized [59, 60] where initially separated reactant species are

isentropically compressed to their respective partial pressures and a certain temperature; concentrations of individual species, when allowed to interact with each other, will be in equilibrium. After compression, individual species will be collected forming a mixture that is a priori in equilibrium (thus, there are no irreversibilities due to mixing) and allowed to expand isentropically. Because of the increase in moles of the mixture as it expands isentropically maintaining equilibrium along the path, net work is extracted from the collection of processes. Quantitative analysis [60] of such a concept reveals 0% exergy destruction; because of exergy retention in the exhaust products, however, thermal efficiency of the "reversible combustion engine" is around 28% (for a fuel–air equivalence ratio of 1.0, compression pressure and temperature of 10 MPa and 6,000 K, respectively, and an expansion ratio of 18:1). Of course, such concepts are presently impractical; but the notion of reversible combustion is theoretically possible.

In fact, it is briefly noted that, although not explicitly calling his concept one of "reversible combustion," a close inspection of Diesel's original engine design [61] describes a method of nearly attaining reversible combustion (at least, one which eliminates internal thermal gradients). This is described in more detail in section "A Case Study: Diesel Engines Versus Gasoline Engines".

In closing this section on "Thermodynamic Analysis of Internal Combustion Engine", it is noted that assigning a "maximum possible efficiency" of an internal combustion engine is anything but straightforward. *Fuel–Air Cycle Analysis* reveals that the efficiency of an engine cycle depends on several different characteristics of the control system. A more advanced and appropriate analysis of an engine – that which includes real effects of heat transfer, real combustion, friction, and other irreversibilities – gives realistic senses of tangible factors that engineers can strive to improve. Further, exergetic-based analysis offers the important insight of what future opportunities might be exploited in improving the overall system efficiency associated with an internal combustion engine. In returning to the engineer's "dream" engine concept described above, such computed numbers as shown in Fig. 6.13 are more realistic than the ideal *Fuel/Air Cycle Analysis* numbers shown in Fig. 6.10. Further, the exergy-based analysis of Fig. 6.13 could be used to make the necessary assessment of the engineer's dream, to quantify if there are real opportunities of "downstream" exergy conversion when the engine is operated close to maximum theoretical thermal efficiency levels.

Spark Ignition Combustion

As discussed in section on "Thermodynamic Analysis of Internal Combustion Engines", real combustion in an internal combustion engine is not ideal in the sense that it can be modeled as precisely constant volume, constant pressure, or even the combination of the two (limited pressure). Real combustion, instead, involves several complex and interacting features that, on the one hand, make it difficult to predict and model, but on the other hand create opportunities for further

development. Conventional engines are either spark ignited or compression ignited. Since the type of ignition results in substantially different combustion features, the two are separated into respective sections. This section describes spark ignition combustion.

Spark ignition combustion is the typical form of combustion found in the commonly called "gasoline" engine (also commonly called either "petrol" engine or "otto" engine); it is more descriptive to refer to such engines as spark ignition engines. The "spark" aspect of the term implies that combustion is initiated by introducing a high voltage electrical arc, typically through the use of a spark plug, into the reactant mixture at a point during the cycle where sustained combustion reaction will proceed. Most spark ignition engines (exceptions are described in section on "Future Directions of Internal Combustion Engines") induct a premixed and homogeneous mixture of fuel and air during the intake process. At a point near the end of compression, the spark plug discharges an arc into the mixture. This arc, in the near-by region of the spark plug, forms a high temperature plasma that evolves into a flame kernel. The flame kernel transitions from the plug region in a laminar sense establishing a flame front with initial velocity close to the laminar flame speed. This initial flame development is referred to as the flame-development period. Because of turbulent motion among particles composing the mixture, the flame then rapidly spreads throughout the mixture in a turbulent fashion, enveloping microscale eddies with flame. This rapid flame propagation is referred to as the turbulent entrainment period. Upon flame envelopment of the turbulent eddies within the mixture, the flame then laminarly burns the microscale eddies in the final stage of the process referred to as rapid burnup. Turbulent entrainment and rapid burn practically occur simultaneously, and together compose what is referred to as rapid-burning period. This described sequence of spark ignition combustion for a typical operating point (i.e., 1,400 rev/min, part-load condition) is shown in Fig. 6.14 [62], and, for the same typical operating point, requires about 10 ms from spark to finish.

One of the important features of Fig. 6.14, along with the shown in-cylinder pressure, is the corresponding mass fraction burned. Using the first law of thermodynamics, it possible to develop an expression that determines the chemical-to-thermal energy conversion rate (commonly called the "heat release rate") of the combustion process. Since the mass of fuel present in the cylinder, along with the fuel's heating value, are known, the mass fraction burned rate of fuel can be calculated from the heat release rate. Figure 6.15 illustrates a typical mass fraction burned curve, as a percent of the total fuel, and defines the above-described periods referred to as flame-development period (shown in Fig. 6.15 as the flame-development angle, $\Delta\theta_d$) and the rapid-burning period (shown in Fig. 6.15 as the rapid-burning angle, $\Delta\theta_b$). These periods are conventionally defined as the angle swept from spark release to 10% mass fraction burned and the angle swept from 10% mass fraction burned to 90% mass fraction burned, respectively.

The mass fraction burned profile of spark ignition combustion can be modified, and the parameters that cause alterations to the mass fraction burned profile are often coupled. Perhaps the most obvious parameter that can affect the mass fraction
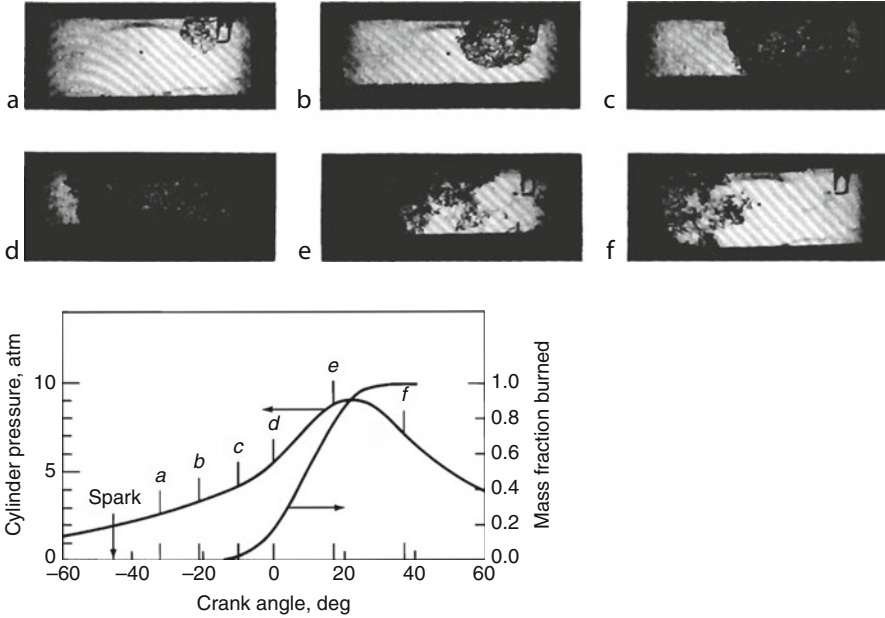
Fig. 6.14 In-cylinder images capture the sequence of spark ignition combustion with the corresponding result on in-cylinder pressure and mass fraction burned. The various steps of spark ignition combustion shown include the spark release (indicated by "Spark"), flame kernel development (**a–c**), turbulent entrainment and burn up (**d–e**), and flame termination (**f**) (Used with permission from [62])
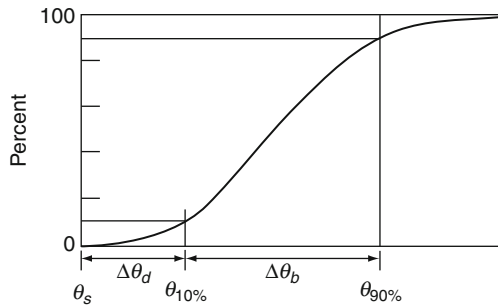

Fig. 6.15 Mass fraction burn rate (as a percent of total fuel) of spark ignition combustion in an internal combustion engine, illustrating the definitions of the flame-development angle, $\Delta\theta_d$, and the rapid-burning angle, $\Delta\theta_b$ (Used with permission from [63])

burned profile is the spark timing, or sometimes called the spark advance. By adjusting the time in the cycle when combustion is initiated (i.e., the spark advance), the times required for flame development and rapid-burning periods are altered. An advance in spark timing (i.e., spark timing is moved earlier into
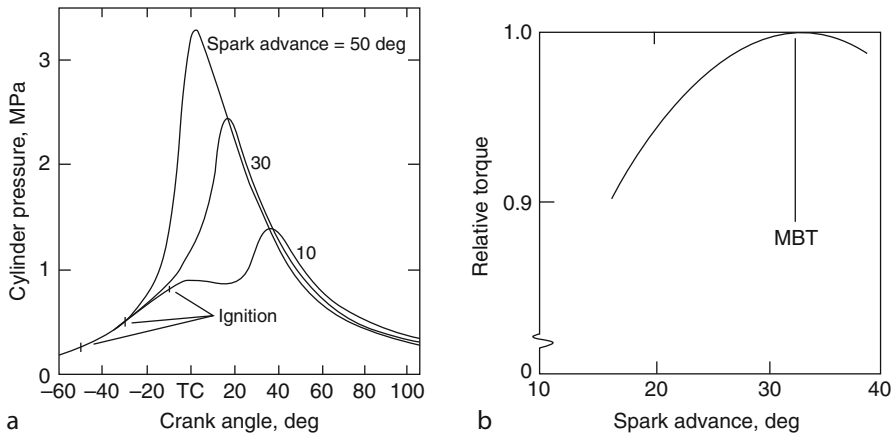
Fig. 6.16 (**a**) In-cylinder pressure as a function of engine crankangle and (**b**) relative torque to maximum brake torque (MBT) as a function of spark advance of a typical spark ignition engine (Used with permission from [64])

compression stroke) generally results in combustion occurring earlier in the cycle and in-cylinder pressures becoming higher in magnitude. Correspondingly, a retard in spark timing (i.e., spark timing is moved later into the compression stroke, or possibly even into the expansion stroke) generally results in combustion occurring later in the cycle and in-cylinder pressures becoming lower in magnitude. There are several other parameters, however, that can affect the burn profile of spark ignition combustion. Examples of such parameters include the mixture's fuel–air ratio (i.e., stoichiometry), the level of EGR, the level of turbulence in the mixture, the level of heat transfer, initial mixture pressure (i.e., load variation), initial mixture temperature, and engine speed. Analysis of each of these parameters is outside the scope of this article; readers are referred to the "Books and Reviews" for additional reference text on the subject.

One item that is important to mention, however, is the effect of burn profile on engine-scale parameters such as performance, efficiency, and emissions. The rate of combustion, and its corresponding effect on in-cylinder pressure, flame temperature, and mixture gas temperature, alters the performance, efficiency, and emissions of the spark ignition engine. For example, Fig. 6.16 illustrates the effect of spark timing (spark advance) on in-cylinder pressure and relative torque to maximum brake torque (MBT). MBT is the maximum torque attained over a given parametric sweep, most commonly over a spark-timing sweep. Note that there is nearly a single spark timing that yields the maximum torque delivered by the engine. Too early of a timing (advanced) or too late of a timing (retarded) renders a lower torque than MBT. Three spark timings are shown in Fig. 6.16: 50°, 30°, and 10° before TDC (BTDC). The most advanced timing (50° BTDC) renders the earliest burn profile (of the studied timings), the highest magnitude of in-cylinder pressure, and the earliest location of peak pressure (occurring around 0° BTDC). Although it may

appear from Fig. 6.16 that the spark advance of 50° BTDC yields the maximum area under the pressure-volume curve (and thus, the maximum work for the cycle), it should be noted that its pressure is lower through most of the expansion stroke than the other two shown spark timings (i.e., 30° and 10° BTDC). This lower pressure through the expansion stroke results from the increased level of heat transfer manifested by the higher gas mixture temperature caused by the faster burn rate-induced higher cylinder pressures. Thus, spark timings advanced of MBT timing result in lower torque due to higher levels of compression work, heat transfer, and friction. Conversely, the most retarded timing (10° BTDC) renders the latest burn profile (of the studied timings), the lowest magnitude of peak pressure, and the latest location of peak pressure (occurring around −40° BTDC). Because of the later-phased combustion, the location of the rise in pressure misses the full opportunity of the expansion stroke. Thus, spark timings retarded of MBT result in lower torque due to expansion losses. It is clear that MBT occurs in the balance of minimized compression work and maximized expansion work, both of which are affected by combustion phasing, heat transfer, and friction. In the example of Fig. 6.16, this balance occurs around 30° BTDC. Notice that peak pressure for this timing is around −20° BTDC; a general "rule of thumb" is that MBT occurs when peak pressure is positioned between −15° and −20° BTDC. As revealed in Fig. 6.16, this rule of thumb extends to the mass fraction burned profile, where maximum brake torque is timed when 50% of the fuel burns by −10° BTDC. In some cases, MBT timing may be "knock limited," meaning that a higher torque could be attained at an earlier timing if fuel knock were not present. Fuel knock is a combustion abnormality of spark ignition engines that, due to the combustion-generated compression of the reactive mixture in the end regions of the cylinder, results from autoignition of the mixture prior to its controlled burn by the propagating flame. Fuel knock can be very damaging because the uncontrolled combustion tends to cause dramatic rises in pressure near critical mechanical components (such as piston rings). The octane rating of a fuel indicates the fuel's resistance to knock in a spark ignition engine; a fuel with a higher octane has a higher resistance to autoignition. Additional discussion about this is provided in the sections. on "A Case Study: Diesel Engines Versus Gasoline Engines" and "Direct Injection, Spark Ignition Engines".

At MBT timing, since fuel flow rate is generally held constant during spark-timing sweeps, efficiency will also be correspondingly maximized. In the case of emissions, however, the trends are not straightforward and a brief discussion of emissions is reserved for the section on "Emissions Formation and Exhaust Pollution".

Finally, it is noted that several parameters affect the spark ignition burn profile. Thus, for each change to a given parameter (e.g., initial pressure, initial temperature, fuel–air equivalence ratio, engine speed, and level of mixture turbulence), a potentially different spark timing will correspond to maximum brake torque. Again, the general trend of spark timing will be such that 50% mass fraction burned occurs at −10° BTDC for maximum brake torque.

Compression Ignition Combustion

In contrast to spark ignition combustion – which uses an electrical arc, or spark, to initiate combustion of a reactive mixture – compression ignition combustion relies on compressive heating – or, the increase in temperature of a gas due to the increase in pressure resulting from the decrease in volume – where combustion initiation is kinetically driven by exceeding the ignition temperature of the fuel–air mixture. Compression ignition combustion is the typical mode of combustion used in the commonly called diesel engine, an engine which more descriptively is called a compression ignition engine.

It is immediately recognized that one challenge of compression ignition engines is the lack of a direct trigger of ignition; the spark acts as the direct trigger of combustion initiation in a spark ignition engine. Conventional applications of compression ignition engines (e.g., the conventional diesel engine) overcome this challenge by using the fuel injection event as the direct trigger. Thus, conventional compression ignition engines typically induct an air and residual mixture (i.e., no premixing of fuel) during the intake stroke. This same unreactive mixture is compressed during the compression stroke until near the point when combustion is desired to begin. At this point fuel is introduced into the mixture, which, due to compression, is a high temperature, high pressure environment. After several complex and coupled processes occur (described below), combustion initiates and chemical to thermal energy conversion takes place.

At this point, it is instructive to briefly describe the role of "glow plugs" often used in compression ignition engines. Glow plugs should not be confused with spark plugs. A glow plug is a resistive heating element inserted into the cylinder of a compression ignition engine to aid in the initial starting of the engine. It acts as a warming device during "cold start"; after engine warm up and stable operation, the glow plugs deactivate. They are not necessary on a cyclic-basis. Spark plugs, on the other hand, are integral components of spark ignition engines; they provide the source of ignition for every combustion cycle of a spark ignition engine.

The method by which fuel is injected into the compressed mixture typically falls into one of two categories: (1) direct injection, or (2) indirect injection. Indirect injection involves injecting fuel into a "prechamber" which is connected to the main chamber. Fuel injected into the prechamber ignites and causes the mixture to issue into the main chamber where main heat release and work extraction occurs. Indirect injection engines are used primarily in applications where motion of the nonreactive mixture in the main chamber is too quiescent for ignition; this typically occurred in the early implementations of small, high speed automotive compression ignition engines. Use of a prechamber, where nonreactive mixture forced through orifices or nozzles connecting the main chamber to the prechamber suitably increased mixture swirl and turbulence, generated sufficient fluid motion to allow ignition to occur. Technological developments in intake port design, combustion chamber design, and fuel injection systems have rendered indirect injection systems nearly obsolete and technically inferior to direct injection techniques.
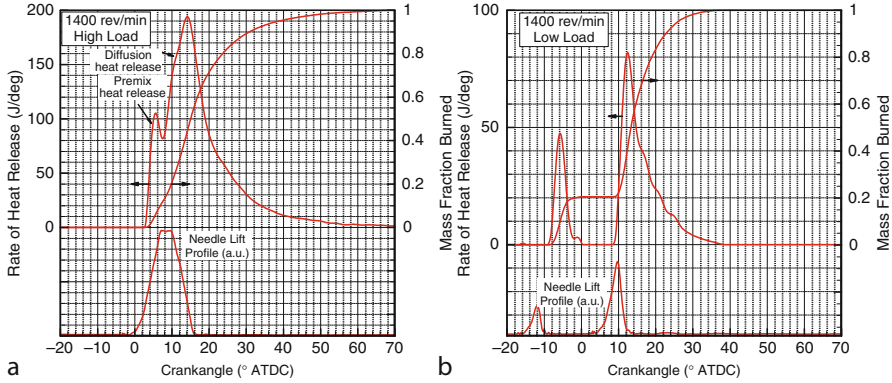
Fig. 6.17 Fuel injector needle lift profile, rate of heat release, and mass fraction burned for a typical diesel engine operating at 1,400 rev/min (**a**) high-load condition (ca. 75% peak load = 11.3 bar BMEP) and (**b**) low-load condition (ca. 25% peak load = 1.9 bar BMEP) with the use of pilot injection (Data from author's laboratory, Texas A&M University)

Direct injection, as the name implies, indicates that the fuel is injected directly into the cylinder at the point near when combustion is desired to begin. Because of the compressed nature of the nonreactive mixture into which fuel is injected, fuel injection pressures must be very high; i.e., modern fuel injection systems inject fuel at pressures on the order of 100–2,000 bar (ca. 3,000–30,000 psi). The processes of fuel injection, combustion initiation, and subsequent combustion propagation (or burning) are very complex in compression ignition engines. The following discussion will briefly highlight these complexities.

Figure 6.17 illustrates the sequence of events that occur in a typical direct injection compression ignition engine. Two load conditions at 1,400 rev/min are shown, with a high-load condition (ca. 75% peak load which is about 11.3 bar BMEP) shown as Fig. 6.17a and a low-load condition (ca. 25% peak load which is about 1.9 bar BMEP) shown as Fig. 6.17b. Both plots illustrate the fuel injector needle lift profile (which roughly correlates to the fuel delivery rate), the rate of heat release profile, and the mass fraction burned profile. Focusing first on the high-load condition (Fig. 6.17a), notice that fuel injection occurs near 2° BTDC. Fuel in typical compression ignition engines is injected as a liquid. Thus, the fuel must undergo a series of physical processes (e.g., penetration, breakup, atomization, and vaporization) before it undergoes its chemical process of bond fragmentation and eventual ignition. This period of time – i.e., the time between start of fuel injection and ignition (commonly called start of combustion) – is often called the ignition delay period. As will be described below, the ignition delay period of a compression ignition engine is an important parameter that affects the remainder of the burn profile and is affected by several other parameters. The end of the ignition delay period after start of injection will witness start of combustion and noticeable heat release, as indicated in Fig. 6.17a as the positive rate of heat release. During heat release, the mass fraction burned profile steadily increases until end of combustion when all the fuel is most nearly completely burned.

An interesting and important feature of Fig. 6.17 is the rate of heat release profile between start of combustion and end of combustion. Notice in Fig. 6.17a there are two distinct components of the rate of heat release: a premixed component and a diffusion component. These two respective components are nearly an exclusive feature of compression ignition combustion that relies on fuel injection as the ignition trigger. The premixed heat release component results from the complex processes involved in physically preparing the initial "packets" of fuel for chemical decomposition [65]; as described above, the injected liquid fuel must be prepared and sufficiently mixed with the cylinder gases before chemical decomposition can begin. Once ignition occurs, the parcels of fuel that prepared for burning during the ignition delay period react under kinetic control, with relatively high rates of burning for the given mixture temperature. Fuel continues to inject during the premixed portion, and the increase in temperature due to premixed combustion accelerates the physical preparation of the newly introduced parcels of fuel. In spite of this acceleration of fuel preparation, the burn rate becomes limited by the fuel's mixing rate with the air. Since the time scales for mixing are larger than the time scales for chemical kinetic decomposition [66], the burn rate becomes mixing-controlled, rather than kinetically controlled. The mixing-controlled, or diffusion-controlled, component of heat release is observed in Fig. 6.17a as the longer heat release following premixed heat release and is identified as "diffusion heat release." The burn rate of diffusion heat release is generally slower than premixed heat release at any given reaction temperature. Figure 6.17a shows diffusion heat release having a much higher burn rate than premixed heat release; this is manifested, however, by the associated higher mixture temperature effected by the combustion process.

The situation is slightly different for the low-load case, shown as Fig. 6.17b. Conventional compression ignition engines that use fuel injection as the ignition trigger typically also use the amount of fuel delivery as the means to control engine load. Thus, the need to produce less power at a given engine speed is effected by decreasing the amount of fuel delivered during the injection process. This is evident in Fig. 6.17b where the area under the fuel injector needle lift profile is less than that of the high-load case (for the moment, disregard the bimodal fuel injection profile, which will be explained below). Consequently, the distribution of premix versus diffusion heat release is not apparent on the low-load heat release profile. This does not imply that mixing-controlled combustion does not occur during low-load operation; it does imply, however, that combustion is predominantly kinetically controlled at low-load conditions whereas it is predominantly diffusion-controlled at high-load conditions.

The bimodal feature of the fuel injection profile in Fig. 6.17b is the result of the use of a double-injection strategy on this particular engine at this particular condition. Modern advanced engine fuel systems are capable of introducing fuel in sequences, or pulses, in what is commonly called "multi-injection strategies." In the case of the engine highlighted in Fig. 6.17b, the first injection is considered a "pilot" injection, while the second injection is considered a "main" injection. The pilot injection introduces only a small portion (e.g., ca. 10%) of the total fuel to be
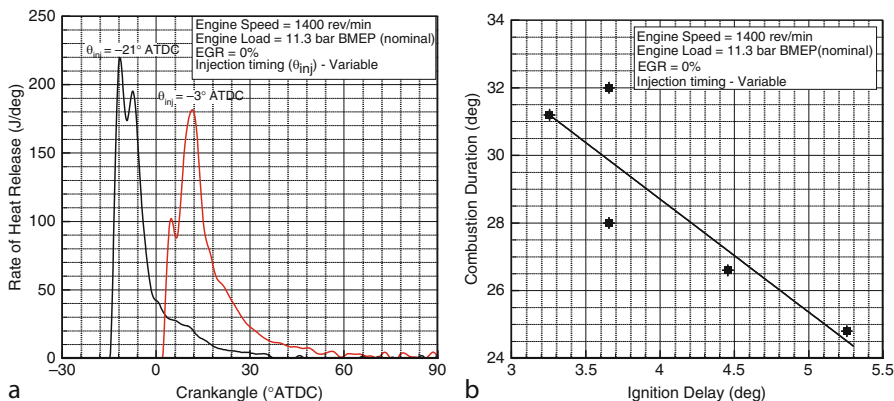
Fig. 6.18 (**a**) Rate of heat release at an advanced injection timing (21° BTDC) and a retarded injection timing (3° BTDC) and (**b**) the relationship between ignition delay and combustion duration via an injection timing sweep for a typical compression ignition engine operating at 1,400 rev/min, high-load condition (11.3 bar BMEP, nominal) (Data from author's laboratory, Texas A&M University)

delivered in the cycle, creating an opportunity to prepare a smaller portion of fuel for burning during the ignition delay period. This small pilot increases cylinder temperature, so that the ignition delay period of the main injection is much shorter, and there correspondingly is a lower fraction of premixed heat release during the main combustion event. Such a strategy is used, for example, to reduce combustion-generated noise of a diesel engine. Interestingly, diesel engine noise is an "age-old" problem, serving as a primary focus of development for Sir Harry Ricardo (a pioneer in the development of high speed diesel engines) [67].

The above-described use of multiple injections hints at an ever-present relationship within compression ignition combustion systems using fuel injection as the direct trigger; i.e., a relationship among ignition delay, fraction of premix heat release, and fraction of diffusion heat release exists. This relationship is shown in Fig. 6.18a, which illustrates the rates of heat release at two different injection timings of a typical compression ignition engine operating at 1,400 rev/min, high-load condition (11.3 bar BMEP, nominal). Notice for the advanced injection timing there is a substantial level of premix heat release and a correspondingly lower fraction of diffusion heat release relative to the retarded injection timing. The advanced injection timing, due to the injection of fuel into a relatively cool environment when rates of physical and chemical processes are low, results in a relatively longer ignition delay. Because of the longer ignition delay, there is more time for fuel preparation prior to ignition; once ignition occurs, there is a relatively higher concentration of fuel that is prepared for burning and correspondingly combusts in a premix and kinetically controlled fashion. As more fuel is prepared during the ignition delay period for premix burn, there consequently is less fuel available for diffusion heat release; thus, as premix burn fraction increases, diffusion burn fraction decreases. Likewise, at the retarded injection timing, fuel is

injected in a relatively hotter environment which enables fuel preparation for burning at a faster rate and combustion commences relatively sooner (i.e., shorter ignition delay). Because of the shorter ignition delay, there is less fuel prepared for burning, thus there is lower fraction of premix heat release and correspondingly larger fraction of diffusion heat release.

Because premix heat release rate is relatively faster (at a given temperature) than diffusion heat release, the "combustion duration" – or, the time taken between start of combustion and end of combustion – for a mostly premix heat release combustion event will be shorter than that of a mostly diffusion heat release combustion event. Thus, there tends to be an inverse relationship between ignition delay and combustion duration, as shown in Fig. 6.18b, where changes to ignition delay are manifested by alterations to injection timing. The shorter ignition delay results in less premix heat release, more diffusion heat release, and correspondingly a longer combustion duration.

Injection timing is just one parameter that can affect ignition delay, and thus the burn profile of a compression ignition combustion event when fuel injection is used as the ignition trigger. Other engine parameters such as EGR level [68], initial temperature and initial pressure [69], injection pressure [70–72], and swirl and turbulence [69, 73] all have certain effects on ignition delay and the resulting relative fractions of premix and diffusion heat release. Like the effect of various parameters of spark ignition engine performance, efficiency, and emissions, the various effects of compression ignition engine parameters on ignition delay and burn profile also have an effect on engine efficiency, performance, and emissions. For example, overly advanced or retarded injection timings will cause decreases in engine torque for a given fuel delivery rate. Additionally, fuel quality is particularly important for conventional diesel combustion operation, where low volatility fuels with high ignitability are generally used to ensure ignition occurs during the cycle. An important fuel parameter – i.e., its cetane number – is used to identify the quality of fuels appropriate for use in a diesel combustion system. A higher cetane corresponds to a fuel with a shorter ignition delay.

Finally, in closing, it is important to recognize the above-described phenomena are largely phenomenological observations that can be made using conventional diagnostics with relatively straightforward analysis. Considerable development (e.g., [49, 74, 75]) has been made to provide substantial insight into the complex fluid, heat transfer, and chemical processes that occur during compression ignition combustion where fuel injection is used as the direct ignition trigger. Description of these details is outside the scope of this work.

Emissions Formation and Exhaust Pollution

A discussion on the basics of internal combustion engines, actually any combustion-based device, is incomplete without a description of the associated harmful species that may exist in the products of combustion. Such harmful species are

generally called "exhaust pollution" and many governing agencies around the world place restrictions on the emission of certain pollutants from combustion-based devices. Because the application of the internal combustion engine is so varied, emission regulations tend to be application-oriented. For example, in the USA, emission regulations are placed on passenger vehicles differently than emission regulations placed on heavy trucks or hand-held engine devices (such as lawn mowers). Because of such variability in regulation, the time-oriented nature of the regulations (i.e., regulations have, to this point, been in a state of flux), and variation in regulation among various governing agencies around the world, further description of specific emissions will not be provided. The basic issues at hand, however, can be briefly described.

As shown in Reaction (6.R1), there are several species formed during the combustion reaction of a typical hydrocarbon–air mixture. Some of these species are generally stable and nonreactive in the atmosphere, thus pose little to no harm to the five kingdoms of nature. Other species, however, are either harmful or reactive in ways that lead to harmful consequences. Specifically, there is little scientific debate about the harmful nature of certain combustion products such as CO, NO/ NO_2, unburned hydrocarbons (HC), and particulate matter (PM, or the solid/liquid components of exhaust that can be collected on a filter. It is noted that historically sulfur oxides and sulfates have been considered either separately from [76] or in combination with [77, 78] particulate matter). There is, perhaps, continued debate about the potential consequences of other products of combustion; in particular, there is current debate on the consequence of CO_2 and the role it may play in the presently observed warming of the planet (i.e., so-called global warming or global climate change). Because of the certainty of the effects of the former species, attention will be given to them and basic information on their formation during combustion. Because of the uncertainty of the effects of the latter species, readers are referred to other literature to uncover the current state-of-debate of CO_2 and its potential impact on global trends currently believed to occur (see, e.g. [79, 80]).

The first such species to describe is CO, which due to its fatal effects on human/animal life is one of the first combustion products to be considered a pollutant [76]. It is well established [76] that all precursor hydrocarbon decomposition reactions firstly form CO. Thus, increased concentrations of CO in engine exhaust result from incomplete reaction of the principal CO oxidation step [26], given as Reaction (6.R2):

$$CO + OH \leftrightarrow CO_2 + H \qquad (6.R2)$$

In internal combustion engines, the incomplete oxidation of CO most typically occurs during rich engine operation [76], where available oxidants for final CO oxidation are lacking [81]. Fuel–air mixtures close to stoichiometric or even slightly lean, however, result in nonnegligible concentrations of CO.

The next species to consider is NO. NO emerged as combustion-generated pollutant due to its observed effect of reacting with hydrocarbons in the presence of sunlight to produce tropospheric ozone [82]. Its formation in a combustion system is rather complex, as there are several major pathways through which

it can form. These major pathways include thermal (or commonly called Zeldovich), prompt, and fuel-based nitrogen [26]. For reciprocating-type internal combustion engines, the primary NO formation mechanism is the thermal mechanism where atmospheric air serves as the principal source of nitrogen in the mechanism [27]. There are three reactions that compose the mechanism, given as Reactions (6.R3)–(6.R5):

$$O + N_2 \leftrightarrow NO + N \qquad (6.R3)$$

$$N + O_2 \leftrightarrow NO + O \qquad (6.R4)$$

$$N + OH \leftrightarrow NO + H \qquad (6.R5)$$

The forward and reverse reaction rate constants of Reactions (6.R3) and (6.R4) are generally exponentially dependent on temperature. To demonstrate the substantial role temperature plays on NO formation, several simplifying assumptions (see below) are applied to Reactions (6.R3) and (6.R4) to yield Eq. 6.15 [26]:

$$\frac{d[NO]}{dt} = \frac{6 \times 10^{16}}{T^{\frac{1}{2}}} \exp\left(\frac{-69090}{T}\right) \left[O_{2,eq}\right]^{\frac{1}{2}} \left[N_{2,eq}\right] \cdot \left(mol/cm^3\text{-}s\right) \qquad (6.15)$$

where [NO] is the concentration (mol/cm^3) of NO at time, t (s), and $[O_{2,eq}]$ and $[N_{2,eq}]$ represent the equilibrium concentrations (mol/cm^3) at temperature T (K) of oxygen and nitrogen, respectively. The several simplifying assumptions that go into Eq. 6.15 include the following. The first assumption is that the nitrogen chemistry is de-coupled from the combustion reactions. Although combustion reactions generally occur much faster than nitrogen chemistry [25], the presence of O and OH radicals in the thermal mechanism (which are also important species in combustion reactions) may require the chemistries to be coupled for accurate NO prediction [27]. By assuming the chemistries are de-coupled, O, O_2, OH, H, and N_2 can be approximated by their equilibrium concentrations at equilibrium temperature; assuming equilibrium temperature is the second assumption applied to Eq. 6.15. The third and last assumption applied to Eq. 6.15 is that nitrogen radical (N) is in steady-state concentration (i.e., $\frac{d[N]}{dt} = 0$). Finally, it is reinforced that the forward reaction rate constant of Reaction (6.R3) is used from [26] in Eq. 6.15; updated reaction rates are available in Dean and Bozzelli [29]. It is clear from Eq. 6.15 the strong dependency NO formation rate has on temperature.

Because of the dominance of the thermal mechanism on NO formation in internal combustion engines, combustion-based efforts to reduce NO center on reducing the reaction temperature and O_2 concentration. Such techniques include altering spark advance [83, 84] or injection timing [85–87] for spark ignition or compression ignition engines, respectively, and introducing EGR into the mixture [87–90].

As described above with NO, unburned hydrocarbons (HC) play a role in the formation of tropospheric ozone. Further, they represent unreacted fuel; if left to

discharge into the atmosphere, there is lost opportunity to convert that chemical energy into useful work. Formation of HC species during combustion reaction are complex and varied depending on the type of fuel used [91]. A general cause for HC emissions, however, is insufficient mixing between fuel and air [76], where most of the HC emission species are formed during low temperature ($T < 1,000$ K) reactions [26]. Typically, in conventional reciprocating internal combustion engines, most of these species are oxidized as combustion enters high temperature mechanisms [26]. Reciprocating engines, however, contain "sources" for hydrocarbon storage that, if not oxidized upon release in the gas mixture in the later portion of the cycle, emit as HC emissions in the exhaust [92]. Although there are special considerations given to engines operating under cold-start conditions [93], the general storage locations of HC species (and thus, the major source of HC emissions [94]) include cylinder head gasket crevice, spark plug crevices, piston ring pack crevices, and valve seat crevices. In addition to crevice HC storage, other sources of HC emissions [94] include single-wall flame quenching, oil film layers, combustion-chamber deposits, exhaust valve leakage, and liquid fuel (i.e., HC species not vaporized during the process). Because of the relative importance of crevice storage on HC emissions, much of the combustion-based effort to decrease HC emissions has centered on reducing the volume and flow pattern of crevices in the piston/cylinder arrangement.

The final major pollutant to briefly describe is particulate matter, or PM. PM is essentially any exhaust specie that can be collected on a filter; it typically is structured on a solid organic component (which is mostly pyrolized carbon particles, or "soot") upon which organic (e.g., unburned HC) and nonorganic (sulfates) components build. Since soot serves as the building block for PM, much of the research efforts are dedicated to understanding soot formation processes [95–99]. The general path for soot formation begins with the high temperature pyrolysis, or fragmentation to hydrocarbon radical, of the hydrocarbon fuel. Such pyrolysis typically leads to a certain group of hydrocarbon radicals called polyaromatic hydrocarbons. Such polyaromatic hydrocarbons that do not oxidize during the reaction serve as the nucleation site for soot growth. Nuclei then begin to coalesce and substantially increase surface area. Now particles, the high surface area soot particles agglomerate and allow other species (i.e., liquid HC, condensed gaseous HC, and sulfates) to absorb onto the surfaces [99]. Soot formation is strongly dependent on mixture fuel–air ratio and also temperature (both tend to govern the rate of pyrolysis). Formed soot can, however, undergo oxidation as well. Soot oxidation is likewise a function of temperature, but has a stronger dependency on temperature than soot formation [99]. The difference between soot that is formed and soot that oxidizes is ultimately released in the engine exhaust, and is commonly called "net soot release." Compression ignition engines (e.g., diesel engines) are often plagued with high PM emissions, largely due to the heterogeneity induced into the fuel/air mixture by use of fuel injection as a direct trigger for ignition. Spark ignition engines, however, are recently being considered as sources of nanoscale soot particles [100, 101].

It is clear from the above discussion that soot and PM emission processes are complex. From phenomenological considerations, however, Khan et al. [86, 102] and Ahmad [103] provide insight into the general behavior of PM with, in particular, diesel engine combustion. Specifically, it is observed that increases in diffusion heat release generally increase the emissions of PM. A portion of this may be due to the overall lower reaction temperatures, manifested by relatively lower premix heat release, decreasing the rate of soot oxidation through the diffusion flame. This observation is also supported, however, by Dec's [74] observation that precursor soot formation occurs in the standing premix reaction zone within a diffusion flame sheet that exists in diesel combustion. Thus, increased diffusion burn correspondingly results in increased soot emission. Either way, it becomes apparent when relating this discussion to the known effects of injection timing, for example, on diffusion combustion and NO emissions that an attempt to decrease soot by creating higher temperatures with, perhaps, more premix heat release will correspondingly increase NO emissions; hence, the establishment of the conventional "soot-NO$_x$" tradeoff of diesel engines.

As described briefly above with each species, there are in-cylinder and combustion-based methods to reduce the formation of various exhaust pollutants. Advanced development of combustion systems continues to focus on these in-cylinder methods, as described in section on "Future Directions of Internal Combustion Engines". Also in use to eliminate exhaust pollution are exhaust after treatment systems, which conventionally are catalyzed devices (hence, their common name of "catalyst" or "catalytic converter") [104–106]. The basic idea of a catalyst is to promote a reaction that otherwise would not proceed. In aftertreatment of engine exhaust, there are several catalysts in use depending on the species composition of the exhaust. For example, the conventionally named "three-way catalyst" is often used with conventional gasoline spark ignition engines. Such a catalyst is typically an exhaust flow-through device composed of a ceramic monolith (the substrate) with thru-hole passages to allow exhaust gases to flow and a metal-oxide "washcoat" that suspends catalytic particles on the surfaces of the monolithic passages. The monoliths are usually constructed of coerderite and the metal-oxide washcoat is often an alumina washcoat. Catalyzed particles are often from the precious metals group (e.g., platinum, palladium, and rhodium), with platinum being the most commonly used metal. The general operation of the three-way catalyst is to promote the reduction–oxidation reactions among CO and HC (the reductants) and NO/NO_2 (the oxidants). In other words, the catalyst promotes the reduction of NO or NO_2 to yield stabilized N_2 and the oxidation of CO and HC to yield stabilized CO_2 and H_2O. The efficiency of the three-way catalyst – or, the conversion effectiveness of converting a given species to its more stable species, e.g., CO to CO_2 – is strongly dependent on the constituent composition on the inlet mixtures, as shown in Fig. 6.19 [104]. Notice that maximum conversion efficiencies of NO$_x$, CO, and HC occur very near stoichiometric air–fuel ratios; a small departure from stoichiometric conditions – e.g., about 0.5% increase or decrease in A/F ratio – results in a nearly 20% decrease in conversion efficiency. Because of such an

Fig. 6.19 Catalyst efficiency of a typical three-way catalyst interacting with exhaust from a typical gasoline spark ignition engine as a function of mixture air–fuel ratio (A/F ratio) (Used with permission from [104])
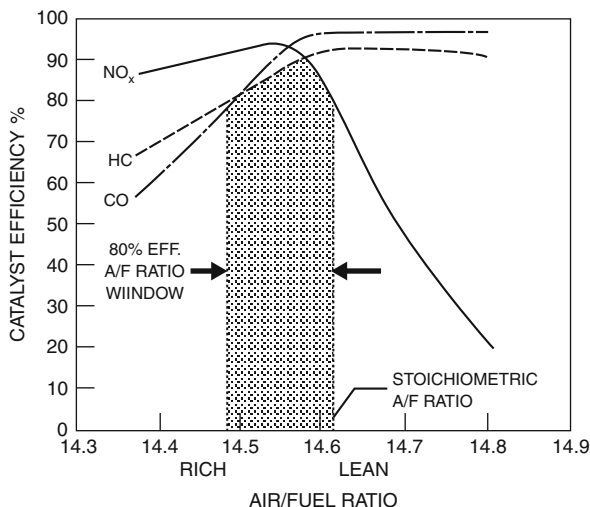
intolerable response of the catalyst, effective control of gasoline spark ignition engines requires precise control of the mixture stoichiometry during operation.

Also clear from Fig. 6.19 is the challenge of outfitting a typical platinum-based catalyst with non-gasoline spark ignition engine technology. Diesel engines which typically operate fuel lean (i.e., oxygen rich), for example, create constituent exhaust species that are difficult to catalyze using conventional techniques. Advanced aftertreatment technologies for diesel engines, or other advanced combustion/engine systems, are under development and beginning to appear in production applications [105, 106].

A Case Study: Diesel Engines Versus Gasoline Engines

At this point, with the basics of internal combustion engines having been described, it is useful to do a "state-of-the-art" technology comparison between the two dominant conventional internal combustion engines – i.e., the gasoline spark ignition engine and the diesel compression ignition engine – to set the stage for discussions on future directions in internal combustion engines. The elementary comparison is provided in Table 6.2; note that this comparison is neither comprehensive nor general. It is intended to highlight the common state-of-the-art of the two technologies, and create a sense for the trends of future directions in internal combustion engine development.

It is clear from Table 6.2 that the two engine types differ in virtually every way, save for their common use of the kinematic elements of the crank-slider and piston/cylinder components. Consider first the spark ignition engine; its use of spark requires a relatively tight control on the fuel–air mixture equivalence ratio. The

Table 6.2 Elementary comparison between conventional spark ignition engines and conventional compression ignition engines. The comparison is not comprehensive nor general, rather it is intended to highlight the common state-of-the-art and create a sense for the trends of future directions in internal combustion engine development

Feature	Conventional spark ignition engine (homogeneous charge spark ignition)	Conventional compression ignition engine (heterogeneous charge compression ignition)
Also known as	Gasoline, petrol, Otto engine	Diesel engine
Method of ignition	Spark	Compression
Fuel equivalence ratio	Precisely controlled to stoichiometric	Varies depending on engine load; typically remains lean
Fuel/air mixture preparation	Carburetion, throttle-body injection, port injection	Direct injection or indirect injection
Degree of fuel/air mixing	Homogeneous	Heterogeneous
Fuels used	Gasoline, alcohols, ethanol, hydrogen, high volatility hydrocarbons	Diesel, oils (transesterified), hydrogen, low volatility hydrocarbons
Compression ratios	Ca. 8–11	Ca. 14–21
Method of load control	Throttle restriction of air or fuel/air mixture (depending on mixture preparation)	Fuel injection duration (i.e., fuel quantity)

flame travel time is minimized when fuel–air equivalence ratio is near stoichiometric, and greater than about 20–40% departure from stoichiometric will typically cause misfire [107]. It is noted that this requirement [107] is only for ignition and flame propagation. The integration of a three-way catalyst with a conventional spark ignition engine requires even more precise control over fuel equivalence ratio, as described in section on "Emissions Formation and Exhaust Pollution" [104]. Thus, conventional spark ignition engines are not able to tolerate substantial departures from stoichiometric conditions. Further, because of this requirement, it is necessary to avoid mixture striations between fuel and air. Thus, conventional spark ignition engines make use of premixing devices (e.g., carburetors or throttle body and port fuel injectors) to prepare the fuel–air mixture prior to induction. The result is a homogeneous mixture of fuel, air, and other residuals such as residual fraction and purposefully introduced constituents such as EGR. In order to assist with the homogenization of the fuel–air mixture, light distillates and aromatics with high volatility are typically used as the fuel for conventional spark ignition engines. The most common, of course, is the mixture of light distillates and aromatics commonly called "gasoline" or "petrol." Interestingly, largely due to the reactive nature of the homogeneous and near-stoichiometric mixture [108], compression ratios of spark ignition engines are limited to around 11. Compression ratios higher than this promote the undesirable autoignition of certain regions of the fuel–air mixture during the combustion process; a phenomenon known as "fuel knock." Fuel knock can be very damaging to an engine, mostly due to its occurrence in the end regions of cylinder, where uncontrolled autoignition of the mixture results in excessively high rises in pressure near susceptible components of the

piston/cylinder arrangement (e.g., piston rings). As an aside, initially tetra-ethyl lead and eventually other lead alkyls were used in gasoline as anti-knock agents, allowing for compression ratios to increase in spark ignition engines [109]. Concerns over the effect of lead on human and animal life, as well as the implications of lead on catalytic devices, have mostly eliminated lead-based additives from fuels [110]. Finally, and similarly related to the spark's need to ignite a near-stoichiometric and homogeneous mixture, conventional spark ignition engines control their load, or power output, via the use of a throttle. In other words, the power of the engine at any given speed is minimized by decreasing the efficiency by which the engine can induct fresh mixture.

Consider now the conventional compression ignition engine, or commonly called diesel engine. It seems that from its first inception, the diesel engine was to be a compressively ignited machine, as Diesel intended to create an isothermal reaction where air and fuel were (separately) compressed to the combustion temperature and expansion occurred isothermally as chemical energy converted directly into mechanical energy [61]. Of course, Diesel never succeeded in attaining this isothermal process (nor has anyone since); but, the foundation had been laid for a compression ignition engine. By using compression ignition combustion, many of the constraints placed on the spark ignition engine (i.e., near-stoichiometric fuel equivalence ratio, homogeneous mixture, and throttle of intake mixture) no longer need apply to the diesel engine. It also seems that from its initial inception, Diesel intended the fuel to be directly delivered to the compressed air [61]; of course, it is also clear that direct (or indirect) injection of the fuel to compressed air is necessary so as to control start of combustion. Further, to aid in the gradual and isothermal conversion from chemical to mechanical energy, Diesel stipulated that there be a chemical abundance of air (i.e., fuel lean) in the mixture [61]; practical implementations of diesel engines make use of varying fuel-equivalence ratios to control the load of the engine. The direct (or indirect) injection of fuel into the compressed air, contrary to Diesel's conception, establishes a heterogeneous mixture that within itself has widely varying fuel–air ratios and reaction temperatures. Further, since ignition is manifested through compression, fuels with high ignitability (e.g., heavy distillates and aromatics) are required (correspondingly, the fuels also have relatively lower volatility, further establishing the heterogeneous nature of diesel combustion). Also related to the use of compression to ignite the mixture is the need to have high compression ratios (i.e., typically varying between 14 and 21). Finally, as already described, engine load is controlled by the direct and exclusive control of fuel; thus, engine load varies as mixture fuel–air equivalence ratio varies. Due to the heterogeneous nature of diesel combustion, and the dependency of soot formation on mixture stoichiometry under conventional conditions, fuel equivalence ratios rarely exceed 90% of stoichiometric to avoid "smoke limitation."

In the context of the three common attributes of internal combustion engines (i.e., performance, efficiency, and emissions), and the comparison given in Table 6.2, brief comments will be made about the pros and cons of each conventional technology.

Table 6.3 Comparison of the various parameters that control an engine's ability to make power (Eq. 6.7) between typical applications of conventional spark ignition and conventional compression ignition engines at wide-open throttle or full load condition. It should be noted that these are qualitative assessments of typical technology, and should not be viewed as absolute truths of the respective technologies. Since typical applications of internal combustion engines operate on the four-stroke principle, its effect on the comparison is neutral

Parameter	Parameter's effect on power[a]	Conventional spark ignition engine	Conventional compression ignition engine
η_f	↑		↑
η_v	↑		↑
$\rho_{a,i}$	↑		↑
(F/A)	↑	↑	
Q_{HV}	↑		↑
V_d	↑		↑
N	↑	↑	
n_R	↓	↔	

[a]In other words, an increase in the parameter will have the listed effect on power

In terms of power, consider Eq. 6.7 and the qualitative relation of each parameter for a given technology at their "wide-open throttle" or full-load operation, as given in Table 6.3. Described in more detail below, diesel engines tend to have higher fuel conversion efficiencies. Because of not premixing the fuel with air and generally lower engine speeds (which allows the avoidance of flow choke), diesel engines tend to have higher volumetric efficiencies (refer to the discussion surrounding Fig. 6.6 and [111] for more detail on factors affecting volumetric efficiency of engines). Typical applications of diesel engines use turbocharging, which increase the inlet mixture density to above atmospheric conditions (typical spark ignition engines operate naturally aspirated). As described above, spark ignition engines generally operate stoichiometric, whereas compression ignition engines generally operate lean even at full power. The heating value of diesel fuel is marginally higher than that of gasoline. Displaced volumes of typical compression ignition engines tend to be larger than those of typical spark ignition engines. Correspondingly, peak power speeds for typical spark ignition engines tend to be higher than compression ignition engines. Mostly due to the larger displaced volumes, but also assisted by higher fuel conversion and volumetric efficiencies, higher inlet density, and higher heating value of the fuel, typical compression ignition engines tend to exhibit higher peak powers than typical spark ignition engines. If the power is normalized by displaced volume to render the specific power, however, spark ignition engines tend to have higher specific power than compression ignition engines (in spite of compression ignition engines having high efficiencies, density, and heating value, the stoichiometric and high speed operation of the spark ignition engine tends to yield higher specific power).

As described in section on "Thermodynamic Analysis of Internal Combustion Engines", engines operating with high compression ratios and lean fuel equivalence ratios will tend to have higher efficiencies than engines operating with relatively lower compression ratios and near-stoichiometric equivalence ratios. As a result,

typical compression ignition engines tend to have higher efficiencies than typical spark ignition engines. In some instances, turbocharging that is typically found on compression ignition engines could increase efficiency if intake manifold pressure is boosted to higher than exhaust manifold pressure; the primary function of a turbocharger, however, is to increase inlet mixture density to increase the power capabilities of the engine (as shown in Eq. 6.7). The efficiency improvement of compression ignition engines at part-load conditions becomes amplified as (a) the mixture becomes leaner for the compression ignition engine and (b) the use of throttle to manifest part load in the spark ignition engine introduces a thermodynamic loss parameter in the cycle.

Finally, a brief comparison of emissions between the two engines at wide-open throttle or full load condition is made. Because of their near-stoichiometric operation, engine-out emissions of HC and CO for spark ignition engines tend to be relatively higher for those of compression ignition engines (where the latter uses fuel-lean mixtures, creating an oxygen rich exhaust and high level of oxidation of partially oxidized species such as CO and HC). Engine-out emissions of NO tend to be nearly the same between engine technologies. Engine-out emissions of PM are much higher for compression ignition engines, where mixture heterogeneity creates numerous opportunities for soot formation. Of course, it is important to note that conventional spark ignition engines are typically coupled with an effective catalyst, substantially lowering the catalyst-out emissions of the various species to levels well below the engine-out emissions of compression ignition engines. While the use of aftertreatment systems with conventional compression ignition engines are less straightforward, technology is becoming available to also allow substantial reduction of their engine-out emissions [106].

At this point, it becomes clear both engine technologies have features which are more favorable than the other for power, efficiency, and emissions considerations. For example, the high compression ratio and lean mixture required by compression ignition are attractive from efficiency perspectives. The homogeneous mixture of spark ignition is attractive from uniform combustion and emissions perspectives. The lack of a throttle to control load of a compression ignition engine is attractive, but the direct ignition trigger of a spark ignition engine is also attractive. Thus, it is clear that future engine developments could exploit the favorable features of engine technologies to create the next generation internal combustion engine. The next section will describe such efforts and offer insight into the likely future direction of internal combustion engines.

Future Directions of Internal Combustion Engines

There are several types of advanced technology that exist for internal combustion engines. Some of this technology is very prevalent on engines (e.g., turbocharging on diesel engines). Some technology is beginning to appear on production models (e.g., variable valve timing). Other technology is still in its development stages,

awaiting its potential entry into full-scale production (e.g., homogeneous charge compression ignition combustion). This section will briefly describe such technology.

Engine Downsizing

A general idea that permeates much of the technology under development of internal combustion engines is that of engine downsizing – or, the effort to use advanced technology to enable the use of smaller-sized engines to produce the same power (i.e., increase power density) [112]. The application-oriented benefit is that a smaller engine likely weighs less, thus could improve application efficiency (e.g., better vehicle fuel economy with a lighter engine). The engine itself, however, will likely realize improved efficiency. For example, a smaller engine that uses turbocharging to maintain same power (as a larger-sized engine) will generally operate more often near the location of peak efficiency. Some benefit of such is realized in diesel engines, which typically have best efficiencies near mid-speed, and 75% peak load conditions. More benefit, however, is realized in gasoline engines where throttles are used; a smaller engine with higher power density will require less throttle and thus realize larger gains in efficiency improvement. Overall, the major purpose of engine downsizing is to improve parameters other than V_d in Eq. 6.7. This idea will become more apparent as specific technology is discussed below.

Turbocharging/Supercharging/Boosting

Boosting an engine – i.e., increasing the inlet mixture density to increase the trapped mass per cycle – is a very common means to increase the power density of an engine. As evident from Eq. 6.7, an increase in the inlet air density will directly increase the power of the engine. There are two major types of boosting technology: turbocharging and supercharging. The major difference between the two technologies is the former uses a centrifugal device (i.e., a turbine) to exploit available exhaust energy for conversion to shaft work whereas the latter absorbs shaft work via a direct mechanical connection to the engine. In both cases, the shaft work of the device is coupled to a boosting component – typically, either a centrifugal-based compressor or a positive-displacement compressor – which acts as an "air pump" to increase density of the inlet air. In most applications, a turbocharger uses a centrifugal turbine/compressor configuration while a supercharger uses a positive-displacement compressor.

Diesel engines are typically favorable engines to outfit with boosting devices (further, usually with turbochargers as they assist over the entire engine operating map). Boosting compensates for the diesel engine's typical use of fuel-lean

mixtures to improve its power density. Further, since diesel engines do not employ throttles for load control, boosting of a diesel engine provides benefit over the entire operating map. For this latter reason, and due to the general higher efficiency of a turbocharger over a supercharger, turbochargers are the most common boosting device on a diesel engine. Gasoline engines also can be outfitted with boosting devices, with additional complexity to consider. First, boosting devices on gasoline engines usually offer benefit only at wide-open throttle conditions; a point of operation for most applications of engines that is rarely used. Also, for this reason, directly coupled superchargers are often used where improved response to boost is provided. Second, increasing density of the inlet mixture tends to increase the propensity to knock. Thus, while boosting provides additional trapped mass to deliver increased power, a potential retard in timing to avoid fuel knock likely decreases fuel conversion efficiency and introduces a tradeoff in how much additional power can be expected from the boost. Third, because of the typical use of superchargers when boosting is applied to gasoline engines, the system efficiency tends to decrease as shaft work is transferred to provide the boosting action (whereas turbocharging uses available energy of the exhaust).

Often times, boosting devices – in particular, turbochargers – are viewed as devices to increase the overall system efficiency of the engine. In other words, it is thought that because a turbocharger uses exhaust energy that would otherwise be wasted, its conversion to useful work (i.e., boosting) should increase efficiency. This work transfer, however, usually does not leave the control system (i.e., the shaft work of the turbine is directly coupled to the pumping action of the compressor). In some instances, efficiency improvements can be realized if a "negative pumping loop" is created by boosting the intake manifold to a higher pressure than the exhaust manifold. It is also possible to improve the overall engine efficiency if boosting an engine enables the use of a smaller-sized engine for a given application; an effort, as described above, known as downsizing. In most instances, however, overall system efficiency may decrease even with the use of a turbocharger, as the major objective of increasing inlet density to increase power density requires additional energy transfer through the exhaust than out as shaft work. This latter aspect is often realized as an increase in exhaust manifold pressure to "drive" the turbocharger.

An area of technology development for turbocharging is the use of waste-gated and variable geometry turbochargers. In non-waste-gated or non-variable geometry turbochargers, the turbine has fixed geometry and thus fixed flow characteristics. As such, a fixed geometry turbocharger is restrictively designed to provide maximum benefit to the engine at a narrow operating range which usually centers on the peak power condition of the engine. As such, at low speed or low-load conditions, the turbocharger's boosting benefits are diminished. While decreasing the turbocharger's maximum benefit over a broader regime of the engine's operating map, such a constraint also typically creates a dynamic issue during engine accelerations known as "turbo lag." The large turbine designed for maximum engine flow rates requires substantial inertia to rotationally accelerate to maximum boosting benefit. Waste-gated and variable geometry turbochargers offer

opportunities to overcome such issues. In the case of a waste-gated turbocharger, usually a smaller turbine with less inertia is used with an exhaust "waste-gate." The smaller turbine provides faster response and better boosting at low-loads and speeds; upon approach of the engine's peak power condition (where either boosting becomes excessive for the intake system or turbocharger speeds exceed maximum limits) a waste-gate opens that allows exhaust energy to bypass the turbine. Thus, the turbocharger is supplied by a fraction of the available exhaust energy providing suitable boost at allowable rotational speeds of the turbocharger at the engine's peak power condition. A variable geometry turbocharger uses a similar concept, except that it is designed to change the flow momentum of the exhaust gases and create multiple pressure ratios for a given exhaust flow rate [113, 114].

Finally, it is noted that outfitting a boosting device to an engine is not a trivial task [115]. There is not an exclusive match between an engine and a given boosting device; instead, the match of a boosting device to its engine application is dictated by the objectives of adding the device, whether it is to exclusively increase power density, efficiency, and/or emissions of the engine system [116–118].

Advanced Engine Controls

Much of the advanced technology that appears on modern engines, or will appear on future engines, is enabled by advanced engine controls. Engine control has always been an integral component of the engine's success at delivering cost-effective and efficient power. Early engine control systems were purely mechanical and only concerned with controlling the level of power (e.g., throttle or rack position) or holding a constant speed with variable load (such as a generator system, using a speed governor). Modern-day engine control systems, however, are virtually all electrical-based, and at the very least sense several aspects of the engine's operation (e.g., cam position, throttle position, manifold temperatures and pressures, and air flow) and typically control most aspects of the engine (e.g., spark timing, injection timing, injection pressure, EGR level, and boost pressure) [119].

Like the engine itself, engine controls are becoming more sophisticated and advanced. Specifically, a general trend to use in-cylinder information as a feedback signal is an example of the type of complexity future engine control systems intend to resolve. Knowledge of in-cylinder pressure, for example, can provide immediate information to the engine controller about load produced by the engine. Additionally, in-cylinder pressure is the major property necessary for assessing the rate of energy release during the combustion process; having such information could allow engine control systems to change parameters based on a desired burn profile in the cylinder [120–122]. Further, along with the continuing advancements in engine model development (e.g., [46–50]), a trend toward model-based engine control [123, 124] intends to decrease engine development time and improve control over the several parameters now present on internal combustion engines.

Variable Geometry Engine Designs

Along with efforts to effect engine downsizing, and made possible with advancements in engine control systems, is the notion of variable geometry engine designs. In other words, conventional reciprocating internal combustion engines have mechanically fixed geometries, i.e., constant compression ratios and constant displaced volumes. A variable compression ratio is attractive, for example, since it might enable high compression ratio operation for a spark ignition engine at part-load conditions where there is decreased propensity for knock. Similarly, it could be used in diesel engines to avoid excessively high peak pressures at full load conditions. Variable displacement engines are attractive since they enable high peak powers, but use less throttle at part-load conditions (thus diminishing pumping losses).

An early concept of variable compression ratio was proposed by J. Atkinson, for whom the Atkinson Cycle is named. As described by [125], Atkinson's original conception involved mechanical linkages to displace the piston such that the engine's compression ratio is lower than its expansion ratio; the main idea being that more expansion work will yield higher efficiency. Similar in idea, but different in implementation, is the Miller concept [125] which uses either late intake valve closing or late exhaust valve opening to shorten or extend the compression or expansion strokes, respectively. Both concepts are attempted in modern-day applications, using both variable compression ratio techniques [112, 126, 127] and altered valve timing techniques [112, 127, 128]. The latter approach, of using altered valve timing techniques, is fluidly made possible through the use of variable valve timing [112, 127], a concept discussed in more detail in the next section.

While variable compression ratio concepts attempt to increase engine efficiency by way of increasing expansion, variable displacement engines attempt to increase engine efficiency by decreasing the use of throttle (thereby, decreasing the pumping work associated with the gas exchange process of a conventional spark ignition engine). With variable displacement, an engine is able to deliver high power using full displacement; at part-load conditions, rather than use throttle, cylinders can be "deactivated" so that they produce no power and allow the engine to deliver part-load power [129]. The deactivated cylinders typically continue to stroke and exchange gases; the closed portion of the cycle (i.e., compression and expansion) realize some loss due to heat transfer and friction but this is intended to be less than the gain realized through decreased pumping work.

Variable Valve Timing

Briefly discussed in the above sections is the idea of variable valve timing, or, the ability to change the valve events (i.e., intake and exhaust valve opening and closing) at any given point during the engine's operation [130]. In conventional

engine design, the valve events are "fixed" by mechanical positions of the lobes on the cam shaft. The effectiveness of an engine to induct and exhaust mixture depends on many things including, for example, engine speed, engine load, the use of EGR, spark timing, and injection timing. Thus, there are not valve events that will universally yield peak power, efficiency, and/or emission for any given engine design.

The idea of variable valve timing allows the engineer to decouple the valve events from the in-cylinder processes. In other words, flexibility of intake and exhaust is afforded with the use of variable valve timing. This not only allows the valve events to be uniquely tuned for each operating point of the engine for improved performance [131] and efficiency [132], but it also can enable other advanced technology. For example, having variable valve timing allows the implementation of the Miller approach to effecting variable compression ratio at part-load conditions, but enabling conventional operation at peak power conditions [127]. Another example is the use of variable valve timing to control the amount of residual fraction in the cylinder, which can affect emission of certain pollutants [133]; controlling residual fraction is a way to enable advanced modes of combustion, described in more detail below. Finally, variable valve timing can be used to replace a throttle, for example, for load control [134]; in doing so, trapped mass can be controlled without inducing pumping work in the engine. Although not widespread technology, variable valve timing is becoming more prevalent on modern-day engines.

Waste Heat Recovery

Waste heat recovery is the effort to take advantage of temperature gradients created between the engine and its environment. As described in section on "Thermodynamic Analysis of Internal Combustion Engines", thermal energy is transferred out of the system through heat transfer (e.g., through engine coolant) and exhaust flow. Because of the temperature gradient that exists between, for example, the high temperature exhaust and the low temperature surroundings, an orderly flow of thermal energy tends to cause the exhaust system to attain thermal equilibrium with the environment. Conventionally, this orderly flow of thermal energy is wasted (i.e., the heat transfer completely dissipates as generated entropy). It is practically possible to instead intercept the orderly flow of thermal energy and convert it to useful work. In the theoretical limit, this conversion of thermal energy to work energy is given by the completely reversible cycle, often called the Carnot cycle. The corresponding efficiency of useful work converted from thermal energy is thusly the Carnot efficiency, as is given by Eq. 6.16:

$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \qquad (6.16)$$

where $\eta_{th,Carnot}$ is the Carnot efficiency (maximum possible conversion of thermal energy to work energy), T_L is the sink's temperature (e.g., the environment temperature), and T_H is the source's temperature (e.g., the exhaust temperature). Considering that a typical automotive exhaust temperature may be 900 K operating in a 300 K environment, ideal efficiencies could be on the order of 67%. Of course, real process irreversibilities diminish actual device efficiencies from the ideal Carnot efficiency. In spite of the diminished efficiency from ideal, though, waste heat recovery in an automotive application, for example, is reported to decrease vehicle fuel consumption by as much as 7.4% [135].

There are several technologies available to make use of internal combustion engine exhaust waste heat energy; because of its spatial accessibility and concentrated high temperature, exhaust energy has been the focus of much of the waste heat recovery. Example major technologies include: (1) mechanical or electrical turbo-compounding/generating devices, (2) Rankine cycle-type devices, and (3) thermoelectric-type devices. Turbo-compounding or turbo-generating [136] converts exhaust thermal energy to mechanical energy (in the form of pressure and kinetic energy) and couples the mechanical energy either directly to the engine driveshaft (to deliver additional brake power) or to an electric generator. One disadvantage of turbo-compounding (like its turbocharging companion) is the method of converting thermal energy to mechanical energy; the centrifugal device increases an engine's exhaust pressure based on its operation. The increased exhaust pressure affects the engine's operation by altering the pumping work and initial mixture composition (increased exhaust species in the initial mixture); such factors can deteriorate the engine's cycle efficiency.

Rankine cycle-type devices make use of the thermodynamic Rankine cycle – typically with an organic fluid designed to undergo phase change within the temperature ranges of a typical engine exhaust system – to output shaft work for either direct-coupling to the engine driveshaft or electrical generation. In such a device, the engine exhaust stream provides the thermal energy to boil or vaporize the organic working fluid of the cycle. After becoming saturated vapor, the fluid expands through a turbine converting the thermal energy to mechanical energy; the cycle completes with the usual condenser and pump processes. Such a device is reported to increase the combined engine + waste heat recovery efficiency by up to 10% [137]; a potential downside is the added complexity of adding four processes (as opposed to one, for example, in the case of a turbomachine) to waste heat recovery system.

Another example major waste heat recovery device is the thermoelectric device. The thermoelectric effect, first observed by Seebeck in 1821, is the generation of a voltage due to a temperature difference between two junctions of two dissimilar materials [138]; principally, the Seebeck effect describes the operation of a thermocouple measurement of temperature. The practical use of the Seebeck effect to produce electricity as a thermoelectric device has recently emerged with semiconductor materials having favorable properties to transmit electricity with little resistance heating (Joule heating) and thermal conductivity (which would tend to

"short-circuit" the thermoelectric device). In fact, the advent of semiconductor materials first made possible practical thermoelectric devices as refrigerators exploiting the Peltier Effect (i.e., the opposite of the Seebeck effect, where an applied voltage creates a temperature difference between two junctions of two dissimilar metals) [138]. Now, however, thermoelectric devices create promise to exploit the available thermal energy in an internal combustion engine's exhaust for conversion to electricity [135, 139].

Direct Injection, Spark Ignition Engines

At the end of the section on "A Case Study: Diesel Engines Versus Gasoline Engines", it is suggested that spark ignition and compression ignition engines each have favorable features for power, efficiency, and emissions, but that each has conventionally designed limitations. Thus, it becomes attractive to design each ignition system's limitations out of the engine, and potentially realize gains in the engine's attributes (i.e., power, efficiency, and emissions). Two now-common technologies exist to do so: (1) direct injection spark ignition combustion and (2) homogeneous charge compression ignition combustion. This and the next section will describe these two technologies.

Direct injection spark ignition combustion attempts to create a stratified fuel–air mixture "charge" around the spark plug so that, at the point of spark release, the spark ignites a near-stoichiometric mixture. It is intended that outside of the stratification zone there is little to no fuel, thus creating an overall lean fuel–air mixture. The use of the word "stratified" to describe the fuel–air mixture is used here, as opposed to heterogeneous, to reinforce the notion that under ideal conditions the fuel–air mixture would be homogeneous (i.e., homogeneously stoichiometric) throughout the fuel–air mixture, but pure air outside the stratification zone (i.e., outside the fuel–air mixture). This is in contrast to the use of "heterogeneous" to describe a mixture (e.g., diesel engine mixture), where it is expected that substantial fuel–air ratio gradients exist throughout the fuel sprays.

In order to manifest the stratified mixture concept (see, e.g., [140–148]), fuel is injected directly into the cylinder. Typically, the combustion chamber of a direct injection spark ignition engine is specially designed to assist the stratification of the fuel–air mixture, and center it on the spark plug. The intake stroke draws air and residual mixture into the cylinder – notably, fuel is not inducted during intake as is done in conventional spark ignition operation. After intake, direct fuel injection into the cylinder usually occurs at some point during the piston's travel from BDC to TDC during the compression stroke. Spark advance is typically timed at around the same point as that in a conventional spark ignition engine (i.e., at a point near the piston reaching TDC-compression). The remaining processes of the direct injection spark ignition engine are basically the same as the conventional spark ignition engine.

There are several potential benefits of direct injection spark ignition combustion. Perhaps the clear benefit is the ability to use lean mixtures in a spark ignition engine. Because the combustion chamber is designed to stratify the mixture and create a stoichiometric mixture near the spark plug, the overall equivalence ratio of the mixture filling the entire chamber can be lean. As described in section on "Thermodynamic Analysis of Internal Combustion Engines", overall lean mixtures possess higher ratios of specific heats (γ), which translate to higher fuel conversion efficiencies. Another, less obvious, benefit of a stratified mixture is the decreased propensity to fuel knock (or, at least decreased propensity of fuel knock in the regions of the cylinder able to cause harm such as near cylinder walls and piston rings). The lack of a reactive mixture – manifested by charge stratification – in the regions furthest from the spark plug – which are the last to be controllably burned by the propagating flame – decreases the propensity that the mixture will autoignite and burn uncontrollably. Such a feature enables the direct injection spark ignition engine to have increased compression ratios relative to conventional spark ignition engines; again, this is an attribute that promotes an increase in efficiency of the novel engine concept. Finally, the use of direct fuel injection into the cylinder enables the elimination of a throttle to control engine load. In other words, engine load is controlled by the quantity of fuel injected into the cycle, similar to the load control of a diesel engine. Elimination of the throttle, as repeatedly described, will improve part-load efficiency by eliminating pumping losses effected by throttle.

Although the benefits are plentiful, the challenges are also present. From practical perspectives, perfect attainment of a stratified charge is difficult to accomplish. As such, heterogeneities within the stratified mixture emerge and can lead to products of incomplete combustion such as CO, HC, and PM. This, of course, is amplified at full load conditions; thus, peak power attainment through direct injection means alone would likely be limited by smoke limitations (similar to full load limitations of a diesel engine). Further, and like the challenges faced by diesel engines, outfitting direct injection spark ignition engines with after treatment devices is complicated by the use of overall lean mixtures (as described in section on "Emissions Formation and Exhaust Pollution"). In spite of such challenges, direct injection spark ignition engines are in production; continued development of in-cylinder flow modeling tools and engine controls contribute toward the concept's potential success.

Homogeneous Charge Compression Ignition Engines

An attractive feature of the conventional spark ignition engine is its use of a homogeneous mixture. Although this tends to promote knock (as described in section "Direct Injection Spark Ignition Engines"), it provides the benefit of being kinetically rate-limiting as opposed to mixing rate-limiting (as in the case of diesel engines). One issue with using a spark, or single point, to ignite a homogeneous mixture is the establishment of a flame that must propagate the mixture to convert

the chemical energy. In order to more quickly react the mixture, in a volumetric sense, multi-point ignition is required; i.e., ignition that occurs in several locations throughout the mixture will result in a faster burn rate. Such a voluminous ignition can be effected through compression of a homogenous mixture.

This is the basic idea of homogeneous charge compression ignition (HCCI); use compression to ignite a homogeneous mixture (similar to a diesel engine, except that diesel engines use compression to ignite an inherently heterogeneous mixture). The homogeneous mixture could, for example, be formed through premixing of fuel and air prior to mixture induction during the intake stroke. A faster burn rate effected by homogeneous charge compression ignition allows heat release to occur at near constant volume conditions; thus establishing the possibility to approach theoretical limits of maximum efficiencies of internal combustion engines. Further, because compression is used to ignite the mixture rather than spark, lean mixtures can be used at part-load conditions which further promotes higher efficiencies of the HCCI concept. Similar to the direct injection spark ignition concept, HCCI engines could eliminate throttles as load is controlled directly by the quantity of fuel mixed with the intake mixture. Finally, the common issue of particulate matter emissions faced by diesel engines (which also use compression ignition) are substantially decreased through the avoidance of locally rich fuel–air mixture zones (due to the use of a homogeneous mixture in an HCCI concept).

Of course, immediately the obvious problem becomes the method of ignition control. HCCI concepts do not have a direct ignition trigger, as do conventional gasoline (i.e., a spark) or diesel (i.e., direct fuel injection) engines. Controlling ignition in the HCCI concept depends on very precise control of the mixture's initial state at start of compression and the compression path followed up to the point of ignition. Several factors which are present – even in tightly controlled research environments – such as heat transfer, turbulence, and the history of preceding combustion events make the practical application of HCCI very challenging. The payoffs, of course, are correspondingly very high.

Practical implementation of HCCI is first reported by Onishi et al. [149] with theoretical developments experimentally provided by Najt and Foster [150]. Several control parameters could be adjusted such as compression ratio (e.g., with the use of variable valve timing), initial temperature, and quantity of residual fraction (e.g., effected either through exhaust gas recirculation or variable valve timing). Identifying the key control parameters, and the optimal way to adjust them during real-time operation of the engine, continue to be on-going research activities [151–156].

Advanced Compression Ignition Engines

A technique to control combustion of an HCCI engine is to use precisely metered amounts of residual fraction, which not only act to alter the kinetics of combustion but also result in substantially lower combustion temperatures. As such, much of

HCCI combustion is characterized by low temperature mechanisms commonly referred to as *low temperature combustion* (LTC). LTC offers a few benefits. First, efficiency improvements in the engine can be realized (in spite of increased exergy destruction due to low reaction temperatures) due to more favorable thermodynamic properties (i.e., higher ratio of specific heats, see Fig. 6.11) of the burned mixture and lower rates of heat transfer. Second, and typically the driver for LTC technology development, lower nitric oxide formation per the discussion in section on "Emissions Formation and Exhaust Pollution".

With this in mind, and reconsidering the prevailing issue of HCCI implementation – i.e., control of start of combustion – it becomes plausible to consider developing an "HCCI-type" mode of combustion in a diesel engine. That is, rather than induct a homogeneous mixture of fuel and air and rely upon indirectly controlled parameters to control ignition, perhaps fuel can be injected directly into the cylinder allowing for better control of ignition. In order to manifest LTC and harvest its benefits (e.g., possibly higher efficiency and substantially emission), high levels of EGR and strategic injection timings are used to extend ignition delay and create a nearly all-premixed combustion event. The long ignition delay, coupled with low temperature mechanisms, establishes the phenomenological observation of two-stage ignition characterized by the presence of cool-flame reactions [150, 157]. Interestingly, because of attainment of LTC, soot precursor formation is substantially abated, and the engine is made to operate with very low emissions of nitric oxide and particulate matter [158–168]. The combustion concept has become known as premixed compression ignition or premixed charge compression ignition combustion. The ability to attain LTC in compression ignition engines is attributed to the advancement of technology now in place on such machines, such as common-rail and electronic fuel pressure systems, variable geometry turbochargers, and exhaust gas recirculation systems.

Alternative Fuels

The term "alternative fuels" for an internal combustion engine is somewhat baseless, as an internal combustion engine has considerable flexibility in the type of fuel it uses. Of course, conventional fuels are the commonly called "gasoline" and "diesel" fuels, but generally engines have been shown to operate on virtually any gaseous, liquid, and solid dust particle specie that has heating value (i.e., will release thermal energy in a chemical oxidation process). Because of the wide variability of fuels available to internal combustion engines, this topic will not be expanded in this article. There are, however, certain considerations that should be given to the use of a fuel in an engine which was not intently designed for use with such fuel (e.g., use of ethanol in a gasoline engine or use of biodiesel in a diesel engine). First, ignition characteristics of the fuel may not be favorable for the particular engine design. For example, short-chain volatile hydrocarbons do not generally ignite well in conventional unmodified compression ignition engines of

typical compression ratios. Likewise, long-chain nonvolatile hydrocarbons do not generally vaporize well in conventional unmodified spark ignition engines. Second, flame temperatures of the combustion of the fuel may exceed material limits of any given engine construction. Third, fuels may react with other support components of the engine system (e.g., solvency of fuels with rubber hoses). Finally, combustion process will likely be altered when using an unconventional fuel in a conventional unmodified engine yielding different emissions, efficiency, and peak power capabilities. Thus, although internal combustion engines have inherent fuel-flexibility, their use with unconventional fuels is not straightforward and requires careful design and engineering considerations.

Abbreviations

BDC   Bottom dead center
EGR   Exhaust gas recirculation
HCCI Homogeneous charge compression ignition
IC   Internal combustion
LTC   Low temperature combustion
TDC   Top dead center

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