TABLE 4.10
Coefficients for species thermodynamic properties

Species	T range, K	a_{i1}	a_{i2}	a_{i3}	a_{i4}	a_{i5}	a_{i6}	a_{i7}
CO_2	1000–5000	0.44608(+1)	0.30982(−2)	−0.12393(−5)	0.22741(−9)	−0.15526(−13)	−0.48961(+5)	−0.98636(0)
	300–1000	0.24008(+1)	0.87351(−2)	−0.66071(−5)	0.20022(−8)	0.63274(−15)	−0.48378(+5)	0.96951(+1)
H_2O	1000–5000	0.27168(+1)	0.29451(−2)	−0.80224(−6)	0.10227(−9)	−0.48472(−14)	−0.29906(+5)	0.66306(+1)
	300–1000	0.40701(+1)	−0.11084(−2)	0.41521(−5)	−0.29637(−8)	0.80702(−12)	−0.30280(+5)	−0.32270(0)
CO	1000–5000	0.29841(+1)	0.14891(−2)	−0.57900(−6)	0.10365(−9)	−0.69354(−14)	−0.14245(+5)	0.63479(+1)
	300–1000	0.37101(+1)	−0.16191(−2)	0.36924(−5)	−0.20320(−8)	0.23953(−12)	−0.14356(+5)	0.29555(+1)
H_2	1000–5000	0.31002(+1)	0.51119(−3)	0.52644(−7)	−0.34910(−10)	0.36945(−14)	−0.87738(+3)	−0.19629(+1)
	300–1000	0.30574(+1)	0.26765(−2)	−0.58099(−5)	0.55210(−8)	−0.18123(−11)	−0.98890(+3)	−0.22997(+1)
O_2	1000–5000	0.36220(+1)	0.73618(−3)	−0.19652(−6)	0.36202(−10)	−0.28946(−14)	−0.12020(+4)	0.36151(+1)
	300–1000	0.36256(+1)	−0.18782(−2)	0.70555(−5)	−0.67635(−8)	0.21556(−11)	−0.10475(+4)	0.43053(+1)
N_2	1000–5000	0.28963(+1)	0.15155(−2)	−0.57235(−6)	0.99807(−10)	−0.65224(−14)	−0.90586(+3)	0.61615(+1)
	300–1000	0.36748(+1)	−0.12082(−2)	0.23240(−5)	−0.63218(−9)	−0.22577(−12)	−0.10612(+4)	0.23580(+1)
OH	1000–5000	0.29106(+1)	0.95932(−3)	−0.19442(−6)	0.13757(−10)	0.14225(−15)	0.39354(+4)	0.54423(+1)
NO	1000–5000	0.31890(+1)	0.13382(−2)	−0.52899(−6)	0.95919(−10)	−0.64848(−14)	0.98283(+4)	0.67458(+1)
O	1000–5000	0.25421(+1)	−0.27551(−4)	−0.31028(−8)	0.45511(−11)	−0.43681(−15)	0.29231(+5)	0.49203(+1)
H	1000–5000	0.25(+1)	0.0	0.0	0.0	0.0	0.25472(+5)	−0.46012(0)

Source: NASA Equilibrium Code.[9]

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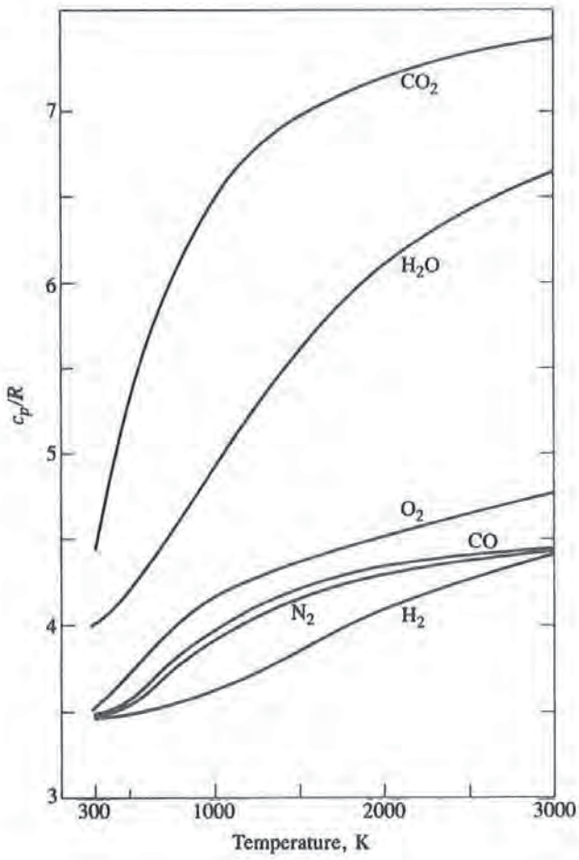


FIGURE 4-11
Specific heat at constant pressure, c_p/R, as function of temperature for species CO_2, H_2O, O_2, N_2, H_2, and CO. (*From JANAF tables.*[8])

$$\tilde{h}_f = A_{f1}\,t + A_{f2}\,\frac{t^2}{2} + A_{f3}\,\frac{t^3}{3} + A_{f4}\,\frac{t^4}{4} - \frac{A_{f5}}{t} + A_{f6} + A_{f8} \qquad (4.43)$$

where $t = T(\mathrm{K})/1000$. A_{f6} is the constant for the datum of zero enthalpy for C, H_2, O_2, and N_2 at 298.15 K. For a 0 K datum, A_{f8} is added to A_{f6}. For pure hydrocarbon compounds, the coefficients A_{fi} were found by fitting Eqs. (4.42) and (4.43) to data from Rossini *et al.*[16] Values for relevant pure fuels are given in Table 4.11. The units for $\tilde{c}_{p,f}$ are cal/gmol·K, and for $\tilde{h}_f$ are kcal/gmol.

Multicomponent fuel coefficients were determined as follows.[14] Chemical analysis of the fuel was performed to obtain the H/C ratio, average molecular weight, heating value, and the weight percent of aromatics, olefins, and total paraffins (including cycloparaffins). The fuel was then modeled as composed of a representative aromatic, olefin, and paraffin hydrocarbon. From atomic conservation of hydrogen and carbon and the chemical analysis results, component molar fractions and average carbon numbers can be determined. Table 4.11 gives values for the coefficients A_{f1} to A_{f8} for typical petroleum-based fuels. The units of the coefficients give $\tilde{c}_{p,f}$ and $\tilde{h}_f$ in cal/gmol·K and kcal/gmol, respectively, with $t = T(\mathrm{K})/1000$.

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TABLE 4.11
Coefficients for polynomials [Eqs. (4.42) and (4.43)] for fuel enthalpy and specific heat

Fuel	Formula	Molecular weight	$(A/F)_s$	$(F/A)_s$	A_{f1}	A_{f2}	A_{f3}	A_{f4}	A_{f5}	A_{f6}	A_{f8}
Methane	CH_4	16.04	17.23	0.0580	−0.29149	26.327	−10.610	1.5656	0.16573	−18.331	4.3000
Propane	C_3H_8	44.10	15.67	0.0638	−1.4867	74.339	−39.065	8.0543	0.01219	−27.313	8.852
Hexane	C_6H_{14}	86.18	15.24	0.0656	−20.777	210.48	−164.125	52.832	0.56635	−39.836	15.611
Isooctane	C_8H_{18}	114.2	15.14	0.0661	−0.55313	181.62	−97.787	20.402	−0.03095	−60.751	20.232
Methanol	CH_3OH	32.04	6.47	0.1546	−2.7059	44.168	−27.501	7.2193	0.20299	−48.288	5.3375
Ethanol	C_2H_5OH	46.07	9.00	0.1111	6.990	39.741	−11.926	0	0	−60.214	7.6135
Gasoline	$C_{8.26}H_{15.5}$	114.8	14.64	0.0683	−24.078	256.63	−201.68	64.750	0.5808	−27.562	17.792
	$C_{7.76}H_{13.1}$	106.4	14.37	0.0696	−22.501	227.99	−177.26	56.048	0.4845	−17.578	15.235
Diesel	$C_{10.8}H_{18.7}$	148.6	14.4	0.0694	−9.1063	246.97	−143.74	32.329	0.0518	−50.128	23.514

Units of A_{fi} such that $\tilde{h}_f$ is in kcal/gmol and $\tilde{c}_{p,f}$ is in cal/gmol·K with $t = T(\mathrm{K})/1000$.
A_{f6} gives enthalpy datum at 298.15 K; $(A_{f6} + A_{f8})$ gives enthalpy datum at 0 K.

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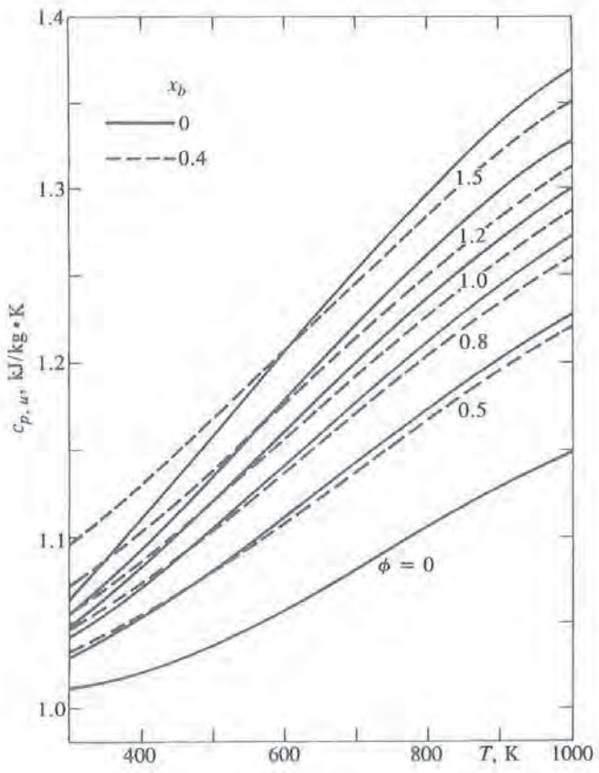


FIGURE 4-12
Specific heat at constant pressure of unburned gasoline, air, burned gas mixtures as function of temperature, equivalence ratio, and burned gas fraction. Units: kJ/kg mixture · K.

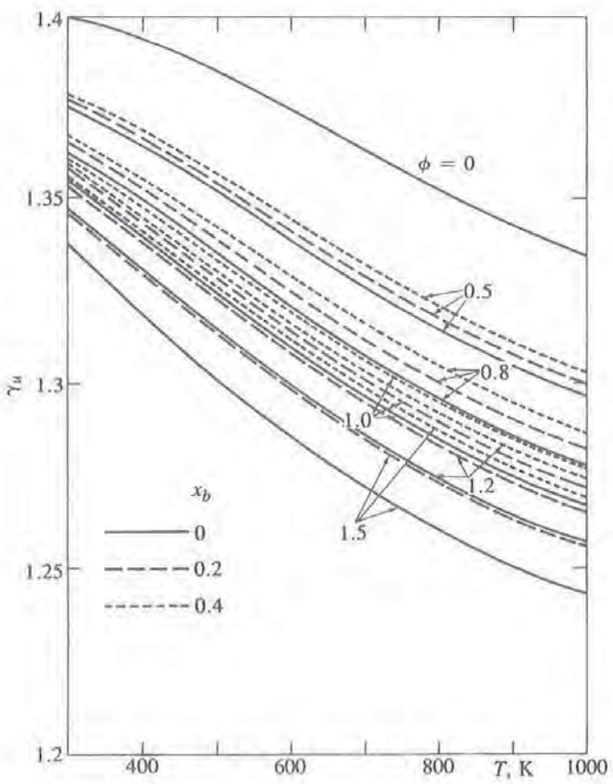


FIGURE 4-13
Ratio of specific heats, $\gamma_u = c_{p,u}/c_{v,u}$, of unburned gasoline, air, burned gas mixtures as function of temperature, equivalence ratio, and burned gas fraction.

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The thermodynamic properties of the unburned mixture can now be obtained. With the moles of each species per mole O_2, n_i, determined from Table 4.4, and the mass of mixture per mole O_2, m_{RP}, determined from Table 4.5, the unburned mixture properties are given by

$$c_{p,u} = \frac{1}{m_{RP}} \sum_i n_i \tilde{c}_{p,i} \tag{4.44a}$$

$$h_u = \frac{1}{m_{RP}} \sum_i n_i \tilde{h}_i \tag{4.44b}$$

$$s_u = \frac{1}{m_{RP}} \left\{ \sum n_i \left[\tilde{s}_i^\circ - \tilde{R} \ln \left(\frac{n_i}{n_u} \right) \right] - n_u \tilde{R} \ln p \right\} \tag{4.44c}$$

where p is in atmospheres.

Figures 4-12 and 4-13, obtained with the above relations, show how $c_{p,u}$ and $\gamma_u (= c_{p,u}/c_{v,u})$ vary with temperature, equivalence ratio, and burned gas fraction, for a gasoline-air mixture.

4.7.2 Burned Mixtures

The most accurate approach for burned mixture property and composition calculations is to use a thermodynamic equilibrium program at temperatures above about 1700 K and a frozen composition below 1700 K. The properties of each species at high and low temperatures are given by polynomial functions such as Eqs. (4.39) to (4.41) and their coefficients in Table 4.10. The NASA equilibrium program (see Sec. 3.7) is readily available for this purpose and is well documented.[9, 10] The following are examples of its output.

Figure 3-10 showed species concentration data for burned gases as a function of equivalence ratio at 1750, 2250, and 2750 K, at 30 atm. Figure 4-14 shows the burned gas molecular weight M_b, and Figs. 4-15 and 4-16 give $c_{p,b}$ and γ_b as functions of equivalence ratio at 1750, 2250, and 2750 K, at 30 atm. Figures 4-17 and 4-18 show $c_{p,b}$ and γ_b as a function of temperature and pressure for selected equivalence ratios for mixtures lean and rich of stoichiometric.[17] For rich mixtures ($\phi > 1$), for $T > 2000$ K, $c_{p,b}$ and γ_b are equilibrium values. For 1200 K $\leq T \leq$ 2000 K, "frozen" composition data are shown where the gas composition is in equilibrium at the given T and p but is frozen as c_p and c_v are computed. Below about 1500 K, fixed composition data are shown corresponding to a value of 3.5 for the water-gas equilibrium constant which adequately describes exhaust gases (see Sec. 4.9).

Because the computational time involved in repeated use of a full equilibrium program can be substantial, simpler equilibrium programs and approximate fits to the equilibrium thermodynamic data have been developed. The approach usually used is to estimate the composition and/or properties of undissociated combustion products and then to use iterative procedures or corrections to account for the effects of dissociation.

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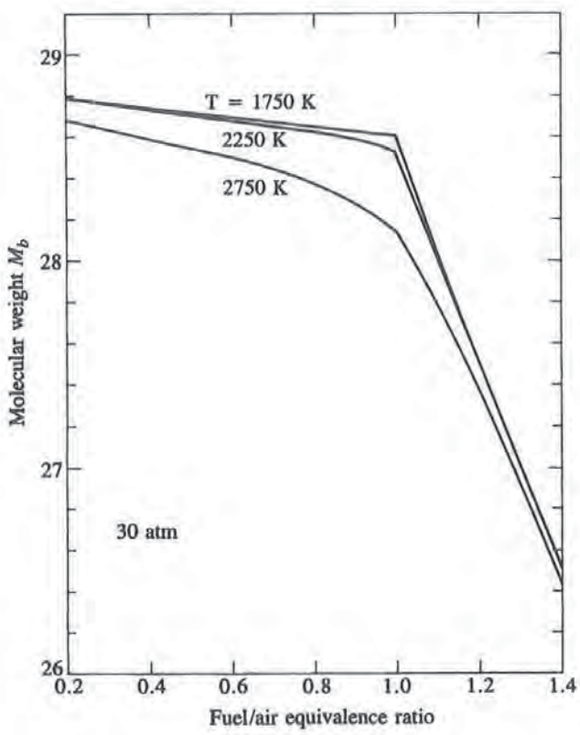


FIGURE 4-14
Molecular weight of equilibrium burned gases as a function of equivalence ratio at $T = 1750$, 2250, and 2750 K, and 30 atm. Fuel: isooctane.

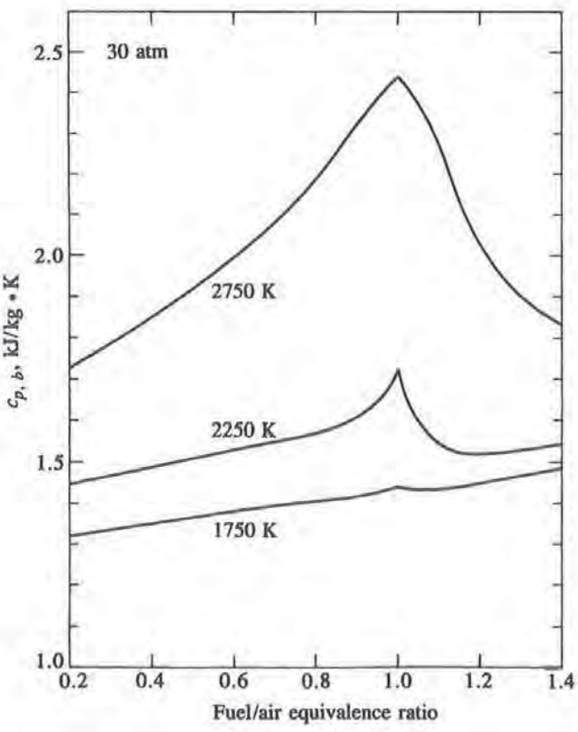


FIGURE 4-15
Specific heat at constant pressure of equilibrium burned gases as a function of equivalence ratio at $T = 1750$, 2250, and 2750 K, and 30 atm. Fuel: isooctane. Units: kJ/kg mixture · K.

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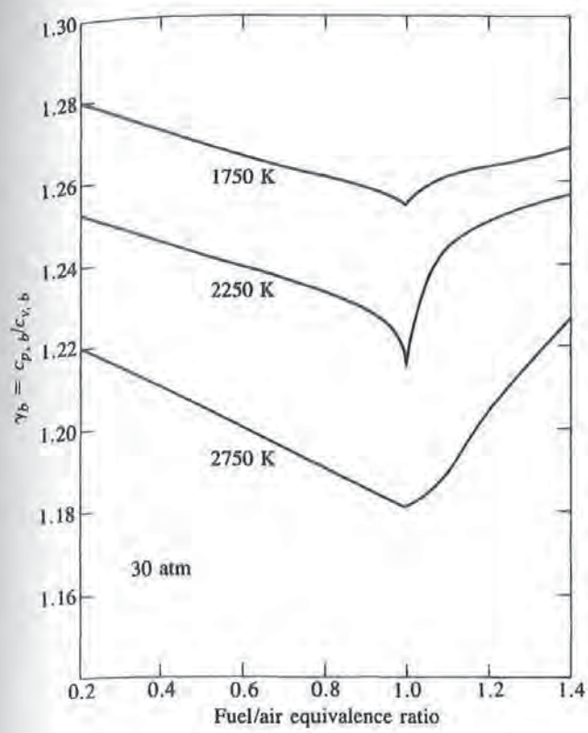
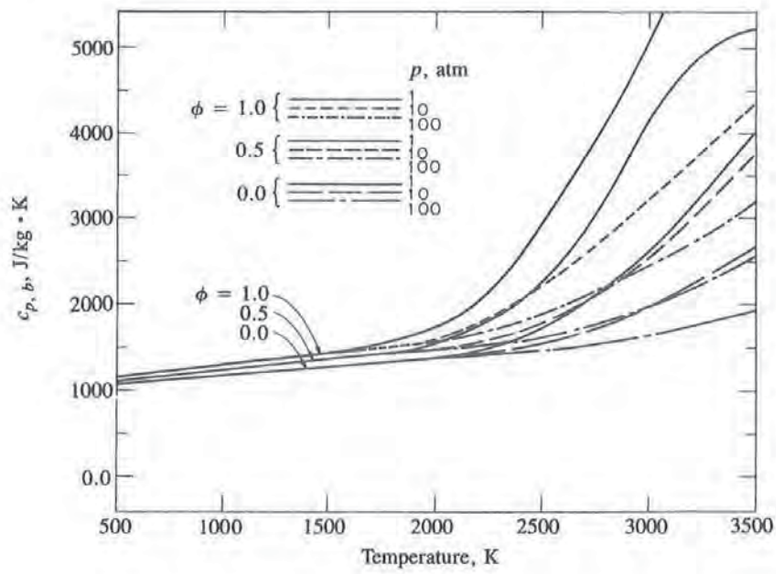


FIGURE 4-16
Ratio of specific heats, $\gamma_b = c_{p,b}/c_{v,b}$, for equilibrium burned gases as a function of equivalence ratio at $T = 1750$, 2250, and 2750 K, and 30 atm. Fuel: isooctane.

A computer program for calculating properties of equilibrium combustion products, designed specifically for use in internal combustion engine applications, has been developed by Olikara and Borman and is readily available.[18] The fuel composition ($C_nH_mO_lN_k$), fuel/air equivalence ratio, and product pressure and temperature are specified. The species included in the product mixture are: CO_2, H_2O, CO, H_2, O_2, N_2, Ar, NO, OH, O, H, and N. The element balance equations and equilibrium constants for seven nonredundant reactions provide the set of 11 equations required for solution of these species concentrations (see Sec. 3.7). The equilibrium constants are curve fitted from data in the JANAF tables.[8] The initial estimate of mole fractions to start the iteration procedure is the nondissociated composition. Once the mixture composition is determined, the thermodynamic properties and their derivatives with respect to temperature, pressure, and equivalence ratio are computed. This limited set of species has been found to be sufficiently accurate for engine burned gas calculations, and is much more rapid than the extensive NASA equilibrium program.[9, 10]

Several techniques for estimating the *thermodynamic properties* of high-temperature burned gases for engine applications have been developed. One commonly used approach is that developed by Krieger and Borman.[19] The internal energy and gas constant of undissociated combustion products were first described by polynomials in gas temperature. The second step was to limit the range of T and p to values found in internal combustion engines. Then the deviations between the equilibrium thermodynamic property data published by Newhall and Starkman[4, 5] and the calculated nondissociated values were fitted

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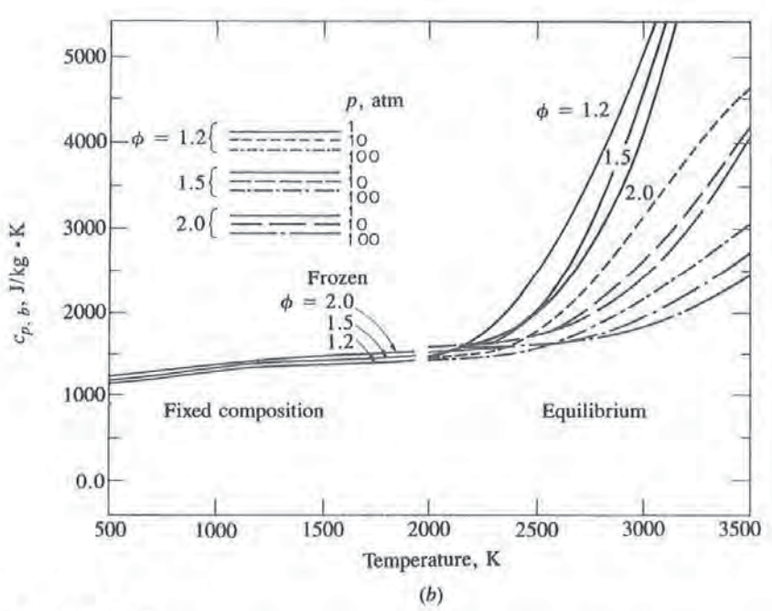


FIGURE 4-17
Specific heat at constant pressure for equilibrium, frozen, and fixed composition burned gases as a function of temperature and pressure: (*a*) equivalence ratio $\phi \leq 1.0$; (*b*) equivalence ratio $\phi > 1$. Units: J/kg mixture · K. Fuel: C_nH_{2n}.

by an exponential function of T, p, and ϕ. For $\phi \leq 1$, a single set of equations resulted. For $\phi \geq 1$, sets of equations were developed, each set applying to a specific value of equivalence ratio (see Ref. 19). In general, the fit for internal energy is within $2\frac{1}{2}$ percent over the pressure and temperature range of interest, and the error over most of the range is less than 1 percent. For many applica-

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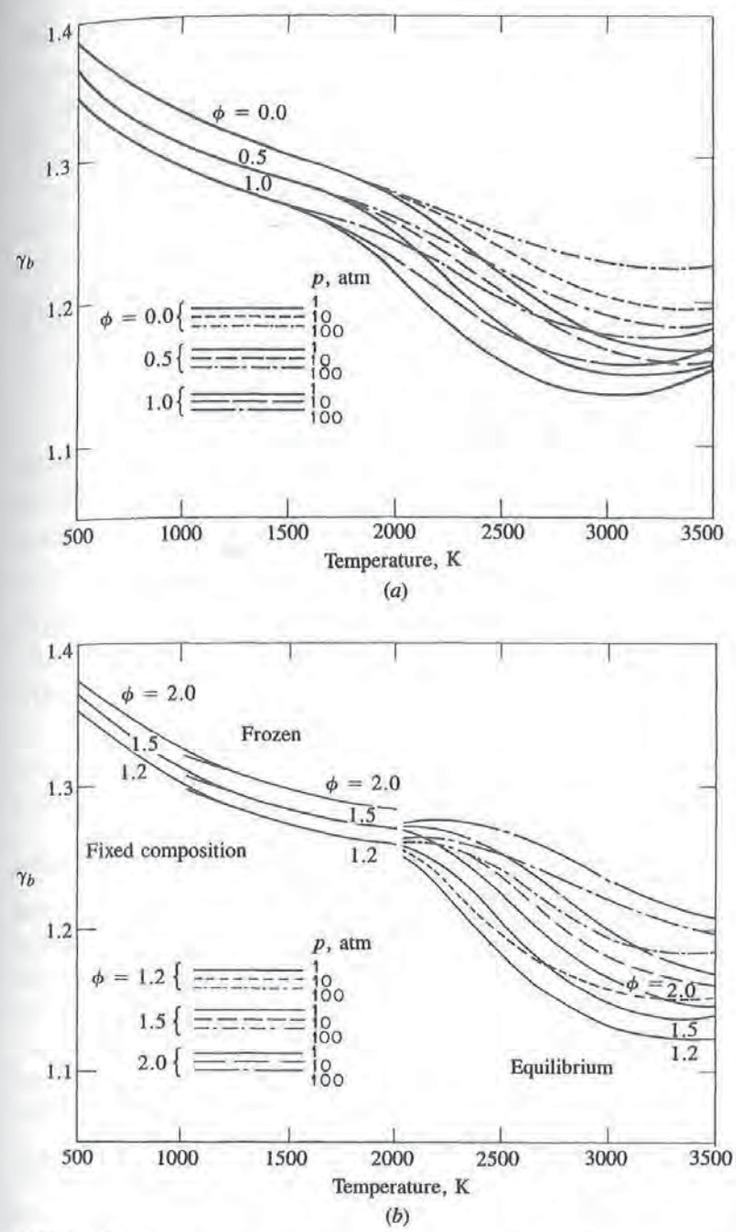


FIGURE 4-18
Ratio of specific heats, $\gamma_b = c_{p,b}/c_{v,b}$, for equilibrium, frozen, and fixed composition burned gases as a function of temperature and pressure: (*a*) equivalence ratio $\phi \leq 1.0$; (*b*) equivalence ratio $\phi > 1$. Fuel: C_nH_{2n}.

tions, the undissociated equations for thermodynamic properties are sufficiently accurate.

An alternative approach for property calculations, applicable to a wide range of hydrocarbon and alcohol fuels, is used extensively in the author's laboratory.[20] With this method, the products of combustion of hydrocarbon (or

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alcohol)-air mixtures are divided into triatomic, diatomic, and monatomic molecules, M_3, M_2, and M_1, respectively. Then, if Y is the extra number of moles of diatomic molecules due to dissociation of triatomic molecules and U is the extra number of monatomic molecules due to dissociation of diatomic molecules, the combustion reaction can be written as

$$\varepsilon\phi C + 2(1-\varepsilon)\phi H_2 + O_2 + \psi N_2 \rightarrow$$
$$[(2-\varepsilon)\phi - 2Y]M_3 + [1 - \phi + 3Y - U + \psi]M_2 + 2UM_1 \qquad \text{for } \phi \leq 1 \qquad (4.45)$$
$$[(2-\varepsilon\phi)\phi - 2Y]M_3 + [2(\phi-1) + 3Y - U + \psi]M_2 + 2UM_1 \qquad \text{for } \phi > 1$$

The method is based upon a fitting of data obtained from sets of detailed chemical equilibrium calculations to this functional form. Two general dissociation reactions:

$$2M_3 = 3M_2 \qquad \text{and} \qquad M_2 = 2M_1$$

are then used with fitted equilibrium constants $K_1(T)$ and $K_2(T)$ to calculate the relative species concentrations. This approach has been developed to give equations for enthalpy which sum the translational, rotational, and vibrational contributions to the specific heat, and the enthalpy of formation:

$$m_{RP}h = \frac{R}{2}(8N_3 + 7N_2 + 5N_1)T + R(3N_3 + N_2)\frac{T_v}{\exp(T_v/T) - 1} + m_{RP}h_f \qquad (4.46)$$

where N_3, N_2, and N_1 are the number of moles of triatomic, diatomic, and monatomic molecules respectively per mole O_2 reactant, T_v is a fitted vibrational temperature, m_{RP} is the mass of products per mole O_2 reactant [Eq. (4.9)], and h_f is the average specific enthalpy of formation of the products.

The molecular weight is given by

$$M_b = \frac{m_{RP}}{1 + (1-\varepsilon)\phi + \psi + Y + U} \qquad \text{for } \phi \leq 1 \qquad (4.47)$$

or

$$M_b = \frac{m_{RP}}{(2-\varepsilon)\phi + \psi + Y + U} \qquad \text{for } \phi > 1$$

U and Y are found using an approximate solution to the equations obtained by applying the fitted equilibrium constants to the dissociation reactions; h_f is obtained by fitting a correction to the undisssociated products enthalpy of formation. Equations are presented for the partial derivatives of enthalpy h and density ρ with respect to T, p, and ϕ.[20] These relationships have been tested for fuels with H/C ratios of 4 to 0.707, equivalence ratios 0.4 to 1.4, pressures 1 to 30 atm, and temperatures 1000 to 3000 K. The error for burned mixture temperatures relevant to engine calculations is always less than ± 10 K. The errors in density are less than ± 0.2 percent.

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4.8 TRANSPORT PROPERTIES

The processes by which mass, momentum, and energy are transferred from one point in a system to another are called rate processes. In internal combustion engines, examples of such processes are evaporation of liquid fuel, fuel-air mixing, friction at a gas/solid interface, and heat transfer between gas and the walls of the engine combustion chamber. In engines, most of these processes are turbulent and are therefore strongly influenced by the properties of the fluid flow. However, turbulent rate processes are usually characterized by correlations between dimensionless numbers (e.g., Reynolds, Prandtl, Nusselt numbers, etc.), which contain the fluid's transport properties of viscosity, thermal conductivity, and diffusion coefficient as well as the flow properties.

The simplest approach for computing the transport properties is based on the application of kinetic theory to a gas composed of hard-sphere molecules. By analyzing the momentum flux in a plane Couette flow,† it can be shown (Chapman and Cowling, Ref. 21, p. 218) that the viscosity μ of a monatomic hard-sphere gas [where $\mu = \tau/(du/dx)$, τ being the shear stress and (du/dx) the velocity gradient] is given by

$$\mu = \frac{[5/(16\sqrt{\pi})](m\tilde{k}T)^{1/2}}{d^2} \tag{4.48}$$

where m is the mass of the gas molecule, d is the molecular diameter, and $\tilde{k}$ is Boltzmann's constant, 1.381×10^{-23} J/K.

For such a gas, the viscosity varies as $T^{1/2}$, but will not vary with gas pressure or density. Measurements of viscosity show it does only vary with temperature, but generally not proportionally to $T^{1/2}$. The measured temperature dependence can only be explained with more sophisticated models for the intermolecular potential energy than that of a hard sphere. Effectively, at higher temperatures, the higher average kinetic energy of a pair of colliding molecules requires that they approach closer to each other and experience a greater repulsive force to be deflected in the collision. As a result, the molecules appear to be smaller spheres as the temperature increases.

An expression for the thermal conductivity k of a monatomic hard-sphere gas [$k = \dot{q}/(dT/dx)$, where $\dot{q}$ is the heat flux per unit area and dT/dx is the temperature gradient] can be derived from an analysis of the thermal equivalent of plane Couette flow (Ref. 21, p. 235):

$$k = \frac{[75/(64\sqrt{\pi})](\tilde{k}^3 T/m)^{1/2}}{d^2} \tag{4.49}$$

† In Couette flow, the fluid is contained between two infinite plane parallel surfaces, one at rest and one moving with constant velocity. In the absence of pressure gradients, the fluid velocity varies linearly across the distance between the surfaces.

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which has the same temperature dependence as μ. Equations (4.48) and (4.49) can be combined to give

$$k = \tfrac{5}{2}\mu c_v$$

since, for a monatomic gas, the specific heat at constant volume is $3\tilde{k}/(2m)$. This simple equality is in good agreement with measurements of μ and k for monatomic gases.

The above model does not take into account the vibrational and rotational energy exchange in collisions between polyatomic molecules which contribute to energy transport in gases of interest in engines. Experimental measurements of k and μ show that k is less than $\frac{5}{2}\mu c_v$ for such polyatomic gases, where c_v is the sum of the translational specific heat and the specific heat due to internal degrees of freedom. It was suggested by Eucken that transport of vibrational and rotational energy was slower than that of translational energy. He proposed an empirical expression

$$k = \frac{9\gamma - 5}{4}\mu c_v \tag{4.50}$$

or

$$\mathrm{Pr} = \frac{\mu c_p}{k} = \frac{4\gamma}{9\gamma - 5}$$

where Pr is the Prandtl number, which is in good agreement with experimental data.

A similar analysis of a binary diffusion process, where one gas diffuses through another, leads to an expression for the binary diffusion coefficient D_{ij}. D_{ij} is a transport property of the gas mixture composed of species i and j, defined by Fick's law of molecular diffusion which relates the fluxes of species i and j, Γ_{xi} and Γ_{xj}, in the x direction to the concentration gradients, dn_i/dx and dn_j/dx (n is the molecular number density):

$$\Gamma_{xi} = -D_{ij}\left(\frac{dn_i}{dx}\right) \qquad \Gamma_{xj} = -D_{ij}\left(\frac{dn_j}{dx}\right)$$

The binary diffusion coefficient for a mixture of hard-sphere molecules is (Ref. 21, p. 245)

$$D_{ij} = \frac{3}{16nd^2}\left(\frac{2\tilde{k}T}{\pi m_{ij}}\right)^{1/2} \tag{4.51}$$

where m_{ij} is the reduced mass $m_i m_j/(m_i + m_j)$.

A more rigorous treatment of gas transport properties, based on more realistic intermolecular potential energy models, can be found in Hirschfelder *et al.*,[22] who also present methods for computing the transport properties of mixtures of gases. The NASA computer program "Thermodynamic and Transport Properties of Complex Chemical Systems"[10] computes the viscosity, thermal conduc-

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tivity, and Prandtl number in addition to the thermodynamic calculations described in Secs. 3.7 and 4.7 for high-temperature equilibrium and frozen gas composition mixtures. The procedures used in the NASA program to compute these transport properties are based on the techniques described in Hirschfelder *et al.*[22] The NASA program has been used to compute the transport properties of hydrocarbon-air combustion products.[17] These quantities are functions of temperature T, equivalence ratio ϕ, and (except for viscosity) pressure p. Approximate correlations were then fitted to the calculated data of viscosity and Prandtl number. The principal advantage of these correlations is computational speed. For Prandtl number ($\mu c_p/k$), it was found convenient to use γ, the specific heat ratio (c_p/c_v), as an independent variable. Values of γ and c_p then permit determination of the thermal conductivity.

The viscosity of hydrocarbon-air combustion products over the temperature range 500 up to 4000 K, for pressures from 1 up to 100 atm, for $\phi = 0$ up to $\phi = 4$ is shown in Fig. 4-19. The viscosity as a function of temperature of hydrocarbon-air combustion products differs little from that of air. Therefore, a power law based on air viscosity data was used to fit the data:

$$\mu_{\text{air}}(\text{kg/m} \cdot \text{s}) = 3.3 \times 10^{-7} \times T^{0.7} \tag{4.52}$$

where T is in kelvins. The viscosity of combustion products is almost indepen-

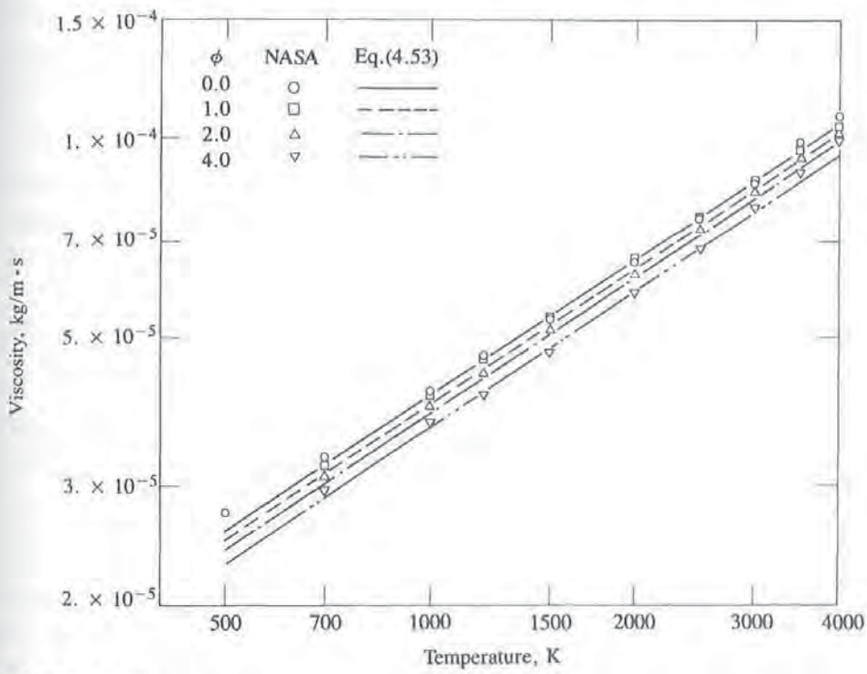


FIGURE 4-19
Viscosity, kg/m·s, of combustion products as a function of temperature and equivalence ratio. Equations shown are (4.52) and (4.53).

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dent of pressure. This correlation was corrected to include the effect of the equivalence ratio ϕ on the viscosity of hydrocarbon-air combustion products:

$$\mu_{\text{prod}} = \frac{\mu_{\text{air}}}{1 + 0.027\phi} \tag{4.53}$$

Figure 4-19 shows that the viscosity predicted using Eqs. (4.52) and (4.53) is very close to the viscosity values calculated with the NASA program. There is less than 4 percent error.

The Prandtl number of hydrocarbon-air combustion products has also been correlated over the above ranges of temperatures, pressures, and equivalence ratios. Since the expression for Prandtl number of a monatomic hard-sphere molecule gas is a function of γ, a second-order polynomial of γ was used to curve-fit the calculated Prandtl number data. A good fit to the data for lean combustion product mixtures was the following:

$$\text{Pr} = 0.05 + 4.2(\gamma - 1) - 6.7(\gamma - 1)^2 \qquad \phi \leq 1 \tag{4.54}$$

The values of Pr predicted with Eq. (4.54) are within 5 percent of the equilibrium Pr values calculated with the NASA program. For rich mixtures the following equation is a good fit to the equilibrium values of Pr using equilibrium values of γ, for temperatures greater than 2000 K:

$$\text{Pr} = \frac{0.05 + 4.2(\gamma - 1) - 6.7(\gamma - 1)^2}{1 + 0.015 \times 10^{-6}(\phi T)^2} \qquad 1 < \phi \leq 4 \tag{4.55}$$

The predicted values of Pr in this case are also close to the calculated values of Pr, with less than 10 percent error. Equation (4.55) is also a reasonable fit to the frozen values† of Pr for rich mixtures, using frozen values of γ, for the temperature range 1200 to 2000 K. As there are no data for Pr of rich mixtures at low temperatures, we suggest that where a fixed composition for the mixture is appropriate (e.g., during the exhaust process in an internal combustion engine), Eq. (4.55) can also be used with fixed composition values of γ.

The Prandtl number can be obtained from the above relations if γ is known. The thermal conductivity can be obtained from the Prandtl number if values of μ and c_p are known. Values of γ_b and $c_{p,b}$ as functions of temperature, pressure, and equivalence ratio are given in Figs. 4-15 to 4-18.

Since the fundamental relations for viscosity and thermal conductivity are complicated, various approximate methods have been proposed for evaluating these transport properties for gas mixtures. A good approximation for the vis-

† In the NASA program, "frozen" means the gas composition is in equilibrium at the given T and p, but is frozen as c_p, c_v, and k are computed.

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cosity of a multicomponent gas mixture is

$$\mu_{\text{mixt}} = \sum_{i=1}^{v} \frac{\tilde{x}_i^2}{\tilde{x}_i^2/\mu_i + 1.385 \sum_{j=1,\, j\neq 1}^{v} \tilde{x}_i \tilde{x}_j (\tilde{R}T/pM_i D_{ij})} \tag{4.56}$$

where $\tilde{x}_i$ and M_i are the mole fraction and molecular weight of the ith species, μ_i is the viscosity of the ith species, v is the number of species in the mixture, and D_{ij} is the binary diffusion coefficient for species i and j.[22]

4.9 EXHAUST GAS COMPOSITION

While the formulas for the products of combustion used in Sec. 3.4 are useful for determining unburned mixture stoichiometry, they do not correspond closely to the actual burned gas composition. At high temperatures (e.g., during combustion and the early part of the expansion stroke) the burned gas composition corresponds closely to the equilibrium composition at the local temperature, pressure, and equivalence ratio. During the expansion process, recombination reactions simplify the burned gas composition. However, late in the expansion stroke and during exhaust blowdown, the recombination reactions are unable to maintain the gases in chemical equilibrium and, in the exhaust process, the composition becomes frozen. In addition, not all the fuel which enters the engine is fully burned inside the cylinder; the combustion inefficiency even when excess air is present is a few percent (see Fig. 3-9). Also, the contents of each cylinder are not necessarily uniform in composition, and the amounts of fuel and air fed to each cylinder of a multicylinder engine are not exactly the same. For all these reasons, the composition of the engine exhaust gases cannot easily be calculated.

It is now routine to measure the composition of engine exhaust gases. This is done to determine engine emissions (e.g., CO, NO_x, unburned hydrocarbons, and particulates). It is also done to determine the relative proportions of fuel and air which enter the engine so that its operating equivalence ratio can be computed. In this section, typical engine exhaust gas composition will be reviewed, and techniques for calculating the equivalence ratio from exhaust gas composition will be given.

4.9.1 Species Concentration Data

Standard instrumentation for measuring the concentrations of the major exhaust gas species has been developed.[23] Normally a small fraction of the engine exhaust gas stream is drawn off into a sample line. Part of this sample is fed directly to the instrument used for unburned hydrocarbon analysis, a flame ionization detector (FID). The hydrocarbons present in the exhaust gas sample are burned in a small hydrogen-air flame, producing ions in an amount proportional to the number of carbon atoms burned. The FID is effectively a carbon atom counter. It is calibrated with sample gases containing known amounts of hydrocarbons. Unburned hydrocarbon concentrations are normally expressed as a mole fraction

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or volume fraction in parts per million (ppm) as C_1. Sometimes results are expressed as ppm propane (C_3H_8) or ppm hexane (C_6H_{14}); to convert these to ppm C_1 multiply by 3 or 6, respectively. Older measurements of unburned hydrocarbons were often made with a nondispersive infrared (NDIR) analyzer, where the infrared absorption by the hydrocarbons in a sample cell was used to determine their concentration.[23] Values of HC concentrations in engine exhaust gases measured by an FID are about two times the equivalent values measured by an NDIR analyzer (on the same carbon number basis, e.g., C_1). NDIR-obtained concentrations are usually multiplied by 2 to obtain an estimate of actual HC concentrations. Substantial concentrations of oxygen in the exhaust gas affect the FID measurements. Analysis of *unburned* fuel-air mixtures should be done with special care.[23] To prevent condensation of hydrocarbons in the sample line (especially important in diesel exhaust gas), the sample line is often heated.

NDIR analyzers are used for CO_2 and CO concentration measurements. Infrared absorption in a sample cell containing exhaust gas is compared to absorption in a reference cell. The detector contains the gas being measured in two compartments separated by a diaphragm. Radiation not absorbed in the sample cell is absorbed by the gas in the detector on one side of the diaphragm. Radiation not absorbed in the reference cell is absorbed by the gas in the other half of the detector. Different amounts of absorption in the two halves of the detector result in a pressure difference being built up which is measured in terms of diaphragm distention. NDIR detectors are calibrated with sample gases of known composition. Since water vapor IR absorption overlaps CO_2 and CO absorption bands, the exhaust gas sample is dried with an ice bath and chemical dryer before it enters the NDIR instrument.

Oxygen concentrations are usually measured with paramagnetic analyzers. Oxides of nitrogen, either the amount of nitric oxide (NO) or total oxides of nitrogen ($NO + NO_2$, NO_x), are measured with a chemiluminescent analyzer. The NO in the exhaust gas sample stream is reacted with ozone in a flow reactor. The reaction produces electronically excited NO_2 molecules which emit radiation as they decay to the ground state. The amount of radiation is measured with a photomultiplier and is proportional to the amount of NO. The instrument can also convert any NO_2 in the sample stream to NO by decomposition in a heated stainless steel tube so that the total NO_x ($NO + NO_2$) concentration can be determined.[23] Gas chromatography can be used to determine all the inorganic species (N_2, CO_2, O_2, CO, H_2) or can be used to measure the individual hydrocarbon compounds in the total unburned hydrocarbon mixture. Particulate emissions are measured by filtering the particles from the exhaust gas stream onto a previously weighed filter, drying the filter plus particulate, and reweighing.

SPARK-IGNITION ENGINE DATA. Dry exhaust gas composition data, as a function of the fuel/air equivalence ratio, for several different multi- and single-cylinder automotive spark-ignition engines over a range of engine speeds and loads are shown in Fig. 4-20. The fuel compositions (gasolines and isooctane) had H/C ratios ranging from 2.0 to 2.25. Exhaust gas composition is substantially

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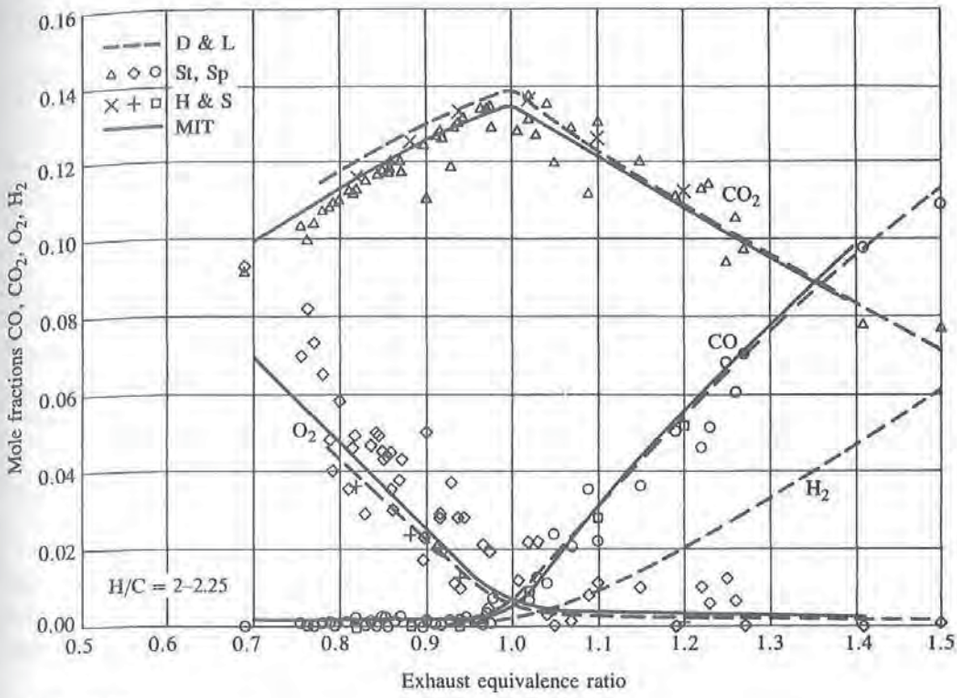


FIGURE 4-20
Spark-ignition engine exhaust gas composition data in mole fractions as a function of fuel/air equivalence ratio. Fuels: gasoline and isooctane, H/C 2 to 2.25. (*From D'Alleva and Lovell,*[24] *Stivender,*[25] *Harrington and Shishu,*[26] *Spindt,*[27] *and data from the author's laboratory at MIT.*)

different on the lean and the rich side of the stoichiometric air/fuel or fuel/air ratios; thus, the fuel/air equivalence ratio ϕ (or its inverse, the relative air/fuel ratio λ) is the appropriate correlating parameter. On the lean side of stoichiometric, as ϕ decreases, CO_2 concentrations fall, oxygen concentrations increase, and CO levels are low but not zero ($\sim$0.2 percent). On the rich side of stoichiometric, CO and H_2 concentrations rise steadily as ϕ increases and CO_2 concentrations fall. O_2 levels are low ($\sim$0.2 to 0.3 percent) but are not zero. At stoichiometric operation, there is typically half a percent O_2 and three-quarters of a percent CO.

Fuel composition has only a modest effect on the magnitude of the species concentrations shown. Measurements with a wide range of liquid fuels show that CO concentrations depend only on the equivalence ratio or relative fuel/air ratio (see Fig. 11-20).[26] A comparison of exhaust CO concentrations with gasoline, propane (C_3H_8), and natural gas (predominantly methane, CH_4) show that only with the high H/C ratio of methane, and then only for CO $\geq$ 4 percent, is fuel composition significant.[28] The values of CO_2 concentration at a given ϕ are slightly affected by the fuel H/C ratio. For example, for stoichiometric mixtures with 0.5 percent O_2 and 0.75 percent CO, as the H/C ratio decreases CO_2 concentrations increase from 13.7 percent for isooctane (H/C = 2.25), to 14.2 to 14.5 percent for typical gasolines (H/C in range 2–1.8), to 16 for toluene (H/C = 1.14).[29]

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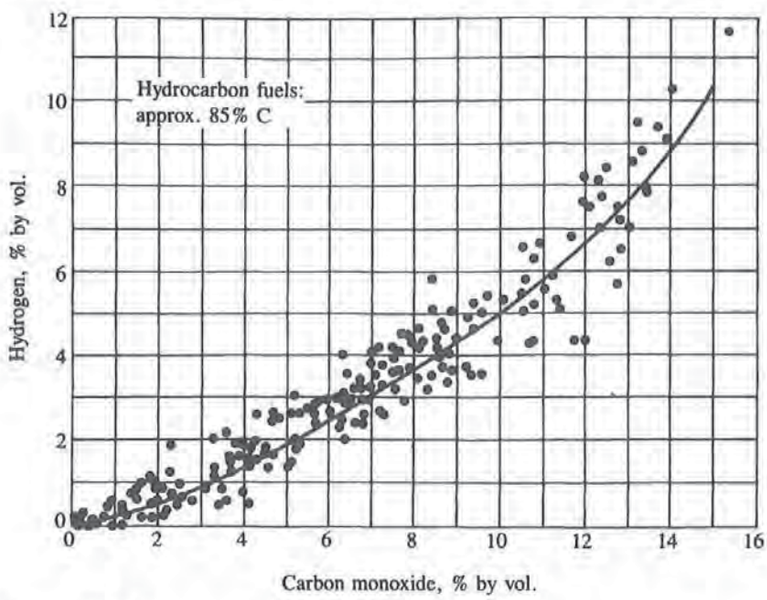


FIGURE 4-21
Hydrogen concentration in spark-ignition engine exhaust as a function of carbon monoxide concentration. Units: percent by volume.[30]

Unburned hydrocarbon exhaust concentrations vary substantially with engine design and operating conditions. Spark-ignition engine exhaust levels in a modern low-emission engine are typically of the order of 2000 ppm C_1 with liquid hydrocarbon fuels, and about half that level with natural gas and propane fuels.

Hydrogen concentrations in engine exhaust are not routinely measured. However, when the mixture is oxygen-deficient—fuel rich—hydrogen is present with CO as an incomplete combustion product. Figure 4-21 summarizes much of the available data on H_2 concentrations plotted as a function of CO.[30]

DIESEL EXHAUST DATA. Since diesels normally operate significantly lean of stoichiometric ($\phi \leq 0.8$) and the diesel combustion process is essentially complete (combustion inefficiency is ≤ 2 percent), their exhaust gas composition is straightforward. Figure 4-22 shows that O_2 and CO_2 concentrations vary linearly with the fuel/air equivalence ratio over the normal operating range. Diesel emissions of CO and unburned HC are low.

4.9.2 Equivalence Ratio Determination from Exhaust Gas Constituents

Exhaust gas composition depends on the relative proportions of fuel and air fed to the engine, fuel composition, and completeness of combustion. These relationships can be used to determine the operating fuel/air equivalence ratio of an engine from a knowledge of its exhaust gas composition. A general formula for

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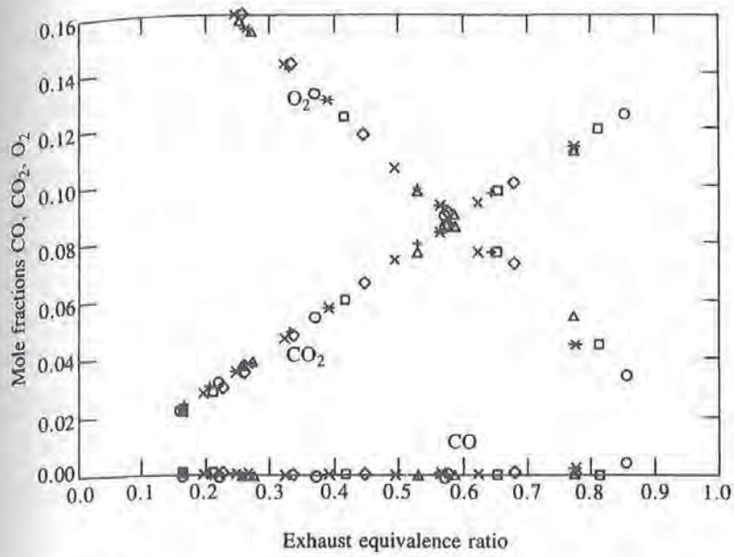


FIGURE 4-22
Exhaust gas composition from several diesel engines in mole fractions on a dry basis as a function of fuel/air equivalence ratio.[31]

the composition of fuel can be represented as $C_nH_mO_r$. For conventional petroleum-based fuels, oxygen will be absent; for fuels containing alcohols, oxygen will be present. The overall combustion reaction can be written as

$$\text{Fuel} + \text{oxidizer} \rightarrow \text{products}$$

The fuel is $C_nH_mO_r$; the oxidizer is air ($O_2 + 3.773N_2$). The products are CO_2, H_2O, CO, H_2, O_2, NO_x, N_2, unburned hydrocarbons (unburned fuel and products of partial fuel reaction), and soot particles (which are mainly solid carbon). The amount of solid carbon present is usually sufficiently small (≤ 0.5 percent of the fuel mass) for it to be omitted from the analysis. The overall combustion reaction can be written explicitly as

$$\begin{aligned} C_nH_mO_r + \frac{n_{O_2}}{\phi}(O_2 + 3.773N_2) = n_P(&\tilde{x}_{C_aH_b}C_aH_b + \tilde{x}_{CO}CO + \tilde{x}_{CO_2}CO_2 \\ &+ \tilde{x}_{O_2}O_2 + \tilde{x}_{N_2}N_2 + \tilde{x}_{NO}NO \\ &+ \tilde{x}_{NO_2}NO_2 + \tilde{x}_{H_2O}H_2O + \tilde{x}_{H_2}H_2) \end{aligned} \qquad (4.57)$$

where ϕ is the measured equivalence ratio $[(F/A)_{\text{actual}}/(F/A)_{\text{stoichiometric}}]$, n_{O_2} is the number of O_2 molecules required for complete combustion ($n + m/4 - r/2$), n_P is the total number of moles of exhaust products, and $\tilde{x}_i$ is the mole fraction of the ith component.

There are several methods for using Eq. (4.57) to determine ϕ, the equivalence ratio, depending on the amount of information available. Normally CO_2, CO, O_2, NO_x concentrations as mole fractions and unburned hydrocarbon (as

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mole fraction or ppm C_1, i.e., $\tilde{x}_{CH_{b/a}}$) are measured. The concentration of the inorganic gases are usually measured *dry* (i.e., with H_2O removed) or *partially dry*. Unburned hydrocarbons may be measured *wet* or *dry* or *partially dry*. NO_x is mainly nitric oxide (NO); its concentration is usually sufficiently low (<0.5 percent) for its effect on equivalence ratio determination to be negligibly small. Thus, in Eq. (4.57) there are seven unknowns which are: ϕ, $\tilde{x}_{H_2}$, $\tilde{x}_{H_2O}$, $\tilde{x}_{N_2}$, n_P, a, b. (There will be additional unknowns if the measurements listed above are incomplete.)

To solve for these unknowns we need seven additional equations. We can obtain five equations using an atomic balance for each element and the definition of mole fraction, as follows:

Carbon balance:

$$n = n_P(a\tilde{x}_{C_aH_b} + \tilde{x}_{CO} + \tilde{x}_{CO_2}) \tag{4.58}$$

Hydrogen balance:

$$m = n_P(b\tilde{x}_{C_aH_b} + 2\tilde{x}_{H_2O} + 2\tilde{x}_{H_2}) \tag{4.59}$$

Oxygen balance:

$$r + \frac{2n_{O_2}}{\phi} = n_P(\tilde{x}_{CO} + 2\tilde{x}_{CO_2} + \tilde{x}_{NO} + 2\tilde{x}_{O_2} + \tilde{x}_{H_2O}) \tag{4.60}$$

Nitrogen balance:

$$\frac{7.546n_{O_2}}{\phi} = n_P(2\tilde{x}_{N_2} + \tilde{x}_{NO}) \tag{4.61}$$

Mole fractions add up to 1:

$$\tilde{x}_{C_aH_b} + \tilde{x}_{CO} + \tilde{x}_{H_2} + \tilde{x}_{H_2O} + \tilde{x}_{N_2} + \tilde{x}_{NO} + \tilde{x}_{CO_2} + \tilde{x}_{O_2} = 1 \tag{4.62}$$

An additional assumption is made, based on available exhaust gas composition data, that CO_2, CO, H_2O, and H_2 concentrations are related by

$$\frac{\tilde{x}_{CO}\,\tilde{x}_{H_2O}}{\tilde{x}_{CO_2}\,\tilde{x}_{H_2}} = K \tag{4.63}$$

where K is a constant.† Values of 3.8[24, 25] and 3.5[27] are commonly used for K. The difference between these values has little effect on the computed magnitude of ϕ. To complete the analysis, various assumptions are made concerning the composition and relative importance of the unburned hydrocarbons. The most common approaches are summarized below.

† Equation (4.63) is often described as assuming a specific value for the water-gas reaction equilibrium constant. In fact K is an empirical constant determined from exhaust gas composition data.

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OXYGEN BALANCE AIR/FUEL AND EQUIVALENCE RATIOS. For fuels comprised of carbon and hydrogen only, when all species are measured with the *same background moisture* (wet, dry, or partially dry), the following expression based on the ratio of measured and computed oxygen-containing species to measured carbon-containing species gives the air/fuel ratio. It has been assumed that the unburned hydrocarbons have the same C/H ratio as the fuel:[32]

$$\left(\frac{A}{F}\right) = 4.773 \left(\frac{M_{\text{air}}}{M_f}\right) \frac{(\text{CO}_2) + (\text{CO})/2 + (\text{H}_2\text{O})/2 + (\text{NO})/2 + (\text{NO}_2) + (\text{O}_2)}{(\text{HC}) + (\text{CO}) + (\text{CO}_2)} \tag{4.64}$$

where ( ) are molar concentrations (all with the same background moisture) in percent, $M_{\text{air}} = 28.96$, $M_f = 12.01 + 1.008y$ where y is the H/C ratio of the fuel, (HC) is molar percent unburned hydrocarbons as C_1, and

$$(\text{H}_2\text{O}) = 0.5y \frac{(\text{CO}_2) + (\text{CO})}{(\text{CO})/[K(\text{CO}_2)] + 1} \tag{4.65}$$

Since nitrogen oxides collectively comprise less than 0.5 percent of the exhaust mixture, their concentrations can be omitted with negligible error.

The fuel/air equivalence ratio ϕ is obtained from the ratio of the stoichiometric air/fuel ratio [Eq. (3.6)] and Eq. (4.64) above.

CARBON BALANCE AIR/FUEL AND EQUIVALENCE RATIOS. When oxygen analysis is not available, for fuels comprised of carbon and hydrogen only, a carbon balance air/fuel ratio may be employed:[25, 32]

$$\frac{A}{F} = \frac{M_{\text{air}}}{M_f} \left[\frac{100 + (\text{HC}) - (\text{CO})/2 + 3(\text{H}_2\text{O})/2 - (\text{H}_2\text{O})_a}{(\text{HC}) + (\text{CO}) + (\text{CO}_2)} - \frac{y}{2} \right] \tag{4.66}$$

The symbols are as defined above. (H_2O) is the molar percent water in the combustion products defined by Eq. (4.65) and $(\text{H}_2\text{O})_a$ is the molar percent water vapor at the analyzers.

This carbon balance (A/F) is sensitive to moisture concentration at the analyzers. The use of ice bath exhaust sample chillers generally reduces the $(\text{H}_2\text{O})_a$ term to less than 1 percent and little accuracy is then lost by neglecting it. For completely "wet" analysis (uncondensed), $(\text{H}_2\text{O})_a = (\text{H}_2\text{O})$, and Eq. (4.66) is accurate. For partially dry exhaust gas analysis, knowledge of the dew point of the mixture will provide the $(\text{H}_2\text{O})_a$ term by reference to steam tables.

The fuel/air equivalence ratio ϕ is obtained from the ratio of the stoichiometric air/fuel ratio [Eq. (3.6)] and Eq. (4.66) above.

EQUIVALENCE RATIO BASED ON WET HC AND DRY INORGANIC GAS ANALYSIS. Engine exhaust gas composition is often determined by analyzing a fully dried sample stream for CO_2, CO, O_2, and NO_x, and a fully wet (uncondensed) stream with an FID for unburned hydrocarbons. Equations (4.64) and (4.66) are not applicable under these circumstances. The following equations

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define the exhaust gas composition and equivalence ratio under these conditions. The notation $\tilde{x}_i$ denotes the wet mole fraction of species i and $\tilde{x}_i^*$ denotes the dry mole fraction of species i. Equations (4.57) to (4.62), with Eq. (4.63) to relate CO_2, CO, H_2O, and H_2 concentrations and the assumption that $b/a = m/n$, were used to derive these results. The equations apply for a fuel of composition $C_mH_nO_r$.

The fuel/air equivalence ratio is given by

$$\phi = \frac{2n_{O_2}}{n_P \tilde{x}_{H_2O} + n_P(1 - \tilde{x}_{H_2O})(\tilde{x}^*_{CO} + 2\tilde{x}^*_{CO_2} + 2\tilde{x}^*_{O_2} + \tilde{x}^*_{NO} + 2\tilde{x}^*_{NO_2}) - r} \tag{4.67}$$

where the wet and dry mole fractions are related by

$$\tilde{x}_i = (1 - \tilde{x}_{H_2O})\tilde{x}_i^*$$

and

$$\begin{aligned} n_P &= \frac{n}{\tilde{x}_{CH_{b/a}} + (1 - \tilde{x}_{H_2O})(\tilde{x}^*_{CO} + \tilde{x}^*_{CO_2})} \\ \tilde{x}_{H_2O} &= \frac{m}{2n} \frac{\tilde{x}^*_{CO} + \tilde{x}^*_{CO_2}}{[1 + \tilde{x}^*_{CO}/(K\tilde{x}^*_{CO_2}) + (m/2n)(\tilde{x}^*_{CO} + \tilde{x}^*_{CO_2})]} \\ \tilde{x}_{H_2} &= \frac{\tilde{x}_{H_2O}\,\tilde{x}^*_{CO}}{K\tilde{x}^*_{CO_2}} \\ \tilde{x}_{N_2} &= \frac{3.773 n_{O_2}}{\phi n_P} - (1 - \tilde{x}_{H_2O})\frac{(\tilde{x}^*_{NO} + \tilde{x}^*_{NO_2})}{2} \end{aligned} \tag{4.68}$$

Note that $\tilde{x}_{CH_{b/a}}$ is the measured (wet) HC concentration as a mole fraction C_1 (ppm $C_1 \times 10^{-6}$): $\tilde{x}_{CH_{b/a}} = a\tilde{x}_{C_aH_b}$. Figure 4-23 shows wet exhaust gas concentrations, based on the MIT measured dry concentrations of CO_2, CO, O_2 shown in Fig. 4-20, and wet HC concentration, as well as Eqs. (4.68).

For lean mixtures, varying the value of K between 1.5 and 5.5 had a negligible effect on the value of ϕ computed from Eq. (4.67). For stoichiometric mixtures, varying K from 2.5 to 4.5 varied the computed ϕ by 2 to 3 percent. For $\phi \approx 1.2$, varying K from 2.5 to 4.5 varied the computed ϕ by 3 to 4 percent. The error in ϕ involved in omitting NO_x is 0.2 percent for an NO_x level of 1000 ppm, increasing to 1 percent for an NO_x level of 5000 ppm. The sensitivity of the computed ϕ to errors in the measurements of CO_2, CO, and O_2 is modest within the normal range of ϕ used. A 2 percent error in CO_2 or CO or O_2 at $\phi \approx 1$ gives about a 0.1 percent error in computed ϕ. For leaner and richer mixtures, the error in ϕ increases for errors in measured CO_2 concentration, and CO ($\phi > 1$) and O_2 ($\phi < 1$) concentrations, but is still significantly less than the measurement error in fuel and air flow.

4.9.3 Effects of Fuel/Air Ratio Nonuniformity

Neither the masses of air inducted into the different cylinders of a multicylinder engine per cycle nor the masses of fuel which enter the different cylinders per

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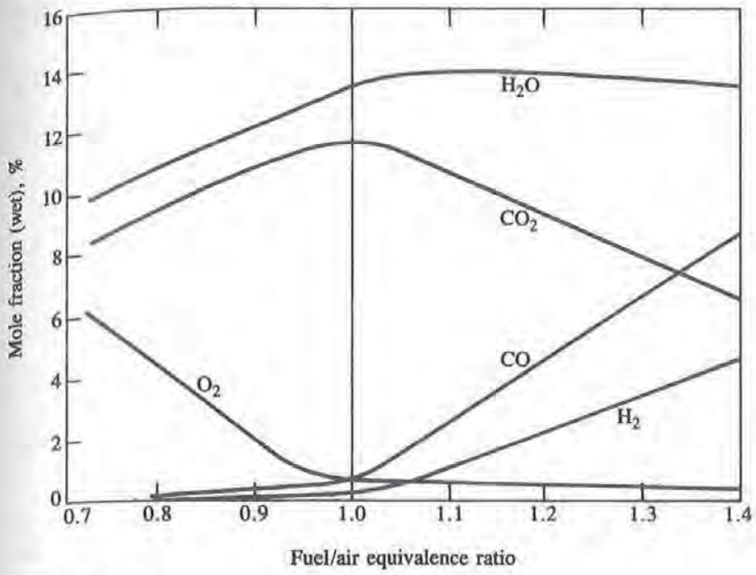


FIGURE 4-23
Wet exhaust gas species concentrations as a function of fuel/air equivalence ratio, based on the dry exhaust gas composition data in Fig. 4-20 and Eqs. (4.68).

cycle are exactly equal. In addition, mixing of fuel and air within each cylinder is not necessarily completely uniform. Thus the exhaust gas composition may correspond to a distribution in the fuel/air ratio in the unburned mixture about the mean value. For example, if the mean fuel/air ratio is stoichiometric, extra oxygen will be contributed by any cylinders operating lean of the average and extra carbon monoxide by any cylinders rich of the average, so that the exhaust gas will have higher levels of O_2 and CO (and a lower level of CO_2) relative to an engine operating with identical fuel/air ratios in each cylinder.

Eltinge has developed a method for defining this nonuniformity in the fuel/air ratio distribution for spark-ignition engines which operate close to stoichiometric.[29] A function $f(x)$ for the fuel/air ratio distribution ($x = F/A$) was assumed, with standard deviation S_x. For each value of x (i.e., F/A), complete utilization of the available oxygen was assumed (i.e., no exhaust HC) and the CO_2, H_2O, CO, H_2 concentrations were related by Eq. (4.63) (with $K = 3.5$). The concentrations of each species for each (F/A) were weighted by the distribution function $f(x)$ and summed to produce the average exhaust concentration. (A correction to allow for the presence of unburned HC in the exhaust was also developed.) Figure 4-24 shows one set of results, for a fuel H/C ratio of 1.8 (typical of gasoline), for a normal distribution in the fuel/air ratio. For each mean (F/A) and maldistribution parameter S_x (the standard deviation of the F/A distribution) the corresponding dry concentrations of CO_2, CO, and O_2 are shown. This type of information can be used to define the fuel-air mixture nonuniformities in spark-ignition engines operating relatively close to stoichiometric. For diesel engines the variations of major exhaust gas species concentrations with fuel/air equiva-

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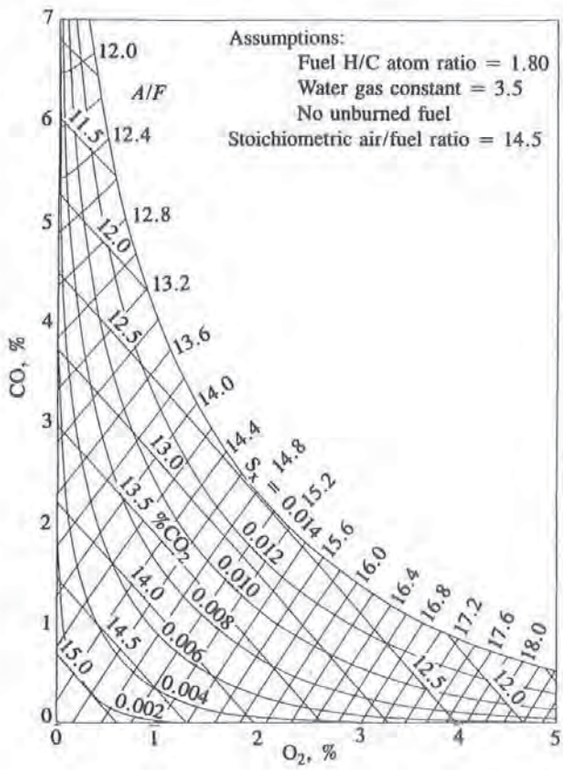


FIGURE 4-24
Computed relationship between dry exhaust gas composition (CO_2, CO, and O_2), air/fuel ratio, and maldistribution parameter S_x. Fuel: $(CH_{1.8})_n$. The correction to be added to the burned gas (F/A) which allows for the measured unburned hydrocarbon concentration is $4.7 \times 10^{-7} \times$ HC (ppm C_1) (*From Eltinge.*[29])

lence ratio are linear, so the effects of any nonuniformities are not apparent in this manner (see Fig. 4-22).

4.9.4 Combustion Inefficiency

Internal combustion engine exhaust contains combustible species: CO, H_2, unburned hydrocarbons, and particulates. When their concentrations are known, the combustion efficiency η_c given by Eq. (3.27) can be calculated. The chemical energy carried out of the engine in these combustibles represents the combustion inefficiency, $1 - \eta_c$:

$$1 - \eta_c = \frac{\sum_i x_i Q_{\mathrm{HV}_i}}{[\dot{m}_f/(\dot{m}_a + \dot{m}_f)]Q_{\mathrm{HV}_f}} \tag{4.69}$$

where the x_i are the mass fractions of CO, H_2, HC, and particulates, respectively, the Q_{HV_i} are the lower heating values of these species, and the subscripts f and a denote fuel and air. The heating values for CO (10.1 MJ/kg) and H_2(120 MJ/kg) are well defined. The composition of the unburned HC is not usually known. However, the heating values of hydrocarbons are closely comparable, so the fuel heating value (typically 42 to 44 MJ/kg) is used. The particulates (only present in diesels) are soot with some adsorbed hydrocarbons; usually the mass fraction is low enough for their contribution to be small, and a heating value for solid carbon (32.8 MJ/kg) can be used. Combustion efficiency data as a function of equivalence ratio have already been presented in Fig. 3-9.

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PROBLEMS

4.1. (*a*) Calculate the low-temperature burned gas composition resulting from the combustion of 7 g/s air with 0.48 g/s ethane (C_2H_6). Assume K = 3.5 in Eq. (4.6).
(*b*) Calculate the low-temperature burned gas composition for the combustion of 7 g/s air with 0.48 g/s ethanol (C_2H_6O). Assume K = 3.5 in Eq. (4.6). (Note the large difference in burned gas composition due to this difference in fuel.)

4.2. To evaluate the accuracy of the simple analytic ideal gas model, use the results of Example 4.1 and Eqs. (4.23) (constant-volume adiabatic combustion) and (4.24) (constant-pressure adiabatic combustion) to calculate T_b for a stoichiometric isooctane-air mixture. Compare this result with that obtained using Figs. 4-3 and 4-8. Assume the following unburned mixture conditions: $T = 700$ K, $v = 0.125$ m^3/kg air, $p = 15$ atm, and $x_b = 0.1$.

4.3. (*a*) In Fig. 4-1, why does M_u decrease as x_b increases?
(*b*) In Fig. 4-14, why does M_b decrease as T increases?
(*c*) In Fig. 4-14, why does M_b decrease as ϕ increases?

4.4. Show how, for an ideal gas with fixed composition, the molecular weight M is related to the specific heats c_p and c_v, and $\tilde{R}$. Use Figs. 4-15 and 4-16 to calculate M_b for $T_b = 1750$, 2250, and 2750 K, and $\phi = 1.0$. Compare these results with values of M_b obtained from Fig. 4-14 and explain any differences.

4.5. EGR, exhaust gas recirculation, is often used to reduce NO_x by acting as a diluent in the intake mixture.
(*a*) For low-temperature isooctane-air combustion products at $\phi = 1.0$, determine the percentage of the burned gases' average specific heat at constant pressure which comes from each component in the burned gas mixture.
(*b*) Compare the specific heat at constant pressure of isooctane-air combustion products at $\phi = 1.0$ to that of air, both at 1750 K. This difference is one reason why EGR instead of leaner fuel-air mixture is used to control NO_x emissions.

4.6. Explain qualitatively the causes of the trends in the curves in Fig. 4-23 as ϕ is increased from 1.0 and decreased from 1.0.

4.7. Compare the O_2, CO_2, and CO data in Fig. 4-22 with the predictions of the elemental balance in Table 4.3. Explain any differences. Assume diesel fuel is a hydrocarbon with H/C ratio of 2.

4.8. The following exhaust data were obtained from a four-stroke cycle spark-ignition engine. CO, CO_2, and NO_x molar concentrations are all measured fully dry; HC is measured fully wet as ppm C_1. Determine the fuel/air equivalence ratio ϕ for the following three sets of data. Make the following assumptions: (1) the constant K in Eq. (4.63) equals 3.5; (2) the fuel composition is $C_8H_{15.12}$; (3) the unburned hydrocarbon H/C ratio is the same as that of the fuel; and (4) NO_x is entirely NO.
(*a*) CO_2 14.0%, CO 0.64%, O_2 0.7%, NO_x 3600 ppm, HC 3200 ppm.
(*b*) CO_2 13.8%, CO 3.05%, O_2 0%, NO_x 1600 ppm, HC 3450 ppm.
(*c*) CO_2 12.5%, CO 0.16%, O_2 4.0%, NO_x 4600 ppm, HC 2100 ppm.

4.9. The exhaust from a spark-ignition engine has the following composition (in mole fractions):

$$CO_2 = 0.12;\ H_2O = 0.14;\ CO = 0.01;\ H_2 = 0.005;$$

$$N_2 = 0.7247;\ C_8H_{18} = 0.0003$$

The fuel is isooctane, C_8H_{18}; as shown above, a small fraction of the fuel escapes

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from the cylinder unburned. The lower heating value of isooctane is 44.4 MJ/kg, of carbon monoxide is 10.1 MJ/kg, and of hydrogen is 120 MJ/kg. The atomic weights of the elements are: C = 12, O = 16, H = 1, N = 14.

(*a*) Calculate the combustion *in*efficiency in the engine; i.e., the percentage of the entering fuel enthalpy which is not fully released in the combustion process and leaves the engine in the exhaust gases (for this problem, the exhaust can be assumed to be at room temperature).

(*b*) What fraction of this inefficiency is due to the unburned fuel emissions?

4.10. A 2-liter displacement four-cylinder engine, operating at 2000 rev/min and 30 percent of its maximum power at that speed, has the following exhaust composition (in percent by volume or mole percent):

CO_2, 11%; H_2O, 11.5%; CO, 0.5%; H_2, 0%; O_2, 2%; unburned hydrocarbons (expressed as CH_2, i.e., with a molecular weight of 14), 0.5%; N_2, 74.5%

The fuel is $(CH_2)_n$ with a heating value of 44 MJ/kg. The atomic weights of the elements are C = 12, H = 1, O = 16, N = 14. The heating values of CO and HC are 10 and 44 MJ/kg, respectively.

(*a*) Is the engine a diesel or spark-ignition engine? Is there enough oxygen in the exhaust to burn the fuel completely? Briefly explain.

(*b*) Calculate the fraction of the input fuel energy ($m_f Q_{HV}$) which exits the engine unburned as (1) CO and (2) unburned HC.

(*c*) An inventor claims a combustion efficiency of 100 percent can be achieved. What percentage improvement in engine specific fuel consumption would result?

4.11. A gasoline engine operates steadily on a mixture of isooctane and air. The air and fuel enter the engine at 25°C. The fuel consumption is 3.0 g/s. The output of the engine is 50 kW. The temperature of the combustion products in the exhaust manifold is 660 K. At this temperature, an analysis of the combustion products yields the following values (on a dry volumetric basis):

CO_2, 11.4%; O_2, 1.6%; CO, 2.9%; N_2, 84.1%

(*a*) Find the composition in moles (number of moles per mole of isooctane) of the reactants and the reaction products.

(*b*) Determine the heat-transfer rate from the working fluid as the working fluid passes through the engine.

Constants for the calculations:

	Enthalpy of formation, kJ/kmol	Sensible enthalpy at 660 K, kJ/kmol
C_8H_{18}	−259280	—
CO_2	−393522	15823
CO	−110529	10789
H_2O	−241827	12710
O_2	—	11200
N_2	—	10749

4.12. A direct-injection four-stroke cycle diesel engine is used to provide power for pumping water. The engine operates at its maximum rated power at 2000 rev/min,

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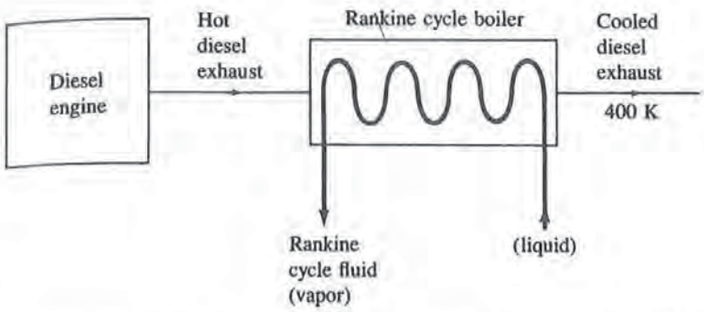


FIGURE P4-12

with an equivalence ratio of 0.8 and an air flow of 0.5 kg/s. The gross indicated fuel conversion efficiency is 45 percent, and the heat losses from the working fluid to the engine coolant and elsewhere within the engine are 280 kW. Diesel fuel has a heating value of 42 MJ/kg and stoichiometric fuel/air ratio of 0.067. Fuel and air enter the engine at ambient conditions. The mechanical efficiency of the diesel engine is 85 percent.

(*a*) Calculate the rated brake power of the engine, the average sensible enthalpy of the exhaust gases as they leave the engine, and the average exhaust gas temperature.

(*b*) Since the exhaust gas temperature is significantly above ambient, the advantages of using the diesel exhaust gas stream to heat the boiler of a Rankine cycle (see sketch) and generate additional power are to be explored. If the exhaust gases *leave* the Rankine-cycle system boiler at 400 K and 30 percent of the heat transferred from the exhaust gas stream in the boiler is converted to power at the Rankine-cycle power plant drive shaft, calculate the additional power obtained and the brake fuel conversion efficiency of the combined cycle system (diesel plus Rankine cycle).

4.13. A diesel engine has a compression ratio of 22 : 1. The conditions in the cylinder at the start of compression are $p = 101.3$ kPa and $T = 325$ K. Calculate the pressure and temperature at the end of compression, assuming the compression process is isentropic:

(*a*) Assume the cylinder contains an ideal gas with $\gamma = 1.4$ and $R = 287$ J/kg·K.

(*b*) Assume the cylinder contains air which may be regarded as a semiperfect gas (use the gas tables).

(*c*) Compare the work of compression in (*a*) and (*b*) above.

In practice, heat losses reduce the final compression temperature by 100 K. For a diesel engine operating at an equivalence ratio of 0.75 (full load):

(*d*) Calculate the ratio of heat loss during compression to the fuel energy added per cycle.

4.14. While the geometric compression ratio of an engine is V_{max}/V_{min}, the actual compression process starts somewhere between bottom-center and when the inlet valve closes, and it is conditions at time of spark (for an SI engine) or fuel injection (for a CI engine) that determine ignition. At low engine speed, compression starts about the time when the inlet valve closes. With this assumption, for the diesel engine of Prob. 4.13, calculate the air pressure and temperature at the start of injection. The inlet valve closes at 30° after BC; injection commences 15° before TC. Use the gas tables. Compare your answers with those of Prob. 4.13(*b*).

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4.15. Use an equilibrium computer code (which calculates the composition and properties of chemically reacting gas mixtures in equilibrium) to calculate the data you need for the following graphs:

(*a*) Values of c_p, γ, molecular weight, and gas composition (mole fractions of N_2, CO_2, H_2O, CO, H_2, O_2, OH, O, H, and NO) as a function of the equivalence ratio ($\phi = 0.2$ to 1.4) for products of combustion of isooctane (C_8H_{18}) with air at $p = 40$ atm and $T = 2500$ K. Put all species concentrations on the same graph. Use a log scale for the composition axis.

(*b*) Unburned mixture consisting of isooctane vapor and air at 700 K and 20 atm is burned first at constant pressure and then at constant volume.

(1) Calculate the enthalpy and internal energy of isooctane vapor at 700 K in cal/gmol; also calculate the volume per unit mass of mixture (cm^3/g) for $\phi = 0.2, 0.4, 0.6, 0.8, 1.0, 1.2, 1.4$.

(2) Use these data and the equilibrium program to calculate the temperature attained after combustion at constant pressure, and temperature and pressure attained after combustion at constant volume. Plot these temperatures and pressures against the equivalence ratio ϕ.

Thermodynamic data for isooctane vapor

T, K	$\tilde{c}_p$, cal/mol·K	$\tilde{h} - \tilde{h}_{298}$, kcal/mol	$\Delta\tilde{h}_f^\circ$, kcal/mol
298	45.14	0.00	−53.57
700	85.66	27.02	−62.79

4.16. A heavy wall bomb with a volume of 1000 cm^3 contains a mixture of isooctane with the stoichiometric air requirement at $p = 101.3$ kPa and $T = 25°C$. The mixture is then ignited with a spark. Find the pressure and temperature of the equilibrium combustion products just after combustion is complete (i.e., before heat losses to the wall are significant). Assume the burned gases are uniform.

4.17. A gas engine, running on a gaseous mixture of butane, C_4H_{10}, and air has the following conditions in the cylinder prior to constant-volume adiabatic combustion: pressure, 6.48×10^5 N/m^2; temperature, 600 K. The charge composition is air plus 50 percent of the stoichiometric quantity of butane fuel. Calculate the pressure and temperature at the end of combustion using the data given below.

For air	T, K	$\tilde{u}$, J/gmol
	298	6,161
	600	12,596
For butane	T, K	$\tilde{u}$, J/gmol
	298	−77
	600	38,424

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Internal energy of combustion of butane at 298 K is $\Delta\tilde{u} = -2.659$ MJ/gmol. Extract from gas tables for products of combustion for 50 percent stoichiometric fuel:

T, K	$\tilde{u}$, J/gmol
298	6,293
2075	54,227
2080	54,380

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CHAPTER 5

IDEAL MODELS OF ENGINE CYCLES

5.1 INTRODUCTION

The operating cycle of an internal combustion engine can be broken down into a sequence of separate processes: intake, compression, combustion, expansion, and exhaust. With models for each of these processes, a simulation of a complete engine cycle can be built up which can be analysed to provide information on engine performance. Models of individual engine processes at various levels of approximation have been developed. In this chapter we consider the simplest set of models which provide useful insights into the performance and efficiency of engines. The cycles analysed are commonly called the constant-volume, constant-pressure, and limited-pressure cycles; each title describes the approximation made for the engine combustion process.† The description of more accurate simulations of engine processes is deferred until Chap. 14.

For each engine cycle, a choice of models for working fluid thermodynamic properties must be made. These models have been reviewed in Chap. 4. Ideal

† These cycles are also referred to by the titles Otto cycle, Diesel cycle, and dual cycle, respectively, for historical reasons. The more descriptive titles used above are preferred because they avoid the often-made assumption that the SI or Otto engine is best approximated by the constant-volume cycle, and the CI or diesel engine is best approximated by the constant-pressure cycle. These assumptions are not necessarily correct.

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engine cycle models combined with a simple ideal gas model (specific heats constant throughout the engine cycle—model 1 in Table 4.2) provide analytic results and are useful for illustrative purposes; we will call these cycles *ideal gas standard cycles*. Ideal engine cycles combined with more realistic models of working fluid properties (a frozen mixture of ideal gases for the unburned mixture and an equilibrium mixture for the burned mixture—model 5 in Table 4.2) are called *fuel-air cycles* and provide more quantitative information on engine operation.

An internal combustion engine is not a heat engine in the thermodynamic definition of the term. It is not a closed system. The working fluid does not execute a thermodynamic cycle. The temperature changes which occur around minimum and maximum cylinder volumes are not primarily a result of heat-transfer interactions. An engine can best be analyzed as an open system which exchanges heat and work with its surrounding environment (the atmosphere). Reactants (fuel and air) flow into the system; products (exhaust gases) flow out. (An overall second law analysis of the engine from this point of view has already been presented in Sec. 3.6.) Thus, the cycles discussed in this chapter are not thermodynamic cycles. Rather, each is a consecutive sequence of processes through which we can follow the state of the working fluid as the engine executes a complete operating cycle.

5.2 IDEAL MODELS OF ENGINE PROCESSES

The sequence of processes which make up a typical SI and CI engine operating cycle has been described in Sec. 1.3. To illustrate these processes, cylinder pressure (p) and volume (V) data from a throttled four-stroke cycle SI engine are plotted as a p-V diagram in Fig. 5-1. The cycle can be divided into compression,

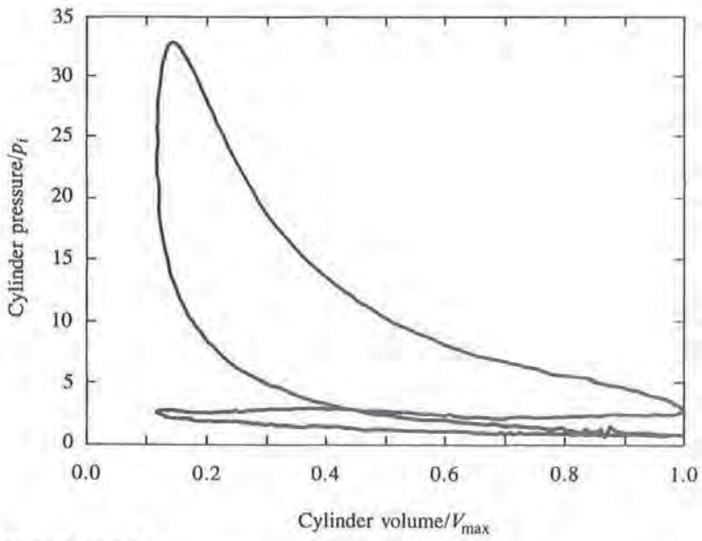


FIGURE 5-1
Pressure-volume diagram of firing spark-ignition engine. $r_c = 8.4$, 3500 rev/min, $p_i = 0.4$ atm, $p_e = 1$ atm, $\text{imep}_n = 2.9$ atm.

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TABLE 5.1
Ideal models of engine processes

Process	Assumptions
Compression (1-2)	1. Adiabatic and reversible (hence isentropic)
Combustion (2-3)	1. Adiabatic 2. Combustion occurs at (*a*) Constant volume (*b*) Constant pressure (*c*) Part at constant volume and part at constant pressure (called limited pressure) 3. Combustion is complete ($\eta_c = 1$)
Expansion (3-4)	1. Adiabatic and reversible (hence isentropic)
Exhaust (4-5-6) and intake (6-7-1)	1. Adiabatic 2. Valve events occur at top- and bottom-center 3. No change in cylinder volume as pressure differences across open valves drop to zero 4. Inlet and exhaust pressures constant 5. Velocity effects negligible

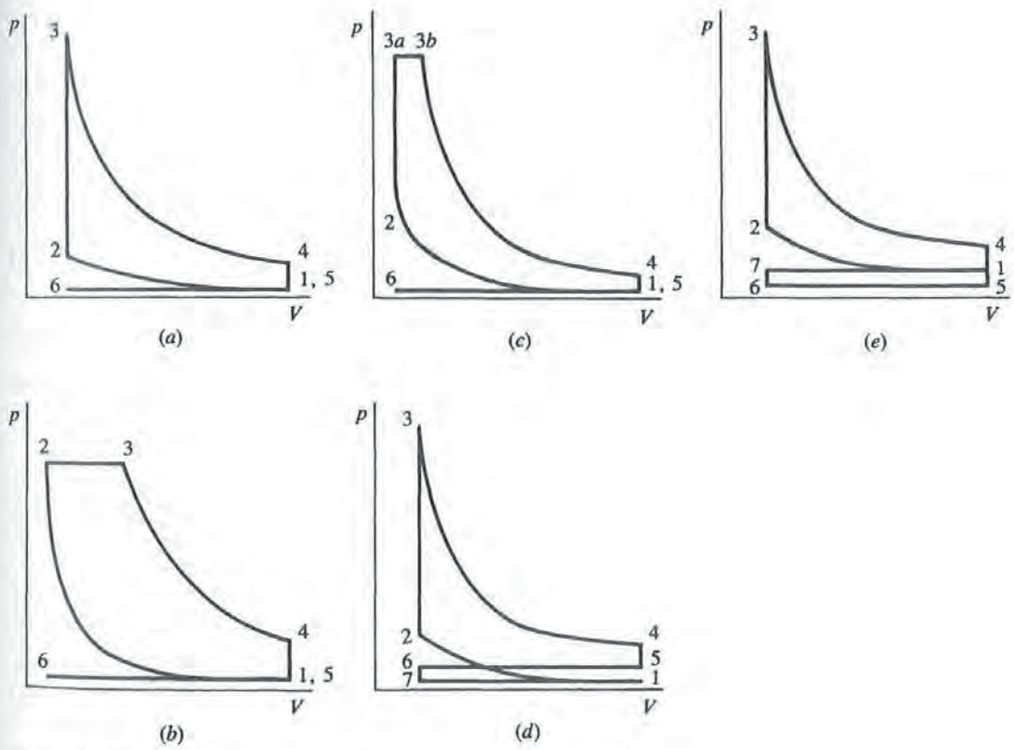


FIGURE 5-2
Pressure-volume diagrams of ideal cycles. Unthrottled operation: (*a*) constant-volume combustion; (*b*) constant-pressure combustion; (*c*) limited-pressure combustion. (*d*) Throttled constant-volume cycle; (*e*) supercharged constant-volume cycle.

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combustion, expansion, exhaust, and intake processes. Sets of assumptions which simplify each of these processes to a form convenient for analysis are given in Table 5.1.

Pressure-volume diagrams for the constant-volume, constant-pressure, and limited-pressure cycles for unthrottled engine operation are illustrated in Fig. 5-2*a* to *c*. Throttled engine operation ($p_i < p_e$) and supercharged engine operation ($p_i > p_e$) are shown in Fig. 5-2*d* and *e*. In each cycle 1-2 is the compression process, 2-3 is the combustion process, 3-4 (or 2-4 in the constant-pressure cycle) is the expansion process, 4-5-6 is the exhaust process, and 6-7-1 is the intake process.

The most critical assumption in determining how useful these ideal cycles are as indicators of engine performance is the form assumed for the combustion process. The real engine combustion process occupies a finite crank angle period (between about 20 and 70 crank angle degrees), and the spark or fuel-injection timing may be retarded from its optimum advance to closer to TC. The constant-volume cycle is the limiting case of infinitely fast combustion at TC; the constant-pressure cycle would correspond to slow and late combustion; the limited-pressure cycle lies in between.

5.3 THERMODYNAMIC RELATIONS FOR ENGINE PROCESSES

The overall engine operating parameters of greatest interest which can be determined from a thermodynamic analysis of the engine operating cycle are:

The indicated fuel conversion efficiency $\eta_{f,i}$:

$$\eta_{f,i} = \frac{W_{c,i}}{m_f Q_{\text{LHV}}} \tag{5.1}$$

(which, since the combustion efficiency is unity, is equal to the indicated thermal conversion efficiency $\eta_{t,i}$; see Sec. 3.6.2)

The indicated mean effective pressure (imep):

$$\text{imep} = \frac{W_{c,i}}{V_d} = \frac{m_f Q_{\text{LHV}} \eta_{f,i}}{V_d} \tag{5.2}$$

$W_{c,i}$, the indicated work per cycle, is the sum of the compression stroke work and the expansion stroke work:

$$W_{c,i} = W_C + W_E \tag{5.3}$$

Using the notation of Fig. 5-2 to define the endpoints of each engine process, the following relationships are obtained by applying the first and second laws of thermodynamics to the cylinder contents:

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Compression stroke:

$$\frac{v_1}{v_2} = r_c \tag{5.4}$$

Since the process is adiabatic and reversible

$$s_2 = s_1 \tag{5.5}$$

The compression work is

$$W_C = U_1 - U_2 = m(u_1 - u_2) \tag{5.6}$$

Combustion process:
For the constant-volume cycle,

$$v_3 = v_2 \qquad u_3 - u_2 = 0 \tag{5.7a, b}$$

For the constant-pressure cycle,

$$p_3 = p_2 \qquad h_3 - h_2 = 0 \tag{5.7c, d}$$

For the limited-pressure cycle,

$$v_{3a} = v_2 \qquad p_{3b} = p_{3a} \tag{5.7e, f}$$

and

$$u_{3a} - u_2 = 0 \qquad h_{3b} - h_{3a} = 0 \tag{5.7g, h}$$

Expansion stroke:
For the constant-volume cycle,

$$\frac{v_4}{v_3} = r_c \qquad s_4 = s_3 \tag{5.8a, b}$$

and the expansion work is

$$W_E = U_3 - U_4 = m(u_3 - u_4) \tag{5.9}$$

For the constant-pressure cycle,

$$p_3 = p_2 \qquad \frac{v_4}{v_2} = r_c \qquad s_4 = s_3 \tag{5.10a, b, c}$$

and the expansion work is

$$\begin{aligned} W_E &= U_3 - U_4 + p_2(V_3 - V_2) \\ &= m[(u_3 - u_4) + p_2(v_3 - v_2)] \\ &= m[(h_3 - h_4) + p_4 v_4 - p_2 v_2] \end{aligned} \tag{5.11}$$

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For the limited-pressure cycle,

$$v_4/v_{3a} = r_c \qquad p_{3b} = p_{3a} \qquad s_4 = s_{3b} \tag{5.12a, b, c}$$

and the expansion work is

$$\begin{aligned} W_E &= U_{3b} - U_4 + p_3(V_{3b} - V_{3a}) \\ &= m[(u_{3b} - u_4) + p_3(v_{3b} - v_{3a})] \\ &= m[(h_{3b} - h_4) + p_4 v_4 - p_3 v_{3a}] \end{aligned} \tag{5.13}$$

The indicated fuel conversion efficiency is found by substitution into Eqs. (5.3) and (5.1):

For the constant-volume cycle:

$$\eta_{f,i} = \frac{m[(u_3 - u_4) - (u_2 - u_1)]}{m_f Q_{\text{LHV}}} \tag{5.14}$$

For the constant-pressure cycle:

$$\eta_{f,i} = \frac{m[(h_3 - h_4) - (u_2 - u_1) + p_4 v_4 - p_2 v_2]}{m_f Q_{\text{LHV}}} \tag{5.15}$$

For the limited-pressure cycle:

$$\eta_{f,i} = \frac{m[(h_{3b} - h_4) - (u_2 - u_1) + p_4 v_4 - p_3 v_{3a}]}{m_f Q_{\text{LHV}}} \tag{5.16}$$

The state of the mixture at point 1 in the cycle depends on the intake mixture properties and the residual gas properties at the end of the exhaust stroke.

When the exhaust valve opens at point 4, the cylinder pressure is above the exhaust manifold pressure and a *blowdown* process occurs. In the ideal exhaust process model, this blowdown occurs with the piston stationary at BC. During this blowdown process, the gas which remains inside the cylinder expands isentropically. The gases escaping from the cylinder undergo an unrestrained expansion or throttling process which is irreversible. It is assumed that the kinetic energy acquired by each gas element as it is accelerated through the exhaust valve is dissipated in a turbulent mixing process in the exhaust port into internal energy and flow work. Since it is also assumed that no heat transfer occurs, the enthalpy of each element of gas after it leaves the cylinder remains constant.

These processes are illustrated on an h-s diagram in Fig. 5-3. The gas remaining in the cylinder expands isentropically along the line 4-5. The first element of gas which leaves the cylinder at point 4 enters the exhaust manifold at state a on the pressure $= p_e$ line. An element that leaves the cylinder at an intermediate state b on the expansion line 4-5 would enter the exhaust manifold at state c. At the end of the blowdown process the gas in the cylinder and the last

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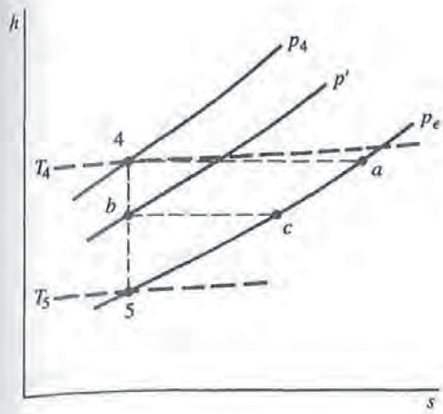


FIGURE 5-3
Enthalpy-entropy diagram of gas state during exhaust process. See text for explanation.

gas to leave have the same state—5. There is, therefore, a gradient in temperature within the exhausted gas. The temperature of the first element exhausted, T_a, is slightly less than T_4; the temperature of the last element exhausted is T_5.

A displacement of gas out of the cylinder follows the blowdown process as the piston moves from BC to TC. If heat-transfer and kinetic energy dissipation effects are neglected, no change in thermodynamic state of the gas occurs. In this *displacement* process, the mass of gas within the cylinder at the end of the blowdown process is further decreased by the ratio V_5/V_6.

The mass of residual gas m_r in the cylinder at point 6 in the cycle is obtained by first determining the state of the gas (T_5, v_5) at the end of the blowdown process following an isentropic expansion from p_4 to p_e and then by reducing the cylinder volume to the clearance volume V_6. The residual mass fraction is thus given by

$$\frac{m_r}{m} = x_r = \frac{v_4/v_5}{r_c} = \frac{v_2}{v_5} \tag{5.17}$$

The average state of the exhausted gas can be determined by considering the open system defined by the piston face, cylinder walls, and cylinder head, shown in Fig. 5-4. Applying the first law of thermodynamics for an open system gives

$$U_6 - U_4 = p_e(V_4 - V_6) - H_e \tag{5.18a}$$

where H_e is the enthalpy of the mass of gas exhausted from the cylinder. The average specific exhaust enthalpy is, therefore,

$$\bar{h}_e = \frac{m_4 u_4 - m_6 u_6 + p_e V_d}{m_4 - m_6} \tag{5.18b}$$

which, with $p = p_e$, defines the average exhausted-gas state.

The mixture temperature at the end of the intake stroke and at the start of the compression stroke (point 1 in Fig. 5-2) can now be determined, again using

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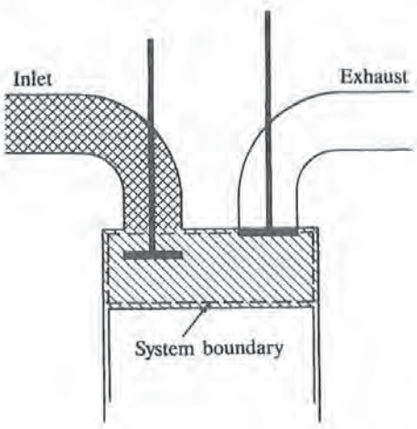


FIGURE 5-4
Definition of system boundary for thermodynamic analysis of ideal cycle processes.

the open system in Fig. 5-4. Application of the first law between points 6 and 1 gives

$$U_1 - U_6 = -p_i(V_1 - V_6) + (m_1 - m_6)h_i \tag{5.19a}$$

or

$$m_1u_1 - m_6u_6 = -p_i(V_1 - V_6) + (m_1 - m_6)h_i \tag{5.19b}$$

or

$$m_1h_1 = m_6h_6 + (m_1 - m_6)h_i + V_2(p_i - p_e) \tag{5.19c}$$

where h_i is the specific enthalpy of the inlet mixture and $p_1 = p_i$.

Note that when $p_i < p_e$, part of the residual gas in the cylinder at the end of the exhaust stroke will flow into the intake system when the intake valve opens. This flow will cease when the cylinder pressure equals p_i. However, provided no heat transfer occurs, this backflow will not affect Eqs. (5.19) above, since the flow of residual through the intake valve is a constant enthalpy process.

In many engines, the closing of the exhaust valve and the opening of the intake valve overlap. Flow of exhausted gases from the exhaust system through the cylinder into the intake system can then occur. Equations (5.18) and (5.19) would have to be modified to account for valve overlap.

In the four-stroke engine cycle, work is done on the piston during the intake and the exhaust processes. The work done by the cylinder gases on the piston during exhaust is

$$W_e = p_e(V_2 - V_1) \tag{5.20}$$

The work done by the cylinder gases on the piston during intake is

$$W_i = p_i(V_1 - V_2) \tag{5.21}$$

The net work to the piston over the exhaust and intake strokes, the *pumping work*, is

$$W_p = (p_i - p_e)(V_1 - V_2) \tag{5.22}$$

which, for the cylinder gas system, is negative for $p_i < p_e$ and positive for $p_i > p_e$.

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The pumping mean effective pressure (pmep) is usually defined as a positive quantity. Thus:

For $p_i < p_e$: $$\text{pmep} = p_e - p_i \tag{5.23a}$$

For $p_i > p_e$: $$\text{pmep} = p_i - p_e \tag{5.23b}$$

The net and gross indicated mean effective pressures are related by

$$\text{imep}_n = \text{imep}_g - (p_e - p_i) \tag{5.24}$$

The net indicated fuel conversion efficiency is related to the gross indicated fuel conversion efficiency by

$$\eta_{f,\text{in}} = \eta_{f,\text{ig}}\left(1 - \frac{p_e - p_i}{\text{imep}_g}\right) \tag{5.25}$$

5.4 CYCLE ANALYSIS WITH IDEAL GAS WORKING FLUID WITH c_v AND c_p CONSTANT

If the working fluid in these ideal cycles is assumed to be an ideal gas, with c_v and c_p constant throughout the engine operating cycle, the equations developed in the previous section which describe engine performance and efficiency can be further simplified. We will use the notation of Fig. 5-2.

5.4.1 Constant-Volume Cycle

The compression work (Eq. 5.6) becomes

$$W_C = mc_v(T_1 - T_2) \tag{5.26}$$

The expansion work (Eq. 5.9) becomes

$$W_E = mc_v(T_3 - T_4) \tag{5.27}$$

The denominator in Eq. (5.14), $m_f Q_{\text{LHV}}$, can be related to the temperature rise during combustion. For the working fluid model under consideration, the $U(T)$ lines for the reactants and products on a U-T diagram such as Fig. 3-5 are parallel and have equal slopes, of magnitude c_v. Hence, for a constant-volume adiabatic combustion process

$$mc_v(T_3 - T_2) = m_f Q_{\text{LHV}} \tag{5.28}\dagger$$

† Note that if insufficient air is available for complete combustion of the fuel, Eq. (5.28) must be modified. The right-hand side of the equation should then be $\eta_c m_f Q_{\text{LHV}}$, where η_c is the combustion efficiency given by Eq. (3.27).

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Note that the heating values at constant volume and constant pressure are the same for this working fluid. For convenience we will define

$$Q^* = \frac{m_f Q_{\text{LHV}}}{m} \tag{5.29}$$

Q^* is the specific internal energy (and enthalpy) decrease, during isothermal combustion, per unit mass of working fluid.

The relation for indicated fuel conversion efficiency (Eq. 5.14) becomes

$$\eta_{f,i} = \frac{(T_3 - T_4) - (T_2 - T_1)}{T_3 - T_2} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{5.30}$$

Since 1-2 and 3-4 are isentropic processes between the same volumes, V_1 and V_2,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r_c^{\gamma-1} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \frac{T_3}{T_4}$$

where $\gamma = c_p/c_v$. Hence:

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

and Eq. (5.30) can be rearranged as

$$\eta_{f,i} = 1 - \frac{1}{r_c^{\gamma-1}} \tag{5.31}$$

Values of $\eta_{f,i}$ for different values of γ are shown in Fig. 5-5. The indicated fuel conversion efficiency increases with increasing compression ratio and decreases as γ decreases.

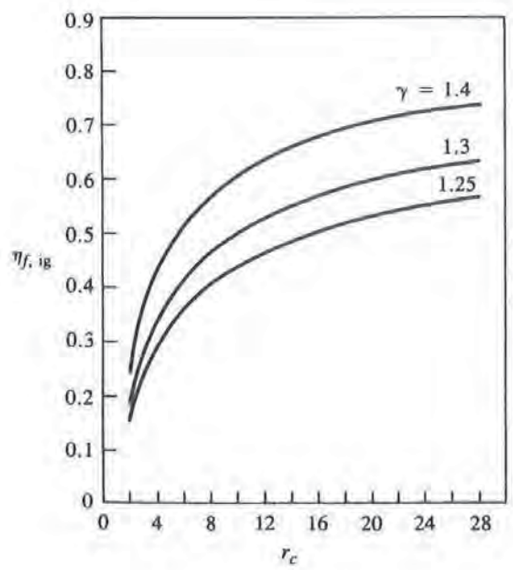


FIGURE 5-5
Ideal gas constant-volume cycle fuel conversion efficiency as a function of compression ratio; $\gamma = c_p/c_v$.

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The indicated mean effective pressure, using Eqs. (5.2) and (5.31), becomes

$$\frac{\text{imep}}{p_1} = \left(\frac{Q^*}{c_v T_1}\right)\left(\frac{1}{\gamma - 1}\right)\left(\frac{r_c}{r_c - 1}\right)\left(1 - \frac{1}{r_c^{\gamma-1}}\right) \tag{5.32}$$

The dimensionless numbers r_c, γ, and $Q^*/(c_v T_1)$ are sufficient to describe the characteristics of the constant-volume ideal gas standard cycle, relative to its initial conditions p_1, T_1.

It is useful to compare the imep—a measure of the effectiveness with which the displaced volume of the engine is used to produce work—and the maximum pressure in the cycle, p_3. The ratio p_3/p_1 can be determined from the ideal gas law applied at points 2 and 3, and the relation

$$\frac{T_3}{T_2} = 1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}} \tag{5.33}$$

obtained from Eq. (5.28). Equations (5.32) and (5.33) then give

$$\frac{\text{imep}}{p_3} = \frac{1}{(\gamma - 1)r_c^{\gamma}}\left(\frac{r_c}{r_c - 1}\right)\frac{1 - 1/r_c^{\gamma-1}}{c_v T_1/Q^* + 1/r_c^{\gamma-1}} \tag{5.34}$$

A high value of imep/p_3 is desirable. Engine weight will increase with increasing p_3 to withstand the increasing stresses in components.

The indicated fuel conversion efficiency and the ratios imep/p_1 and imep/p_3 for this ideal cycle model do not depend on whether the cycle is throttled or supercharged. However, the relationships between the working fluid properties at points 1 and 6 do depend on the degree of throttling or supercharging. For throttled engine operation, the residual gas mass fraction x_r can be determined as follows. From Eq. (5.17), since state 5 corresponds to an isentropic expansion from state 4 to $p = p_e$, x_r is given by

$$x_r = \frac{(p_e/p_4)^{1/\gamma}}{r_c} = \frac{(p_e/p_i)^{1/\gamma}(p_1/p_4)^{1/\gamma}}{r_c}$$

Since

$$\frac{p_1}{p_4} = \frac{p_1}{p_2}\frac{p_2}{p_3}\frac{p_3}{p_4} = \frac{1}{r_c^{\gamma}}\frac{T_2}{T_3}r_c^{\gamma} = \left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}}\right)^{-1}$$

it follows that

$$x_r = \frac{1}{r_c}\frac{(p_e/p_i)^{1/\gamma}}{[1 + Q^*/(c_v T_1 r_c^{\gamma-1})]^{1/\gamma}} \tag{5.35}$$

The residual mass fraction increases as p_i decreases below p_e, decreases as r_c increases, and decreases as $Q^*/(c_v T_1)$ increases.

Through a similar analysis, the temperature of the residual gas T_6 can be determined:

$$\frac{T_6}{T_1} = \left(\frac{p_e}{p_i}\right)^{(\gamma-1)/\gamma}\left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}}\right)^{1/\gamma} \tag{5.36}$$

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The mixture temperature at point 1 in the cycle can be related to the inlet mixture temperature, T_i, with Eq. (5.19). For a working fluid with c_v and c_p constant, this equation becomes

$$c_p T_1 = x_r c_p T_6 + (1 - x_r) c_p T_i - \frac{R T_1}{r_c}\left(\frac{p_e}{p_i} - 1\right) \tag{5.37}$$

Use of Eqs. (5.36) and (5.37) leads to the relation

$$\frac{T_1}{T_i} = \frac{1 - x_r}{1 - 1/(\gamma r_c)[p_e/p_i + (\gamma - 1)]} \tag{5.38}$$

Extensive results for the constant-volume cycle with $\gamma = 1.4$ can be found in Taylor.[1]

5.4.2 Limited- and Constant-Pressure Cycles

The constant-pressure cycle is a limited-pressure cycle with $p_3 = p_2$. For the limited-pressure cycle, the compression work remains

$$W_C = m c_v (T_1 - T_2) \tag{5.39}$$

The expansion work, from Eq. (5.13), becomes

$$W_E = m[c_v(T_{3b} - T_4) + p_3(v_{3b} - v_{3a})] \tag{5.40}$$

For the combustion process, Eqs. (5.7*g*, *h*) give

$$m_{f\,2-3a} Q_{\text{LHV}} = m c_v (T_{3a} - T_2) \tag{5.41a}$$

$$m_{f\,3a-3b} Q_{\text{LHV}} = m c_p (T_{3b} - T_{3a}) \tag{5.41b}$$

or

$$m_f Q_{\text{LHV}} = m[c_v(T_{3a} - T_2) + c_p(T_{3b} - T_{3a})] \tag{5.41c}$$

for a working fluid with c_v and c_p constant throughout the cycle.

Combining Eqs. (5.1), (5.3), and (5.39) to (5.41) and simplifying gives

$$\eta_{f,i} = 1 - \frac{T_4 - T_1}{(T_{3a} - T_2) + \gamma(T_{3b} - T_{3a})}$$

Use of the isentropic relationships for the working fluid along 1-2 and 3*b*-4, with the substitutions

$$\alpha = \frac{p_3}{p_2} \qquad \beta = \frac{V_{3b}}{V_{3a}} \tag{5.42a, b}$$

leads to the result

$$\eta_{f,i} = 1 - \frac{1}{r_c^{\gamma-1}}\left[\frac{\alpha\beta^\gamma - 1}{\alpha\gamma(\beta - 1) + \alpha - 1}\right] \tag{5.43}$$

For $\beta = 1$ this result becomes the constant-volume cycle efficiency (Eq. 5.31). For $\alpha = 1$, this result gives the constant-pressure cycle efficiency as a special case.

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The mean effective pressure is related to p_1 and p_3 via

$$\frac{\text{imep}}{p_1} = \frac{Q^*}{c_v T_1(\gamma - 1)}\left(\frac{r_c}{r_c - 1}\right)\eta_{f,i} \tag{5.44}$$

$$\frac{\text{imep}}{p_3} = \frac{1}{\alpha r_c^{\gamma}}\left(\frac{Q^*}{c_v T_1}\right)\left(\frac{1}{\gamma - 1}\right)\left(\frac{r_c}{r_c - 1}\right)\eta_{f,i} \tag{5.45}$$

5.4.3 Cycle Comparison

The above expressions are most useful if values for γ and $Q^*/(c_v T_1)$ are chosen to match real working fluid properties. Figure 5-5 has already shown the sensitivity of $\eta_{f,i}$ for the constant-volume cycle to the value of γ chosen. In Sec. 4.4, average values of γ_u and γ_b were determined which match real working fluid properties over the compression and expansion strokes, respectively. Values for a stoichiometric mixture appropriate to an SI engine are $\gamma_u \approx 1.3$, $\gamma_b \approx 1.2$. However, analysis of pressure-volume data for real engine cycles indicates that pV^n, where $n \approx 1.3$, is a good fit to the expansion stroke p-V data.[2] Heat transfer from the burned gases increases the exponent above the value corresponding to γ_b. A value of $\gamma = 1.3$ for the entire cycle is thus a reasonable compromise.

Q^*, defined by Eq. (5.29), is the enthalpy decrease during isothermal combustion per unit mass of working fluid. Hence

$$Q^* = \left(\frac{m_f}{m_a}\right)Q_{\text{LHV}}\left(\frac{m_a}{m}\right) \tag{5.46}$$

A simple approximation for (m_a/m) is $(r_c - 1)/r_c$; i.e., fresh air fills the displaced volume and the residual gas fills the clearance volume at the same density. Then, for isooctane fuel, for a stoichiometric mixture, Q^* is given by 2.92×10^6 $(r_c - 1)/r_c$ J/kg air. For $\gamma = 1.3$ and an average molecular weight $M = 29.3$, $c_v =$ 946 J/kg·K. For $T_1 = 333$ K, $Q^*/(c_v T_1)$ becomes 9.3 $(r_c - 1)/r_c$. For this value of $Q^*/(c_v T_1)$ all cycles would be burning a stoichiometric mixture with an appropriate residual gas fraction.

Pressure-volume diagrams for the three ideal cycles for the same compression ratio and unburned mixture composition are shown in Fig. 5-6. For each cycle, $\gamma = 1.3$, $r_c = 12$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c = 8.525$. Overall performance characteristics for each of these cycles are summarized in Table 5.2. The constant-volume cycle has the highest efficiency, the constant-pressure cycle the lowest efficiency. This can be seen from Eq. (5.43) where the term in square brackets is equal to unity for the constant-volume cycle and greater than unity for the limited- and constant-pressure cycles. The imep values are proportional to $\eta_{f,i}$ since the mass of fuel burned per cycle is the same in all three cases.

As the peak pressure p_3 is decreased, the ratio of imep to p_3 increases. This ratio is important because imep is a measure of the useful pressure on the piston, and the maximum pressure chiefly affects the strength required of the engine structure.

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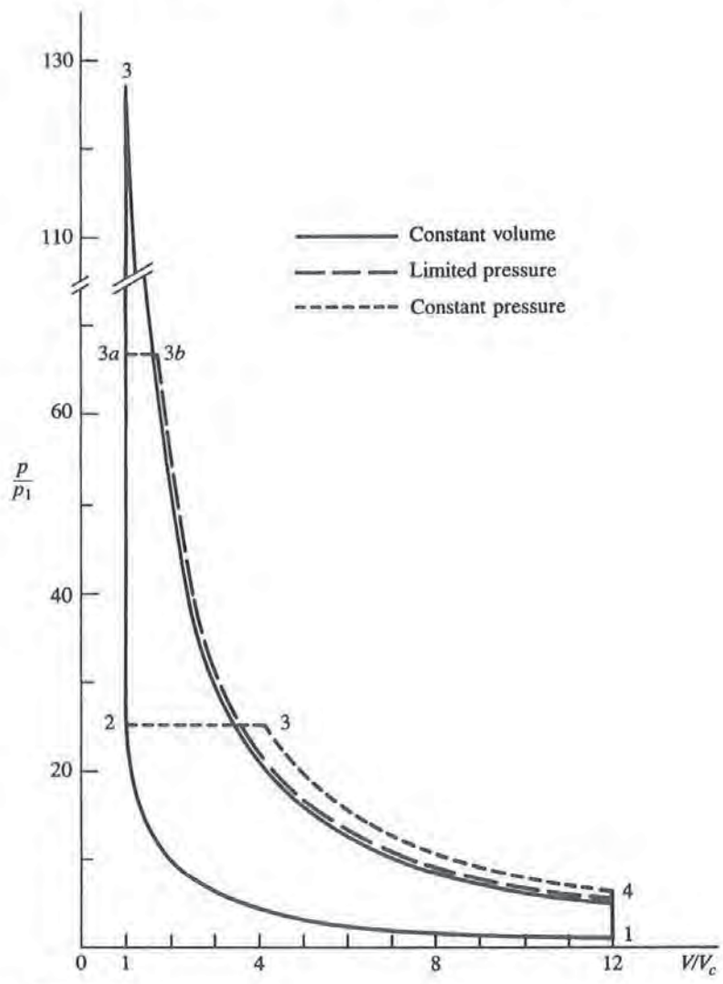


FIGURE 5-6
Pressure-volume diagrams for constant-volume, limited-pressure, and constant-pressure ideal gas standard cycles. $r_c = 12, \gamma = 1.3, Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c = 8.525, p_{3a}/p_1 = 67$.

TABLE 5.2
Comparison of ideal cycle results

	$\eta_{f,i}$	$\frac{\text{imep}}{p_1}$	$\frac{\text{imep}}{p_3}$	$\frac{p_{max}}{p_1}$
Constant volume	0.525	16.3	0.128	128
Limited pressure	0.500	15.5	0.231	67
Constant pressure	0.380	11.8	0.466	25.3

$\gamma = 1.3; r_c = 12; Q^*/(c_v T_1) = 8.525$.

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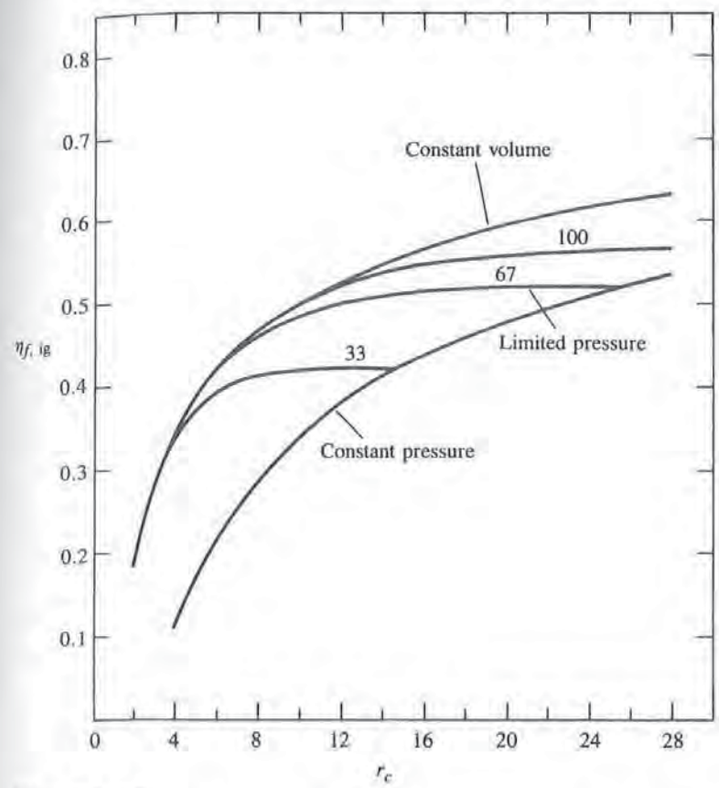


FIGURE 5-7
Fuel conversion efficiency as a function of compression ratio, for constant-volume, constant-pressure, and limited-pressure ideal gas cycles. $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. For limited-pressure cycle, $p_3/p_1 = 33, 67, 100$.

A more extensive comparison of the three cycles is given in Figs. 5-7 and 5-8, over a range of compression ratios. For all cases $\gamma = 1.3$ and $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. At any given r_c, the constant-volume cycle has the highest efficiency and lowest imep$/p_3$. For a given maximum pressure p_3, the constant-pressure cycle has the highest efficiency (and the highest compression ratio). For the limited-pressure cycle, at constant p_3/p_1, there is little improvement in efficiency and imep above a compression ratio of about 8 to 10 as r_c is increased.

Example 5.1 shows how ideal cycle equations relate residual and intake conditions with the gas state at point 1 in the cycle. An iterative procedure is required if intake conditions are specified.

Example 5.1. For $\gamma = 1.3$, compression ratio $r_c = 6$, and a stoichiometric mixture with intake temperature 300 K, find the residual gas fraction, residual gas temperature, and mixture temperature at point 1 in the constant-volume cycle for $p_e/p_i = 1$ (unthrottled operation) and 2 (throttled operation).

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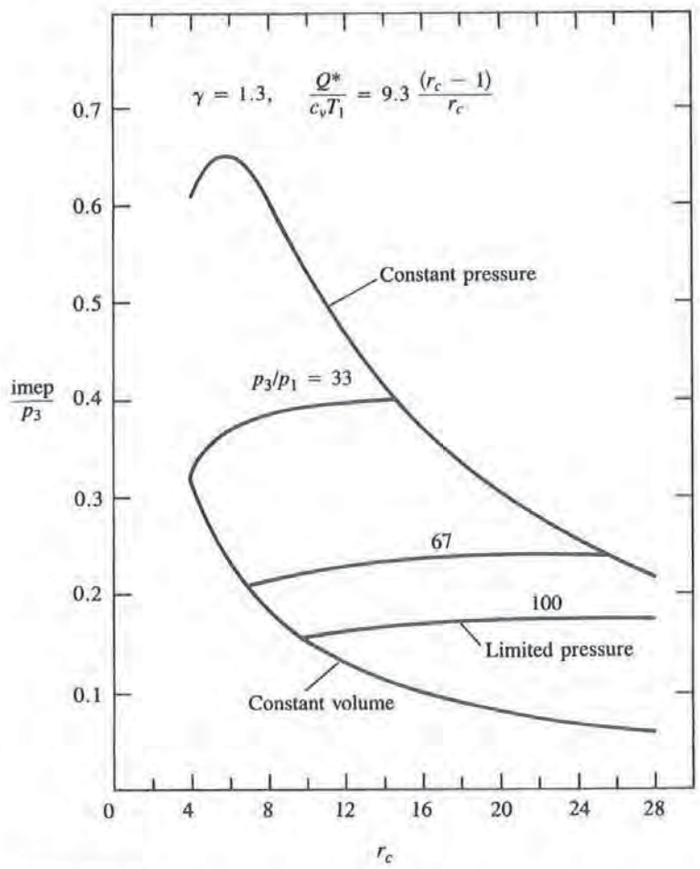


FIGURE 5-8
Indicated mean effective pressure (imep) divided by maximum cycle pressure (p_3) as a function of compression ratio for constant-volume, constant-pressure, and limited-pressure cycles. Details same as Fig. 5-7.

For a stoichiometric mixture, for isooctane,

$$Q^* = \frac{m_f}{m} Q_{\text{LHV}} = \left(\frac{m_f}{m_i}\right)\left(\frac{m_i}{m}\right) Q_{\text{LHV}} = \frac{44.38}{16.14}(1 - x_r) = 2.75(1 - x_r) \qquad \text{MJ/kg}$$

For $\gamma = 1.3$, $c_v = 946$ J/kg · K and

$$\frac{Q^*}{c_v T_1} = \frac{2.75 \times 10^6}{946 T_1}(1 - x_r) = \frac{2910}{T_1}(1 - x_r) \qquad (a)$$

Equations (5.35), (5.36), and (5.38), for $r_c = 6$ and $\gamma = 1.3$, become

$$x_r = \frac{1}{6} \frac{(p_e/p_i)^{0.769}}{[1 + Q^*/(c_v T_1 \times 6^{0.3})]^{0.769}} \qquad (b)$$

$$\frac{T_r}{T_1} = \left(\frac{p_e}{p_i}\right)^{0.23}\left(1 + \frac{Q^*}{c_v T_1 \times 6^{0.3}}\right)^{0.769} \qquad (c)$$

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$$\frac{T_1}{300} = \frac{1 - x_r}{1 - [1/(1.3 \times 6)](p_e/p_i + 0.3)} \qquad (d)$$

A trial-and-error solution of Eqs. (*a*) to (*d*) is required. It is easiest to estimate x_r, solve for T_1 from (*d*), evaluate $Q^*/(c_v T_1)$ from (*a*), and check the value of x_r assumed with that given by (*b*).

For $(p_e/p_i) = 1$ (unthrottled operation) the following solution is obtained:

$$x_r = 0.044, \qquad T_1 = 344 \text{ K}, \qquad \frac{Q^*}{c_v T_1} = 8.1, \qquad T_r = 1316 \text{ K}$$

For $(p_e/p_i) = 2$ the following solution is obtained:

$$x_r = 0.082, \qquad T_1 = 391 \text{ K}, \qquad \frac{Q^*}{c_v T_1} = 6.8, \qquad T_r = 1580 \text{ K}$$

5.5 FUEL-AIR CYCLE ANALYSIS

A more accurate representation of the properties of the working fluid inside the engine cylinder is to treat the unburned mixture as frozen in composition and the burned gas mixture as in equilibrium. Values for thermodynamic properties for these working fluid models can be obtained with the charts for unburned and burned gas mixtures described in Sec. 4.5, or the computer codes summarized in Sec. 4.7. When these working fluid models are combined with the ideal engine process models in Table 5.1, the resulting cycles are called fuel-air cycles.[1] The sequence of processes and assumptions are (with the notation of Fig. 5-2):

1-2 Reversible adiabatic compression of a mixture of air, fuel vapor, and residual gas without change in chemical composition.

2-3 Complete combustion (at constant volume or limited pressure or constant pressure), without heat loss, to burned gases in chemical equilibrium.

3-4 Reversible adiabatic expansion of the burned gases which remain in chemical equilibrium.

4-5-6 Ideal adiabatic exhaust blowdown and displacement processes with the burned gases fixed in chemical composition.

6-7-1 Ideal intake process with adiabatic mixing between residual gas and fresh mixture, both of which are fixed in chemical composition.

The basic equations for each of these processes have already been presented in Sec. 5.3. The use of the charts for a complete engine cycle calculation will now be illustrated.

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5.5.1 SI Engine Cycle Simulation

The mixture conditions at point 1 must be known or must be estimated. The following approximate relationships can be used for this purpose:[3]

$$x_r = \left\{1 + \frac{T_r}{T_i}\left[r_c\left(\frac{p_i}{p_e}\right) - \left(\frac{p_i}{p_e}\right)^{(\gamma-1)/\gamma}\right]\right\}^{-1} \tag{5.47}$$

$$T_1 = T_r r_c x_r\left(\frac{p_i}{p_e}\right) \tag{5.48}$$

where $T_r = 1400$ K and $(\gamma - 1)/\gamma = 0.24$ are appropriate average values to use for initial estimates.

Given the equivalence ratio ϕ and initial conditions T_1 (K), $p_1 = p_i$ (Pa), and v_1 (m^3/kg air), the state at point 2 at the end of compression through a volume ratio $v_1/v_2 = r_c$ is obtained from Eq. (4.25*a*) and the isentropic compression chart (Fig. 4-4). The compression work W_C (J/kg air) is found from Eq. (5.6) with the internal energy change determined from the unburned mixture chart (Fig. 4-3).

The use of charts to relate the state of the burned mixture to the state of the unburned mixture prior to combustion, for adiabatic constant-volume and constant-pressure combustion, has already been illustrated in Sec. 4.5.3.

For the *constant-volume cycle*,

$$u_3 = u_{s2} + \Delta u^\circ_{f,u} \qquad \text{J/kg air} \tag{5.49}$$

where u_{s2} is the sensible internal energy of the unburned mixture at T_2 from Fig. 4-3 and $\Delta u^\circ_{f,u}$ is the internal energy of formation of the unburned mixture [given by Eq. (4.32)]. Since $v_3 = v_2$, the burned gas state at point 3 can be located on the appropriate burned gas chart (Figs. 4-5 to 4-9).

For the *constant-pressure cycle*,

$$h_3 = h_{s2} + \Delta h^\circ_{f,u} \qquad \text{J/kg air} \tag{5.50}$$

Since $p_3 = p_2$, the burned gas state at point 3 can be located (by iteration) on the high-temperature burned gas charts, as illustrated by Example 4.5.

For the *limited-pressure cycle*, application of the first law to the mixture between states 2 and 3*b* gives

$$h_{3b} = u_{3b} + p_3 v_{3b} = u_2 + p_3 v_2 = u_{s2} + \Delta u^\circ_{f,u} + p_3 v_2 \qquad \text{J/kg air} \tag{5.51}$$

Since p_3 for a limited-pressure cycle is given, point 3*b* can be located on the appropriate burned gas chart.

The expansion process 3-4 follows an isentropic line from v_3 to v_4 ($v_4 = v_1$) on the burned mixture charts. Equation (5.9) [or (5.11) or (5.13)] now gives the expansion work W_E. The state of the residual gas at points 5 and 6 in the cycle is obtained by continuing this isentropic expansion from state 4 to $p = p_e$. The residual gas temperature can be read from the equilibrium burned gas chart; the residual gas fraction is obtained from Eq. (5.17). If values of T_r and x_r were assumed at the start of the cycle calculation to determine T_1, the assumed values

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can be checked against the calculated values and an additional cycle computation carried out with the new calculated values if required. The convergence is rapid.

The indicated fuel conversion efficiency is obtained from Eq. (5.1). The indicated mean effective pressure is obtained from Eq. (5.2). The volumetric efficiency (see Sec. 2.10) for a four-stroke cycle engine is given by

$$\eta_v = \frac{r_c(1 - x_r)}{v_1 \rho_{a,i}(r_c - 1)} \tag{5.52}$$

where $\rho_{a,i}$ is the inlet air density (in kilograms per cubic meter) and v_1 is the chart mixture specific volume (in cubic meters per kilogram of air in the original mixture).

Example 5.2. Calculate the performance characteristics of the constant-volume fuel-air cycle defined by the initial conditions of Examples 4.2, 4.3, and 4.5. The compression ratio is 8; the fuel is isooctane and the mixture is stoichiometric; the pressure and temperature inside the cylinder at the start of compression are 1 atm and 350 K, respectively. Use the notation of Fig. 5-2*a* to define the states at the beginning and end of each process.

Example 4.2 analyzed the compression process:

$$T_1 = 350 \text{ K}, \quad p_1 = 101.3 \text{ kPa}, \quad v_1 = 1 \text{ m}^3/\text{kg air}, \quad u_{s1} = 40 \text{ kJ/kg air}$$

$$T_2 = 682 \text{ K}, \quad p_2 = 1.57 \text{ MPa}, \quad v_2 = 0.125 \text{ m}^3/\text{kg air}, \quad u_{s2} = 350 \text{ kJ/kg air}$$

$$W_{1\text{-}2} = W_C = -310 \text{ kJ/kg air}$$

Example 4.5 analyzed the constant-volume adiabatic combustion process (it was assumed that the residual gas fraction was 0.08):

$$u_{b3} = u_{u2} = u_{s,u2} + \Delta u^\circ_{f,u} = -5 \text{ kJ/kg air}, \quad s_3 = 9.33 \text{ kJ/kg air} \cdot \text{K}$$

$$v_3 = v_2 = 0.125 \text{ m}^3/\text{kg air}, \quad T_3 = 2825 \text{ K}, \quad p_3 = 7100 \text{ kPa}$$

Example 4.3 analyzed the expansion process, from these conditions after combustion at TC, to the volume v_4 at BC of 1 m^3/kg air:

$$T_4 = 1840 \text{ K}, \quad p_4 = 570 \text{ kPa}, \quad u_4 = -1540 \text{ kJ/kg air}$$

$$W_{3\text{-}4} = W_E = 1535 \text{ kJ/kg air}$$

To check the assumed residual gas fraction, the constant entropy expansion process on the chart in Fig. 4-8 is continued from state 4 to the exhaust pressure p_5 of 1 atm = 101.3 kPa. This gives $v_5 = 4.0$ m^3/kg air and $T_5 = 1320$ K. The residual fraction from Eq. (5.17) is

$$x_r = \frac{v_2}{v_5} = \frac{0.125}{4.0} = 0.031$$

which is significantly different from the assumed value of 0.08. The combustion and expansion calculations are now repeated with the new residual fraction of 0.031 (the compression process will not be changed significantly and the initial temperature is

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assumed fixed):

$$u_{b3} = 350 - 118.2 - 2956 \times 0.031 = 140 \text{ kJ/kg air}$$

With $v_3 = 0.125$ m^3/kg air, Fig. 4-8 gives

$$p_3 = 7270 \text{ kPa}, \qquad T_3 = 2890 \text{ K}$$

Expand at constant entropy to $v_4 = 1$ m^3/kg air:

$$p_4 = 595 \text{ kPa}, \qquad T_4 = 1920 \text{ K}, \qquad u_4 = -1457 \text{ kJ/kg air}$$

$$W_{3\text{-}4} = W_E = 1597 \text{ kJ/kg air}$$

Continue expansion at constant entropy to the exhaust pressure, $p_5 = 1$ atm:

$$v_5 = 4 \text{ m}^3\text{/kg air}, \qquad T_5 = 1360 \text{ K}$$

Equation (5.17) now gives the residual fraction

$$x_r = \frac{v_2}{v_5} = \frac{0.125}{4} = 0.031$$

which agrees with the value assumed for the second iteration.

The fuel conversion efficiency can now be calculated:

$$\eta_{f,i} = \frac{W_E + W_C}{m_f Q_{\text{LHV}}}$$

where $m_f = \text{kg fuel/kg air at state 1} = \left(\frac{F}{A}\right)(1 - x_r)$

Thus

$$\eta_{f,i} = \frac{1597 - 310}{0.0661 \times (1 - 0.031) \times 44.4 \times 10^3} = 0.45$$

The indicated mean effective pressure is

$$\text{imep} = \frac{W_E + W_C}{V_d} = \frac{1597 - 310}{1 - 0.125} = 1470 \text{ kPa}$$

or

$$\frac{\text{imep}}{p_1} = 14.6$$

5.5.2 CI Engine Cycle Simulation

With a diesel engine fuel-air cycle calculation, additional factors must be taken into account. The mixture during compression is air plus a small amount of residual gas. At point 2 liquid fuel is injected into the hot compressed air at temperature T_2; as the fuel vaporizes and heats up, the air is cooled. For a constant-volume mixing process which is adiabatic overall, the mixture internal energy is unchanged, i.e.:

$$m_f[u_{fg} + c_{v,f}(T_{2'} - T_0)] + m_a c_{v,a}(T_{2'} - T_2) = 0 \qquad (5.53)$$

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where m_f is the mass of fuel injected, u_{fg} is the latent heat of vaporization of the fuel, $c_{v,f}$ is the specific heat at constant volume of the fuel vapor, $T_{2'}$ is the mixture temperature (assumed uniform) after vaporization and mixing is complete, m_a is the mass of air used, and $c_{v,a}$ is the specific heat at constant volume of air. Substitution of typical values for fuel and air properties gives $(T_2 - T_{2'}) \approx 70$ K at full load. Localized cooling in a real engine will be greater.

The limited-pressure cycle is a better approximation to the diesel engine than the constant-pressure or constant-volume cycles.

Note that because nonuniformities in the fuel/air ratio exist during and after combustion in the CI engine, the burned gas charts which assume uniform composition will not be as accurate an approximation to working fluid properties as they are for SI engines.

5.5.3 Results of Cycle Calculations

Extensive results of constant-volume fuel-air cycle calculations are available.[1, 3, 4] Efficiency is little affected by variables other than the compression ratio r_c and equivalence ratio ϕ. Figures 5-9 and 5-10 show the effect of variations in these two parameters on indicated fuel conversion efficiency and mean effective pressure. From the available results, the following conclusions can be drawn:

1. The effect of increasing the compression ratio on efficiency at a constant equivalence ratio is similar to that demonstrated by the constant γ constant-volume cycle analysis (provided the appropriate value of γ is used; see Fig. 5-19).
2. As the equivalence ratio is decreased below unity (i.e., the fuel-air mixture is made progressively leaner than stoichiometric), the efficiency increases. This occurs because the burned gas temperatures after combustion decrease, decreasing the burned gas specific heats and thereby increasing the effective value of γ over the expansion stroke. The efficiency increases because, for a given volume-expansion ratio, the burned gases expand through a larger temperature ratio prior to exhaust; therefore, per unit mass of fuel, the expansion stroke work is increased.
3. As the equivalence ratio increases above unity (i.e., the mixture is made progressively richer than stoichiometric), the efficiency decreases because lack of sufficient air for complete oxidation of the fuel more than offsets the effect of decreasing burned gas temperatures which decrease the mixture's specific heats.
4. The mean effective pressure, from Eq. (5.2), is proportional to the product $\phi\eta_{f,i}$. This exhibits a maximum between $\phi \approx 1.0$ and $\phi \approx 1.1$, i.e., slightly rich of stoichiometric. For ϕ less than the value corresponding to this maximum, the decreasing fuel mass per unit displaced volume more than offsets the increasing fuel conversion efficiency. For ϕ greater than this value, the decreasing fuel conversion efficiency (due to decreasing combustion efficiency) more than offsets the increasing fuel mass.

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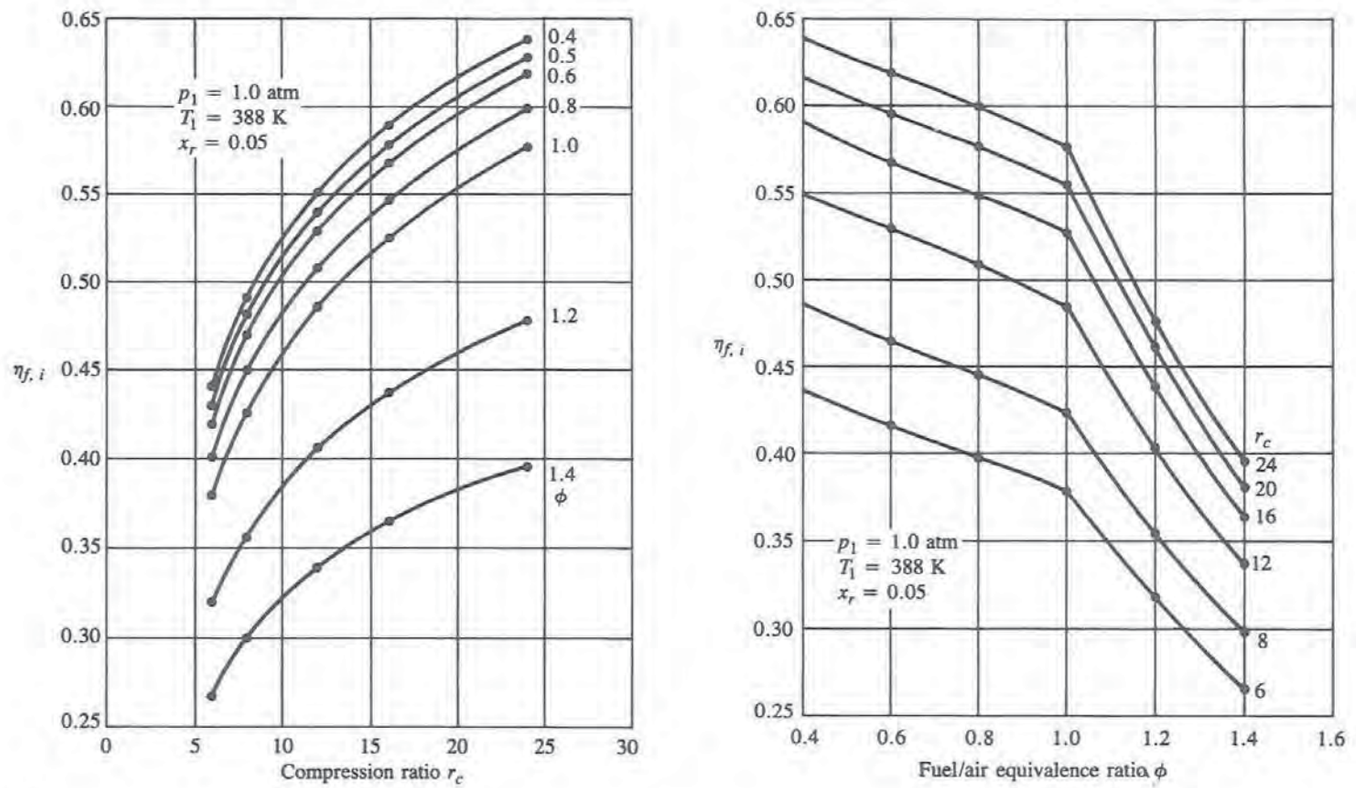


FIGURE 5-9
Fuel-air cycle results for indicated fuel conversion efficiency as a function of compression ratio and equivalence ratio. Fuel: octene; $p_1 = 1$ atm, $T_1 = 388$ K, $x_r = 0.05$. (*From Edson and Taylor.*[4])

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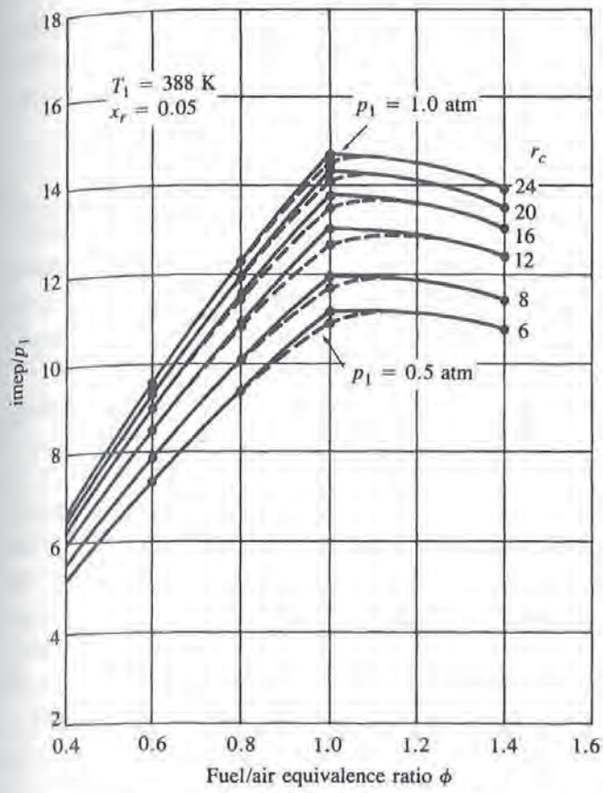


FIGURE 5-10
Fuel-air cycle results for indicated mean effective pressure as a function of equivalence ratio and compression ratio. Fuel: octene; $p_1 = 1$ atm, $T_1 = 388$ K, $x_r = 0.05$. (*From Edson and Taylor.*[4])

5. Variations in initial pressure, inlet temperature, residual gas fraction, and atmospheric moisture fraction have only a modest effect on the fuel conversion efficiency. The effects of variations in these variables on imep are more substantial, however, because imep depends directly on the initial charge density.
6. Comparison of results from limited-pressure and constant-volume fuel-air cycles[1] shows that placing a realistic limit on the maximum pressure reduces the advantages of increased compression ratio on both efficiency and imep.

5.6 OVEREXPANDED ENGINE CYCLES

The gas pressure within the cylinder of a conventional four-stroke engine at exhaust valve opening is greater than the exhaust pressure. The available energy of the cylinder gases at this point in the cycle is then dissipated in the exhaust blowdown process. Additional expansion within the engine cylinder would increase the indicated work per cycle, as shown in Fig. 5-11, where expansion continues beyond point 4′ (the conventional ideal cycle exhaust valve opening point) at $V_{4'} = r_c V_c$ to point 4 at $V_4 = r_e V_c$. The exhaust stroke in this overexpanded cycle is 4-5-6. The intake stroke is 6-1. The area 14′451 has been added

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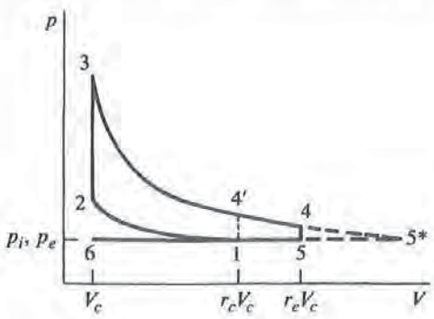


FIGURE 5-11
Pressure-volume diagram for overexpanded engine cycle (1234561) and Atkinson cycle (1235*61). r_c and r_e are volumetric compression and expansion ratios, respectively.

to the conventional cycle p-V diagram area, for the same fuel input, thereby increasing the engine's efficiency.

Complete expansion within the cylinder to exhaust pressure p_e (point 5*) is called the *Atkinson cycle*. Unthrottled operation is shown in Fig. 5-11; throttled operating cycles can also be generated. Many crank and valve mechanisms have been proposed to achieve this additional expansion. For example, it can be achieved in a conventional four-stroke cycle engine by suitable choice of exhaust valve opening and intake valve closing positions relative to BC. If the crank angle between exhaust valve opening and BC on the expansion stroke is less than the crank angle between BC and intake valve closing on the compression stroke, then the actual volumetric expansion ratio is greater than the actual volumetric compression ratio (these *actual* ratios are both less than the *nominal* compression ratio with normal valve timing).

The effect of overexpansion on efficiency can be estimated from an analysis of the ideal cycle shown in Fig. 5-11. An ideal gas working fluid with specific heats constant throughout the cycle will be assumed. The indicated work per cycle for the overexpanded cycle is

$$W_{c,i} = m[(u_3 - u_4) - (u_2 - u_1) - p_1(v_4 - v_1)] \tag{5.54}$$

The isentropic relations for 1-2 and 3-4 are

$$\frac{T_2}{T_1} = r_c^{\gamma-1} \qquad \frac{T_3}{T_4} = r_e^{\gamma-1}$$

With Eq. (5.33) to relate T_3 and T_2, the following expression for indicated fuel conversion efficiency can be derived from Eqs. (5.1), (5.29), and (5.54):

$$\eta_{f,i} = 1 - \frac{1}{(rr_c)^{\gamma-1}}\left\{1 + \frac{c_v T_1}{Q^*} r_c^{\gamma-1}[1 - \gamma r^{\gamma-1} + (\gamma - 1)r^{\gamma}]\right\} \tag{5.55}$$

where

$$r = \frac{r_e}{r_c}$$

Note that the efficiency given by Eq. (5.55) is a function of load (via Q^*), and is a

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maximum at maximum load. This contrasts with the ideal constant-volume cycle efficiency [Eq. (5.31)], which is independent of load. The ratio r_e/r_c for complete expansion is given by

$$r^{\gamma} = 1 + \frac{Q^*}{c_v T_1 r_c^{\gamma - 1}} \tag{5.56}$$

The effect of overexpansion on fuel conversion efficiency is shown in Fig. 5-12 for $r_c = 4$, 8, and 16 with $\gamma = 1.3$. The ratio of overexpanded cycle efficiency to the standard cycle efficiency is plotted against r. The Atkinson cycle (complete expansion) values are indicated by the transition from a continuous line to a dashed line. Significant increases in efficiency can be achieved, especially at low compression ratios.

One major disadvantage of this cycle is that imep and power density decrease significantly because only part of the total displaced volume is filled with fresh charge. From Eqs. (5.2), (5.29), and the relations $V_d = V_1(r_e - 1)/r_c$ and

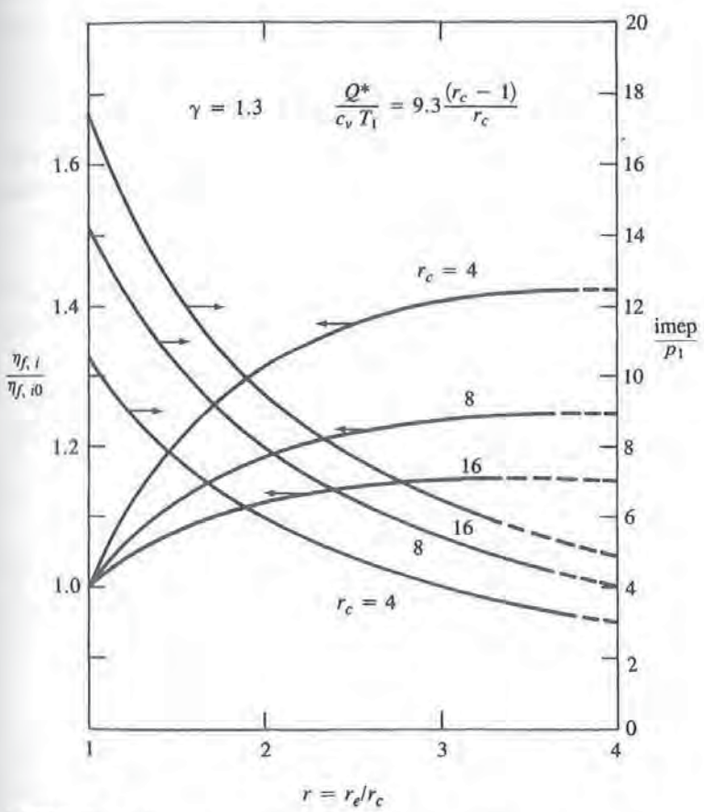


FIGURE 5-12
Indicated fuel conversion efficiency and mean effective pressure for overexpanded engine cycle as a function of r_e/r_c. Efficiencies given relative to $r_e = r_c$ value, $\eta_{f,i0}$. $\gamma = 1.3$, $Q^*/(c_v T_1) = 9.3(r_c - 1)/r_c$. Solid to dashed line transition marks the complete expansion point (Atkinson cycle).

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$p_1 V_1 = mRT_1$ it follows that imep for the overexpanded cycle is given by

$$\frac{\text{imep}}{p_1} = \left(\frac{Q^*}{c_v T_1}\right)\left(\frac{1}{\gamma - 1}\right)\left(\frac{r_c}{r_e - 1}\right)\eta_{f,i} \tag{5.57}$$

Values of imep/p_1 are plotted in Fig. 5-12 as a function of $r(=r_e/r_c)$. The substantial decrease from the standard constant-volume cycle values at $r = 1$ is clear.

5.7 AVAILABILITY ANALYSIS OF ENGINE PROCESSES

5.7.1 Availability Relationships

Of interest in engine performance analysis is the amount of *useful work* that can be extracted from the gases within the cylinder at each point in the operating cycle. The problem is that of determining the maximum possible work output (or minimum work input) when a system (the charge within the cylinder) is taken from one specified state to another in the presence of a specified environment (the atmosphere). The first and second laws of thermodynamics together define this maximum or minimum work, which is best expressed in terms of the property of such a system-environment combination called *availability*[5] or sometimes *exergy*.[6, 7]

Consider the system-atmosphere combination shown in Fig. 5-13. In the absence of mass flow across the system boundary, as the system changes from state 1 to state 2, the first and second laws give

$$W_{1\text{-}2} = -(U_2 - U_1) + Q_{1\text{-}2}$$
$$Q_{1\text{-}2} \leq T_0(S_2 - S_1)$$

Combining these two equations gives the *total work* transfer:

$$W_{t,\,1\text{-}2} \leq -[(U_2 - U_1) - T_0(S_2 - S_1)] \tag{5.58}$$

The work done by the system against the atmosphere is not available for pro-

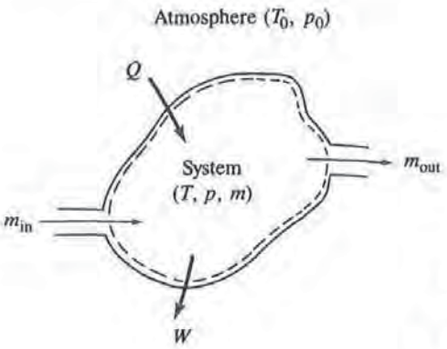


FIGURE 5-13
System-atmosphere configuration for availability analysis.

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ductive use. It must, therefore, be subtracted from the total work to obtain the *useful work transfer*:

$$W_{U,1\text{-}2} \leq -[(U_2 - U_1) + p_0(V_2 - V_1) - T_0(S_2 - S_1)] \tag{5.59}$$

The *maximum* useful work will be obtained when the final state of the system is in thermal and mechanical equilibrium with the atmosphere.† The *availability* of this system which is in communication with the atmosphere

$$A = U + p_0 V - T_0 S \tag{5.60}$$

is thus the property of the system-atmosphere combination which defines its capacity for useful work. The useful work such a system-atmosphere combination can provide, as the system changes from state 1 to state 2, is less than or equal to the change in availability:

$$W_{U,\,1\text{-}2} \leq -(A_2 - A_1) \tag{5.61}$$

When mass flow across the system boundary occurs, the availability associated with this mass flow is

$$B = H - T_0 S \tag{5.62}$$

B is usually called the *steady-flow availability function.*

With these relations, an availability balance for the gas working-fluid system around the engine cycle can be carried out. For any process between specified end states which this system undergoes (interacting only with the atmosphere), the change in availability ΔA is given by

$$\Delta A = A_{\text{in}} - A_{\text{out}} - A_{\text{destroyed}} \tag{5.63}$$

The availability transfers in and out occur as a result of work transfers, heat transfers, and mass transfers across the system boundary. The availability transfer associated with a work transfer is equal to the work transfer. The availability transfer dA_Q associated with a heat transfer δQ occurring when the system temperature is T is given by

$$dA_Q = \delta Q\left(1 - \frac{T_0}{T}\right) \tag{5.64}$$

since both an energy and entropy transfer occurs across the system boundary. The availability transfer associated with a mass transfer is given by Eq. (5.62).

† The issue of chemical equilibrium with the atmosphere must also be considered. Attainment of chemical equilibrium with the environment requires the capacity to extract work from the partial pressure differences between the various species in the working fluid and the partial pressures of those same species in the environment. This would require such devices as ideal semipermeable membranes and efficient low input pressure, high pressure ratio, expansion devices (which are not generally available for mobile power plant systems). Inclusion of these additional steps to achieve full equilibrium beyond equality of temperature and pressure is inappropriate.[8]

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Availability is destroyed by the irreversibilities that occur in any real process. The availability destroyed is given by

$$A_{\text{destroyed}} = T_0 \Delta S_{\text{irrev}} \tag{5.65}$$

where ΔS_{irrev} is the entropy increase associated with the irreversibilities occurring within the system boundary.[7, 8]

5.7.2 Entropy Changes in Ideal Cycles

The ideal models of engine processes examined earlier in this chapter provide useful illustrative examples for availability analysis. First, however, we will consider the variation in the entropy of the cylinder gases as they proceed through these ideal operating cycles.

For an adiabatic reversible compression process, the entropy is constant. For the combustion process in each of the ideal gas standard cycles, the entropy increase can be calculated from the relations of Eq. (4.14) (with constant specific heats):

$$s - s_0 = c_v \ln\left(\frac{T}{T_0}\right) + R \ln\left(\frac{v}{v_0}\right) = c_p \ln\left(\frac{T}{T_0}\right) - R \ln\left(\frac{p}{p_0}\right)$$

For the constant-volume cycle:

$$S_3 - S_2 = m(s_3 - s_2) = mc_v \ln\left(\frac{T_3}{T_2}\right) \tag{5.66a}$$

For the constant-pressure cycle:

$$S_3 - S_2 = m(s_3 - s_2) = mc_p \ln\left(\frac{T_3}{T_2}\right) \tag{5.66b}$$

For the limited-pressure cycle:

$$S_{3b} - S_2 = c_v \ln\left(\frac{T_{3a}}{T_2}\right) + c_p \ln\left(\frac{T_{3b}}{T_{3a}}\right) = c_v \ln \alpha + c_p \ln \beta \tag{5.66c}$$

with α and β defined by Eq. (5.42).

Since the expansion process, after combustion is complete, is adiabatic and reversible, there is no further change in entropy, 3 to 4 (or 3*b* to 4). Figure 5-14 shows the entropy changes that occur during each process of these three ideal engine operating cycles, calculated from the above equations, on a T-s diagram. The three cycles shown correspond to those of the p-V diagrams of Fig. 5-6 with $r_c = 12$, $\gamma = 1.3$, and $Q^*/(c_v T_1) = 8.525$. Since the combustion process was assumed to be adiabatic, the increase in entropy during combustion clearly demonstrates the irreversible nature of this process.

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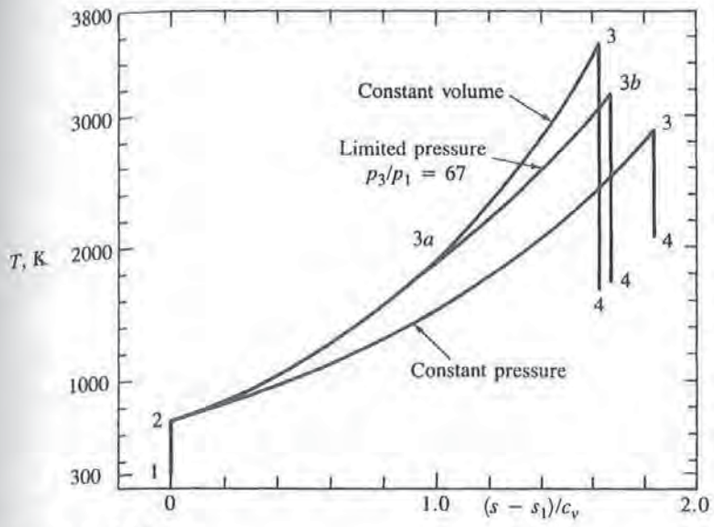


FIGURE 5-14
Temperature-entropy diagram for ideal gas constant-volume, constant-pressure, and limited-pressure cycles. Assumptions same as in Fig. 5-6.

5.7.3 Availability Analysis of Ideal Cycles

An availability analysis for each process in the ideal cycle illustrates the magnitude of the availability transfers and where the losses in availability occur.[9] In general, for the system of Fig. 5-4 in communication with an atmosphere at p_0, T_0 as indicated in Fig. 5-13, the change in availability between states i and j during the portion of the cycle when the valves are closed is given by

$$A_j - A_i = m(a_j - a_i) = m[(u_j - u_i) + p_0(v_j - v_i) - T_0(s_j - s_i)] \qquad (5.67)$$

The appropriate normalizing quantity for these changes in availability is the thermomechanical availability of the fuel supplied to the engine cylinder each cycle, $m_f(-\Delta g_{298})$† (see Sec. 3.6.2). However, it is more convenient to use $m_f(-\Delta h_{298})$‡ $= m_f Q_{\text{LHV}}$ as the normalizing quantity since it can be related to the temperature rise during combustion via Eq. (5.28). As shown in Table 3.3, these two quantities differ by only a few percent for common hydrocarbon fuels. Equation (5.67), with Eq. (5.29), then becomes

$$\frac{A_j - A_i}{m_f Q_{\text{LHV}}} = \frac{m(a_j - a_i)}{m_f Q_{\text{LHV}}} = \frac{a_j - a_i}{Q^*} \qquad (5.68)$$

† Δg_{298} is the Gibbs free energy change for the combustion reaction, per unit mass of fuel.
‡ Δh_{298} is the enthalpy change for the combustion reaction, again per unit mass of fuel.

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The compression process is isentropic, so:

$$\frac{A_2 - A_1}{m_f Q_{LHV}} = \frac{a_2 - a_1}{Q^*} = \frac{(u_2 - u_1) + p_0(v_2 - v_1)}{Q^*}$$
$$= \frac{c_v T_1}{Q^*}\left[\left(\frac{T_2}{T_1} - 1\right) - (\gamma - 1)\left(1 - \frac{V_2}{V_1}\right)\right]$$
$$= \frac{c_v T_1}{Q^*}\left[(r_c^\gamma - 1) - (\gamma - 1)\left(1 - \frac{1}{r_c}\right)\right] \tag{5.69}$$

where we have assumed $p_0 = p_1$. The first term in the square brackets is the compression stroke work transfer. The second term is the work done by the atmosphere on the system, which is subtracted because it does not increase the *useful work* which the system-atmosphere combination can perform.

During combustion, for the constant-volume cycle, the volume and internal energy remain unchanged (Eqs. 5.7*a*, *b*). Thus

$$\frac{A_3 - A_2}{m_f Q_{LHV}} = \frac{a_3 - a_2}{Q^*} = -\frac{T_0(s_3 - s_2)}{Q^*}$$
$$= -\frac{c_v T_0}{Q^*}\ln\left(\frac{T_3}{T_2}\right) = -\frac{c_v T_0}{Q^*}\ln\left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}}\right) \tag{5.70}$$

This loss in availability results from the increase in entropy associated with the irreversibilities of the combustion process. This lost or destroyed availability, as a fraction of the initial availability of the fuel-air mixture, decreases as the compression ratio increases (since T_2 increases as the compression ratio increases, T_3/T_2 decreases for fixed heat addition) and increases as Q^* decreases [e.g., when the mixture is made leaner; see Eq. (5.46)]. The changes in availability during combustion for the constant-pressure and limited-pressure cycles are more complex because there is a transfer of availability out of the system equal to the expansion work transfer which occurs.

For the constant-volume cycle expansion stroke:

$$\frac{A_4 - A_3}{m_f Q_{LHV}} = \frac{a_4 - a_3}{Q^*} = \frac{(u_4 - u_3) + p_0(v_4 - v_3)}{Q^*}$$
$$= \frac{c_v T_3}{Q^*}\left[\left(\frac{T_4}{T_3} - 1\right) + (\gamma - 1)\left(\frac{p_0}{p_3}\right)\left(\frac{V_4}{V_3} - 1\right)\right]$$
$$= -\left[\left(1 + \frac{c_v T_1 r_c^{\gamma-1}}{Q^*}\right)\left(1 - \frac{1}{r_c^{\gamma-1}}\right) - (\gamma - 1)\frac{c_v T_1}{Q^*}\left(\frac{r_c - 1}{r_c}\right)\right] \tag{5.71}$$

The availability of the exhaust gas at state 4 relative to its availability at (T_1, p_1) is given by

$$\frac{A_4 - A_1}{m_f Q_{LHV}} = \frac{c_v T_1}{Q^*}\left[\left(\frac{T_4}{T_1} - 1\right) - \frac{T_0}{T_1}\ln\left(\frac{T_4}{T_1}\right)\right] \tag{5.72}$$

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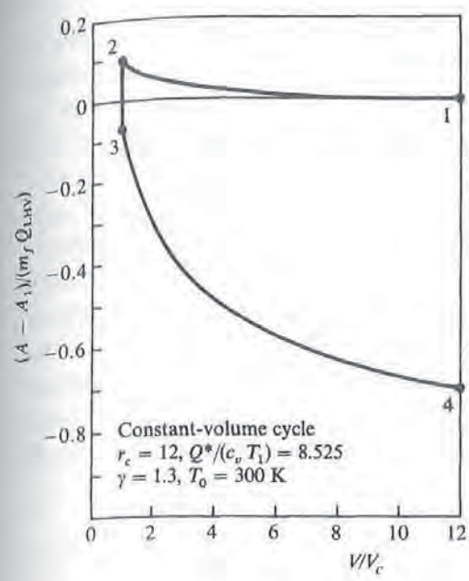


FIGURE 5-15
Availability of cylinder charge relative to availability at state 1 for constant-volume ideal gas cycle as a function of cylinder volume. Availability made dimensionless by $m_f Q_{LHV}$. Assumptions as in Fig. 5-6.

The availability of the gases inside the cylinder relative to their availability at (T_1, p_1) over the compression and expansion strokes of the constant-volume operating cycle example used in Figs. 5-6 and 5-14 is shown in Fig. 5-15. Equations (5.69) and (5.71), with T_2 and T_4 replaced by temperatures intermediate between T_1 and T_2 and T_3 and T_4, respectively, were used to compute the variations during compression and expansion. Table 5.3 summarizes the changes in availability during each process and the availability of the cylinder gases, at the beginning and end of each process, relative to the datum for the atmosphere

TABLE 5.3
Availability changes in constant-volume cycle

Process or state	$\frac{A_j - A_i}{m_f Q_{LHV}}$	$\frac{A_i}{m_f Q_{LHV}}$
1		1.0294
1-2	0.0976	
2		1.1270
2-3	−0.1710	
3		0.9560
3-4	−0.6237	
4		0.3323
Fuel conversion efficiency $\eta_{f,i}$	0.526	
Availability conversion efficiency $\eta_{a,i}$	0.511	

$r_c = 12$, $\gamma = 1.3$, $Q^*/(c_v T_1) = 8.525$, $T_0 = 300$ K, $T_1 = 333$ K.

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(1 atm, 300 K). The availability at state 1 of the fuel, air, residual-gas mixture is $(1.0286 + 0.0008)m_f Q_{LHV}$. 1.0286 is the ratio $(-\Delta g^\circ_{298})/(-\Delta h^\circ_{298})$ for isooctane (see Table 3.3). The second number, 0.0008, allows for the difference between T_1 and T_0. Because both work-transfer processes in this ideal cycle case are reversible, the fuel conversion efficiency $\eta_{f,i}$ is given by $(A_3 - A_4)/(m_f Q_{LHV}) - (A_2 - A_1)/(m_f Q_{LHV})$. It is, of course, equal to the value obtained for $r_c = 12$ and $\gamma = 1.3$ from the formula for efficiency (Eq. 5.31), obtained previously. The availability conversion efficiency is $\eta_{f,i}/1.0286$. Note that it is the availability destroyed during combustion, plus the inability of this ideal constant-volume cycle to use the availability remaining in the gas at state 4, that decrease the availability conversion efficiency below unity. Both these loss mechanisms decrease in magnitude, relative to the fuel availability, as the compression ratio increases. This is the fundamental reason why engine indicated efficiency increases with an increasing compression ratio.

5.7.4 Effect of Equivalence Ratio

The fuel-air cycle with its more accurate models for working fluid properties can be used to examine the effect of variations in the fuel/air equivalence ratio on the availability conversion efficiency. Figure 5-16 shows the temperature attained and the entropy rise that occurs in constant-volume combustion of a fuel-air mixture of different equivalence ratios, following isentropic compression from ambient temperature and pressure through different volumetric compression ratios.[8] The entropy increase is the result of irreversibilities in the combustion process and mixing of complete combustion products with excess air. The significance of these combustion-related losses—the destruction of availability that occurs in this process—is shown in Fig. 5-17 where the availability after constant-volume combustion divided by the availability of the initial fuel-air mixture is shown as a function of equivalence ratio for compression ratios of 12 and 36.[8] The loss of

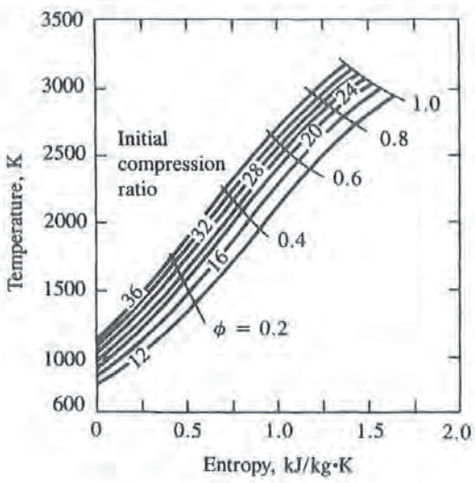


FIGURE 5-16
Temperature and entropy of combustion products after constant-volume combustion following isentropic compression from ambient conditions through specified compression ratio as a function of compression ratio and equivalence ratio. (*From Flynn et al.*[8])

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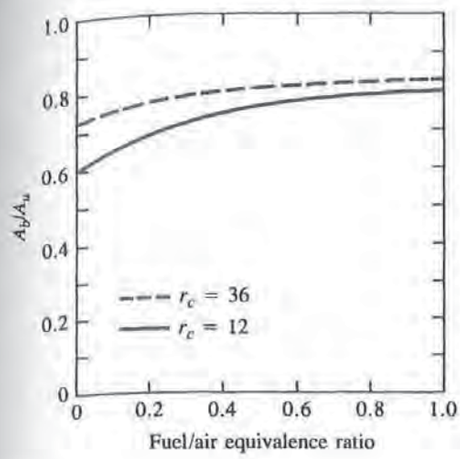


FIGURE 5-17
Availability of combustion products after constant-volume combustion relative to availability before combustion following isentropic compression from ambient through specified compression ratio as a function of equivalence ratio. (*From Flynn et al.*[8])

availability increases as the equivalence ratio decreases.† The combustion loss is a stronger function of the rise in temperature and pressure which occurs than of the change in the specific heat ratio that occurs.

Why then does engine efficiency increase with a decreasing equivalence ratio as shown in Fig. 5-9? The reason is that the expansion stroke work transfer, as a fraction of the fuel availability, increases as the equivalence ratio decreases; hence, the availability lost in the exhaust process, again expressed as a fraction of the fuel availability, decreases. The increase in the expansion stroke work as the equivalence ratio decreases more than offsets the increase in the availability lost during combustion; so the availability conversion efficiency (or the fuel conversion efficiency which closely approximates it) increases.

5.8 COMPARISON WITH REAL ENGINE CYCLES

To put these ideal models of engine processes in perspective, this chapter will conclude with a brief discussion of the additional effects which are important in real engine processes.

A comparison of a real engine p-V diagram over the compression and expansion strokes with an equivalent fuel-air cycle analysis is shown in Fig. 5-18.[4] The real engine and the fuel-air cycle have the same geometric compression ratio, fuel chemical composition and equivalence ratio, residual fraction and mixture density before compression. Midway through the compression stroke,

† This is consistent with the ideal gas standard cycle result (Eq. 5.70). As ϕ decreases, so does $Q^*/(c_v T_1)$. The factor which multiplies the natural logarithm (which increases) has a greater impact than the logarithmic term (which decreases).

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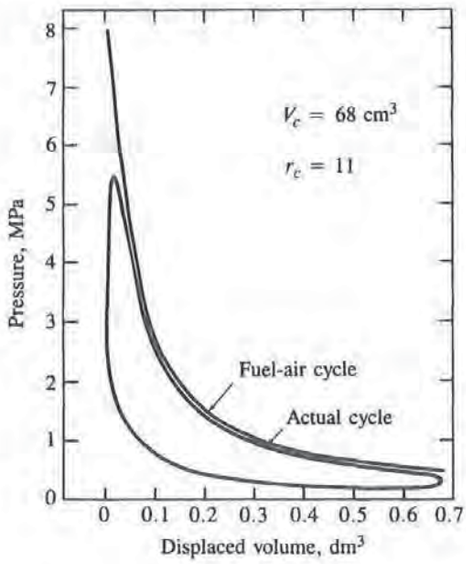


FIGURE 5-18
Pressure-volume diagram for actual spark-ignition engine compared with that for equivalent fuel-air cycle. $r_c = 11$. (*From Edson and Taylor.*[4])

the pressure in the fuel-air cycle has been made equal to the real cycle pressure.† The compression stroke pressures for the two cycles essentially coincide. Modest differences in pressure during intake and the early part of the compression process result from the pressure drop across the intake valve during the intake process and the closing of the intake valve 40 to 60° after BC in the real engine. The expansion stroke pressures for the engine fall below the fuel-air cycle pressures for the following reasons: (1) heat transfer from the burned gases to the walls; (2) finite time required to burn the charge; (3) exhaust blowdown loss due to opening the exhaust valve before BC; (4) gas flow into crevice regions and leakage past the piston rings; (5) incomplete combustion of the charge.

These differences, in decreasing order of importance, are described below. Together, they contribute to the enclosed area on the p-V diagram for a properly adjusted engine with optimum timing being about 80 percent of the enclosed area of an equivalent fuel-air cycle p-V diagram. The indicated fuel conversion or availability conversion efficiency of the actual engine is therefore about 0.8 times the efficiency calculated for the fuel-air cycle.[1] Use is often made of this ratio to estimate the performance of actual engines from fuel-air cycle results.

1. *Heat transfer.* Heat transfer from the unburned mixture to the cylinder walls has a negligible effect on the p-V line for the compression process. Heat transfer from the burned gases is much more important (see Chap. 12). Due to heat transfer during combustion, the pressure at the end of combustion in the real

† Note that in the fuel-air cycle with idealized valve timing, the compression process starts immediately after BC. In most engines, the charge compression starts later, close to the time that the inlet valve closes some 40 to 60° after BC. This matching process is approximate.

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cycle will be lower. During expansion, heat transfer will cause the gas pressure in the real cycle to fall below an isentropic expansion line as the volume increases. A decrease in efficiency results from this heat loss.

2. *Finite combustion time.* In an SI engine with spark-timing adjusted for optimum efficiency, combustion typically starts 10 to 40 crank angle degrees before TC, is half complete at about 10° after TC, and is essentially complete 30 to 40° after TC. Peak pressure occurs at about 15° after TC (see Fig. 1-8). In a diesel engine, the burning process starts shortly before TC. The pressure rises rapidly to a peak some 5 to 10° after TC since the initial rate of burning is fast. However, the final stages of burning are much slower, and combustion continues until 40 to 50° after TC (see Fig. 1-15). Thus, the peak pressure in the engine is substantially below the fuel-air cycle peak pressure value, because combustion continues until well after TC, when the cylinder volume is much greater than the clearance volume. After peak pressure, expansion stroke pressures in the engine are higher than fuel-air cycle values in the absence of other loss mechanisms, because less work has been extracted from the cylinder gases. A comparison of the constant-volume and limited-pressure cycles in Fig. 5-6 demonstrates this point.

   For spark or fuel-injection timing which is retarded from the optimum for maximum efficiency, the peak pressure in the real cycle will be lower, and expansion stroke pressures after the peak pressure will be higher than in the optimum timing cycle.

3. *Exhaust blowdown loss.* In the real engine operating cycle, the exhaust valve is opened some 60° before BC to reduce the pressure during the first part of the exhaust stroke in four-stroke engines and to allow time for scavenging in two-stroke engines. The gas pressure at the end of the expansion stroke is therefore reduced below the isentropic line. A decrease in expansion-stroke work transfer results.

4. *Crevice effects and leakage.* As the cylinder pressure increases, gas flows into crevices such as the regions between the piston, piston rings, and cylinder wall. These crevice regions can comprise a few percent of the clearance volume. This flow reduces the mass in the volume above the piston crown, and this flow is cooled by heat transfer to the crevice walls. In premixed charge engines, some of this gas is unburned and some of it will not burn. Though much of this gas returns to the cylinder later in the expansion, a fraction, from behind and between the piston rings, flows into the crankcase. However, leakage in a well-designed and maintained engine is small (usually less than one percent of the charge). All these effects reduce the cylinder pressure during the latter stages of compression, during combustion, and during expansion below the value that would result if crevice and leakage effects were absent.

5. *Incomplete combustion.* Combustion of the cylinder charge is incomplete; the exhaust gases contain combustible species. For example, in spark-ignition engines the hydrocarbon emissions from a warmed-up engine (which come largely from the crevice regions) are 2 to 3 percent of the fuel mass under

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normal operating conditions; carbon monoxide and hydrogen in the exhaust contain an additional 1 to 2 percent or more of the fuel energy, even with excess air present (see Sec. 4.9). Hence, the chemical energy of the fuel which is released in the actual engine is about 5 percent less than the chemical energy of the fuel inducted (the combustion efficiency, see Sec. 3.5.5, is about 95 percent). The fuel-air cycle pressures after combustion will be higher because complete combustion is assumed. In diesel engines, the combustion inefficiency is usually less, about 1 to 2 percent, so this effect is smaller.

SUMMARY. The effect of all these loss mechanisms on engine efficiency is best defined by an availability balance for the real engine cycle. A limited number of such calculations have been published (e.g., Refs. 8, 10, and 11). Table 5.4 shows the magnitude of the loss in availability (as a fraction of the initial availability) that occurs due to real cycle effects in a typical naturally aspirated diesel engine.[10] The combustion and exhaust losses are present in the ideal cycle models also (they are smaller, however[9]). The loss in availability due to heat losses, flow or aerodynamic losses, and mechanical friction are real engine effects.

Figure 5-19 shows standard and fuel-air cycle efficiencies as a function of the compression ratio compared with engine indicated efficiency data. The top three sets of engine data are for the best efficiency air/fuel ratio. Differences in the data are in part due to different fuels [(12) isooctane; (13) gasoline; (14) propane] which affect efficiency slightly through their different composition and heating values (see Table D.4). They also result from different combustion chamber shapes which affect the combustion rate and heat transfer. The trends in the data with increasing compression ratio and the $\phi = 0.8$ fuel-air cycle curve (which corresponds approximately to the actual air/fuel ratios used) are similar. The factor of 0.8 relating real engine and fuel-air cycle efficiencies holds roughly. At compression ratios above about 14, however, the data show that the indicated efficiency of actual engines is essentially constant. Increasing crevice and heat

TABLE 5.4
Availability losses in naturally aspirated diesel

Loss mechanism	Loss, fraction of fuel availability
Combustion	0.225
Exhaust	0.144
Heat transfer	0.135
Aerodynamic	0.047
Mechanical friction	0.048
Total losses	0.599
Availability conversion efficiency (brake)	0.401

Source: Traupel.[10]

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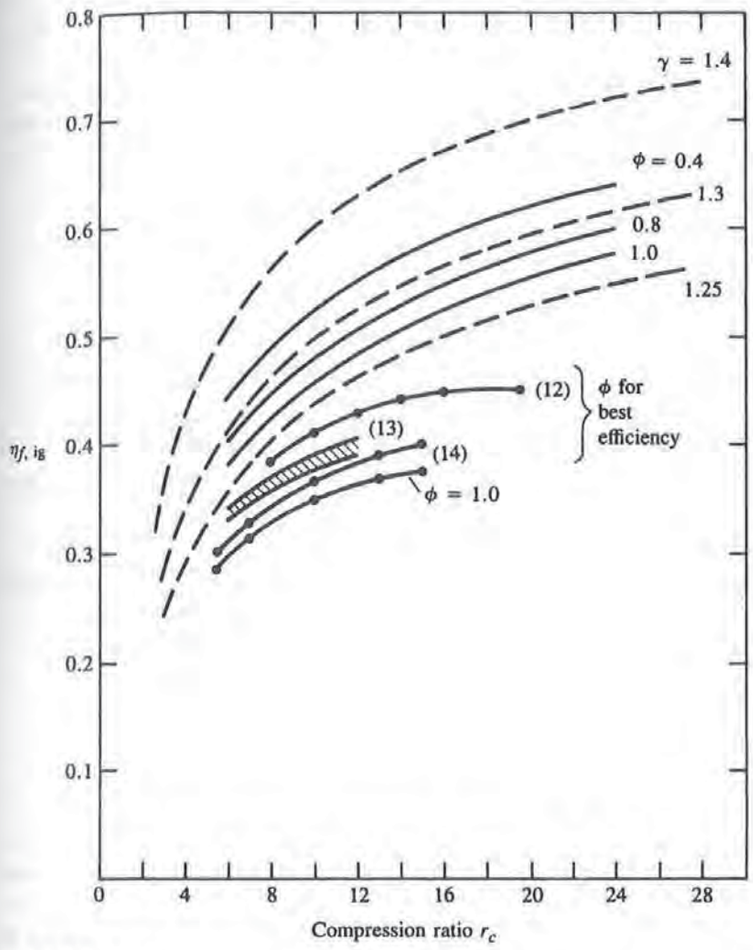


FIGURE 5-19
Indicated fuel conversion efficiency as a function of compression ratio for ideal gas constant-volume cycle (dashed lines, $\gamma = 1.25$, 1.3, 1.4) and fuel-air cycle (solid lines, $\phi = 0.4$, 0.8, 1.0). Also shown are available engine data for equivalence ratios given: best efficiency ϕ;[12–14] $\phi = 1$.[14]

losses offset the calculated ideal cycle efficiency increase as the compression ratio is raised above this value. The standard ideal gas cycle analysis results, with an appropriate choice for the value of γ (1.25 to 1.3), correspond closely to the fuel-air cycle analysis results.

The ideal cycle provides a convenient but crude approximation to the real engine operating cycle. It is useful for illustrating the thermodynamic aspects of engine operation. It can also provide approximate estimates of trends as major engine parameters change. The weakest link in these ideal cycles is the modeling of the combustion processes in SI and CI engines. None of the models examined in this chapter are sufficiently close to reality to provide accurate predictions of engine performance. More sophisticated models of the spark-ignition and diesel engine operating cycles have been developed and are the subject of Chap. 14.

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PROBLEMS

5.1. Many diesel engines can be approximated by a limited-pressure cycle. In a limited-pressure cycle, a fraction of the fuel is burnt at constant volume and the remaining fuel is burnt at constant pressure. Use this cycle approximation with $\gamma = c_p/c_v = 1.3$ to analyze the following problem:

Inlet conditions:	$p_1 = 1.0$ bar, $T_1 = 289$ K
Compression ratio:	15 : 1
Heat added during combustion:	43,000 kJ/kg of fuel
Overall fuel/air ratio:	0.045 kg fuel/kg air

(*a*) Half of the fuel is burnt at constant volume, then half at constant pressure. Draw a p-V diagram and compute the fuel conversion efficiency of the cycle.

(*b*) Compare the efficiency and peak pressure of the cycle with the efficiency and peak pressure that would be obtained if all of the fuel were burnt at constant pressure or at constant volume.

5.2. It is desired to increase the output of a spark-ignition engine by either (1) raising the compression ratio from 8 to 10 or (2) increasing the inlet pressure from 1.0 atm to 1.5 atm. Using the constant-volume cycle as a model for engine operation, which procedure will give:

(*a*) The highest pressure of the cycle?

(*b*) The highest efficiency?

(*c*) The highest mep?

Assume $\gamma = 1.3$ and $(m_f Q_{HV})/(mc_v T_1) = 9.3(r_c - 1)/r_c$.

5.3. When a diesel engine, originally designed to be naturally aspirated, is turbocharged the fuel/air equivalence ratio ϕ at full load must be reduced to maintain the maximum cylinder pressure essentially constant. If the naturally aspirated engine was designed for $\phi = 0.75$ at full load, estimate the maximum permissible value of ϕ for the turbocharged engine at full load if the air pressure at the engine inlet is 1.6 atm. Assume that the engine can be modeled with the limited-pressure cycle, with half the injected fuel burned at constant volume and half at constant pressure. The compression ratio is 16. The fuel heating value is 42.5 MJ/kg fuel. Assume $\gamma = c_p/c_v = 1.35$, that the air temperature at the start of compression is 325 K, and $(F/A)_{stoich} = 0.0666$.

5.4. A spark-ignition engine is throttled when operating at part load (the inlet pressure is reduced) while the fuel/air ratio is held essentially constant. Part-load operation of the engine is modeled by the cycle shown in Fig. 5-2*d*; the inlet air is at pressure p_1, the exhaust pressure is atmospheric p_a, and the ambient temperature is T_a. Derive an expression for the *decrease* in net indicated fuel conversion efficiency due to throttling from the ideal constant-volume cycle efficiency and show that it is proportional to $(p_a/p_1 - 1)$. Assume mass fuel $\ll$ mass air.

5.5. (*a*) Use the ideal gas cycle with constant-volume combustion to describe the operation of an SI engine with a compression ratio of 9. Find the pressure and temperature at points 2, 3, 4, and 5 on Fig. 5-2a. Assume a pressure of 100 kPa and a temperature of 320 K at point 1. Assume $m_f/m = 0.06$, $c_v = 946$ J/kg·K, $\gamma = 1.3$. Q_{LHV} for gasoline is 44 MJ/kg.

(*b*) Find the indicated fuel conversion efficiency and imep for this engine under these operating conditions.

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5.6. Use a limited-pressure cycle analysis to obtain a plot of indicated fuel conversion efficiency versus p_3/p_1 for a compression ratio of 15 with light diesel oil as fuel. Assume $m_f/m = 0.04$, $T_1 = 45°C$. Use $\gamma = 1.3$ and $c_v = 946$ J/kg·K.

5.7. Explain why constant-volume combustion gives a higher indicated fuel conversion efficiency than constant-pressure combustion for the same compression ratio.

5.8. Two engines are running at a bmep of 250 kPa. One is an SI engine with the throttle partially closed to maintain the correct load. The second engine is a naturally aspirated CI engine which requires no throttle. Mechanical friction mep for both engines is 100 kPa. If the intake manifold pressures for the SI and CI engines are 25 kPa and 100 kPa respectively, and both exhaust manifold pressures are 105 kPa, use an ideal cycle model to estimate and compare the *gross* imep of the two engines. You may neglect the pressure drop across the valves during the intake and exhaust processes.

5.9. (*a*) Plot net imep versus p_i for 20 kPa $< p_i <$ 100 kPa for a constant-volume cycle using the following conditions: $m_f/m = 0.06$, $T_1 = 40°C$, $c_v = 946$ J/kg·K, $\gamma = 1.3$, $r_c = 9.5$, $Q_{LHV} = 44$ MJ/kg fuel. Assume $p_e = 100$ kPa.

(*b*) What additional information is necessary to draw a similar plot for the engine's indicated torque, and indicated power?

5.10. (*a*) Draw a diagram similar to those in Fig. 5-2 for a supercharged cycle with constant-pressure combustion.

(*b*) Use the ideal gas cycle with constant-pressure combustion to model an engine with a compression ratio of 14 through such a supercharged cycle. Find the pressure and temperature at points corresponding to 2, 3, 4, and 5 in Fig. 5-2. Assume a pressure of 200 kPa and temperature of 325 K at point 1, and a pressure of 100 kPa at points 5 and 6. $m_f/m = 0.03$ and the fuel is a light diesel oil.

(*c*) Calculate the gross and net indicated fuel conversion efficiency and imep for this engine under these operating conditions.

5.11. Use the appropriate tables and charts to carry out a constant-pressure fuel-air cycle calculation for the supercharged engine described in Prob. 5.10. Assume the same initial conditions at point 1, with $\phi = 0.4$ and a residual gas fraction of 0.025. A single cycle calculation is sufficient.

(*a*) Determine the pressure and temperature at points 2, 3, 4, and 5. Calculate the compression stroke, expansion stroke, and pumping work per cycle per kg air.

(*b*) Find the gross and net indicated fuel conversion efficiency and imep.

(*c*) Compare the calculated residual gas fraction with the assumed value of 0.025.

5.12. One method proposed for reducing the pumping work in throttled spark-ignition engines is *early intake valve closing* (EIVC). The ideal cycle *p-V* diagram shown illustrates the concept. The EIVC cycle is 1-2-3-4-5-6-7-8-1 (the conventional throttled cycle is 1-2-3-4-5-6-7*-1). With EIVC, the inlet manifold is held at a pressure p_i (which is higher than the normal engine intake pressure, p_i^*), and the inlet valve is closed *during* the inlet stroke at 8. The trapped fresh charge and residual is then *expanded* to the normal cycle (lower) intake pressure, p_i^*. You can assume that both cycles have the same mass of gas in the cylinder, temperature, and pressure at state 1 of the cycle.

(*a*) On a sketch of the intake and exhaust process *p-V* diagram, shade in the area that corresponds to the difference between the pumping work of the EIVC cycle and that of the normal cycle.

(*b*) What value of p_i and V_{EIVC} will give the maximum reduction in pumping work for the EIVC cycle.

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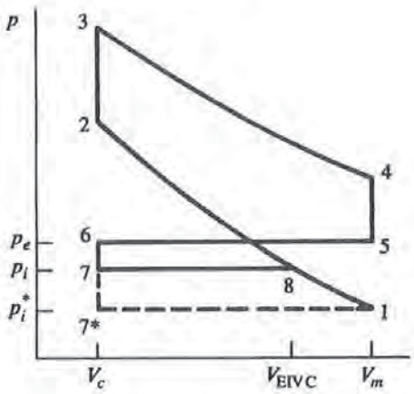


FIGURE P5-12

(*c*) Derive an expression for this maximum difference in pumping work between the normal cycle and the EIVC cycle in terms of p_e, p_i^*, V_c, and V_m. You can make the appropriate ideal cycle assumptions.

5.13. Calculate the following parameters for a constant-volume fuel-air cycle (Fig. 5-2*a*):

(*a*) The pressures and temperatures at states 1, 2, 3, 4, 5, and 6

(*b*) The indicated fuel conversion efficiency

(*c*) The imep

(*d*) The residual fraction

(*e*) The volumetric efficiency

Inlet pressure = 1 atm, exhaust pressure = 1 atm, inlet temperature = 300 K, compression ratio = 8 : 1, equivalence ratio = $\phi = 1$.

Calculate the above parameters (points *a*–*e*) using the SI units charts. Use 44.4 MJ/kg for heating value of the fuel. *Hint:* Start the calculations using the residual mass fraction 0.03 and the residual gas temperature 1370 K.

5.14. The cycle 1-2-3-4-5-6-1 is a conventional constant-volume fuel-air cycle with a compression ratio of 8. The fuel is isooctane, C_8H_{18}, with a lower heating value of 44.4 MJ/kg. The gas state at 1 is $T_1 = 300$ K, $p_1 = 1$ atmosphere with an equivalence ratio of 1.0 and zero residual fraction. The specific volume at state 1 is 0.9 m^3/kg air in the mixture. The temperature at the end of compression at state 2 is 600 K.

(*a*) Find the indicated fuel conversion efficiency and mean effective pressure of this fuel-air cycle model of a spark-ignition engine.

(*b*) The efficiency of the cycle can be increased by increasing the expansion ratio r_e while maintaining the same compression ratio r_c (cycle 1-2-3-4*A*-5*A*-6-1). (This

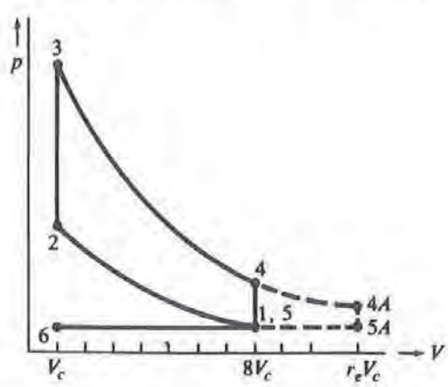


FIGURE P5-14

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can be done with valve timing.) If the expansion ratio r_e is 12, while the compression ratio and other details of the cycle remain the same as in (*a*), what is the indicated efficiency and mean effective pressure (based on the new, larger, displaced volume) of this new engine cycle?

5.15. In spark-ignition engines, exhaust gas is recycled to the intake at part load to reduce the peak burned gas temperatures and lower emissions of nitrogen oxides.

(*a*) Calculate the reduction in burned gas temperature that occurs when, due to exhaust gas recycle, the burned gas fraction in the unburned gas mixture (x_b) inside the cylinder is increased from 10 percent (the normal residual fraction) to 30 percent. Assume combustion occurs at top-center, at constant volume, and is adiabatic. Conditions at the end of compression for both cases are: $T = 700$ K, $p = 1000$ kPa, $v = 0.2$ m^3/kg air in the original mixture; the equivalence ratio is 1.0. The fuel can be modeled as isooctane.

(*b*) The compression ratio is 8. The compression stroke work is 300 kJ/kg air in the original mixture. Find the indicated work per cycle for the compression and expansion strokes, per kilogram of air in the original mixture, for these two cases.

(*c*) Briefly explain how you would increase the work per cycle with 30 percent burned gas fraction in the unburned mixture to the value obtained with 10 percent burned gas fraction, with fixed engine geometry. (A qualitative answer, only, is required here.)

5.16. The following cycle has been proposed for improving the operation of a four-stroke cycle engine. Its aim is to expand the postcombustion cylinder gases to a lower pressure and temperature by extending the expansion stroke, and hence extract more work per cycle.

The cycle consists of: (1) an intake stroke; (2) a compression stroke, where the inlet valve remains open (and the cylinder pressure is constant) for the first portion of the stroke; (3) a combustion process, which occurs rapidly close to top-center; (4) an expansion stroke, where the exhaust valve remains closed until the end of the stroke; (5) an exhaust stroke, where the cylinder pressure blows down to the exhaust pressure rapidly and most of the remaining combustion products are expelled as the piston moves from the BC to the TC position. Thus, for this engine concept, the compression ratio r_c (ratio of cylinder volume at inlet valve closing to clearance volume) is less than the expansion ratio r_e (ratio of cylinder volume at exhaust valve opening to clearance volume).

(*a*) Sketch a p-V diagram for the cylinder gases for this cycle operating unthrottled.

(*b*) Using the charts in SI units developed for fuel-air cycle calculations, carry out an analysis of an appropriate ideal model for this cycle where the compression ratio r_c is 8 and the expansion ratio r_e is (1) 8; (2) 16. Assume the following:

Pressure in the cylinder at inlet valve close 1 atm
Mixture temperature at inlet valve close 300 K
Mixture equivalence ratio = 1.0
Fuel: isooctane C_8H_{18}
Lower heating value = 44.4 MJ/kg
Residual gas mass fraction at inlet valve close 0.05
Stoichiometric fuel/air ratio = 0.066

Calculate the indicated work per cycle per kg of air in the original mixture (the standard chart units) and the indicated mean effective pressure for these two expansion ratios. Base the mean effective pressure on the volume displaced by

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the piston during the *expansion* stroke. Tabulate your answers. (*Note:* You are given the initial conditions for the cycle calculation; changing the value of r_e requires only modest changes in the cycle calculation.)

(*c*) Comment briefly on the effect of increasing the ratio r_e/r_c above 1.0 with this concept on engine efficiency and specific power (power per unit engine weight). Additional calculations are not required.

5.17. In a direct-injection stratified-charge (DISC) engine fuel is injected into the engine cylinder just before top-center (like a diesel); a spark discharge is then used to initiate the combustion process. A four-stroke cycle version of this engine has a displaced volume of 2.5 liters and a compression ratio of 12. At high load, the inlet pressure is boosted by a compressor to above atmospheric pressure. The compressor is geared directly to the engine drive shaft. The exhaust pressure is 1 atm. This DISC engine is to replace an equal displacement conventional naturally aspirated spark-ignition (SI) engine, which has a compression ratio of 8.

(*a*) Draw qualitative sketches of the appropriate constant-volume ideal cycle pressure-volume diagrams for the complete operating cycles for these two engines at maximum load.

(*b*) Use available fuel-air results to estimate how much the DISC engine inlet pressure must be boosted above atmospheric pressure by the compressor to provide the same maximum gross indicated power as the naturally aspirated SI engine. The SI engine operates with an equivalence ratio of 1.2; the DISC engine is limited by smoke emissions to a maximum equivalence ratio of 0.7.

(*c*) Under these conditions, will the brake powers of these engines be the same, given that the mechanical rubbing friction is the same? Briefly explain.

(*d*) At part load, the SI engine operates at an equivalence ratio of 1.0 and inlet pressure of 0.5 atm. At part load the DISC engine has negligible boost and operates with an inlet pressure of 1.0 atm. Use fuel-air cycle results to determine the equivalence ratio at which the DISC engine must be operated to provide the same net indicated mean effective pressure as the SI engine. What is the ratio of DISC engine net indicated fuel conversion efficiency to SI engine efficiency at these conditions?

5.18. The earliest successful reciprocating internal combustion engine was an engine developed by Lenoir in the 1860s. The operating cycle of this engine consisted of two strokes (i.e., one crankshaft revolution). During the first half of the first stroke, as the piston moves away from its top-center position, fuel-air mixture is drawn into the cylinder through the inlet valve. When half the total cylinder volume is filled with fresh mixture, the inlet valve is closed. The mixture is then ignited and burns rapidly. During the second half of the first stroke, power is delivered from the high-pressure burned gases to the piston. With the piston in its bottom-center position, the exhaust valve is opened. The second stroke, the exhaust stroke, completes the cycle as the piston returns to top-center.

(*a*) Sketch a cylinder pressure versus cylinder volume diagram for this engine.

(*b*) Using the charts in SI units developed for fuel-air cycle calculations, carry out a cycle analysis and determine the indicated fuel conversion efficiency and mean effective pressure for the Lenoir engine. Assume the following:

Inlet pressure = 1 atm
Inlet mixture temperature = 300 K
Mixture equivalence ratio = 1.0

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Fuel: isooctane C_8H_{18}
Lower heating value = 44.4 MJ/kg
Clearance volume negligible

(*c*) Compare these values with typical values for the constant-volume fuel-air cycle. Explain (with thermodynamic arguments) why the two cycles have such different indicated mean effective pressures and efficiencies.
(*d*) Explain briefly why the real Lenoir engine would have a lower efficiency than the value you calculated in (*b*) (the actual *brake* fuel conversion efficiency of the engine was about 5 percent).

5.19. Estimate from fuel-air cycle results the indicated fuel conversion efficiency, the indicated mean effective pressure, and the maximum indicated power (in kilowatts) at wide-open throttle of these two four-stroke cycle spark-ignition engines:

A six-cylinder engine with a 9.2-cm bore, 9-cm stroke, compression ratio of 7, operated at an equivalence ratio of 0.8
A six-cylinder engine with an 8.3-cm bore, 8-cm stroke, compression ratio of 10, operated at an equivalence ratio of 1.1

Assume that actual indicated engine efficiency is 0.8 times the appropriate fuel-air cycle efficiency. The inlet manifold pressure is close to 1 atmosphere. The maximum permitted value of the mean piston speed is 15 m/s. Briefly summarize the reasons why:
(*a*) The efficiency of these two engines is approximately the same despite their different compression ratios.
(*b*) The maximum power of the smaller displacement engine is approximately the same as that of the larger displacement engine.

5.20. The constant-volume combustion fuel-air cycle model can be used to estimate the effect of changes in internal combustion engine design and operating variables on engine efficiency. The following table gives the major differences between a diesel and a spark-ignition engine both operating at half maximum power.

	Diesel engine	**Spark-ignition engine**
Compression ratio	16 : 1	9 : 1
Fuel/air equivalence ratio	0.4	1.0
Inlet manifold pressure	1 atm	0.5 atm

(*a*) Use the graphs of fuel-air cycle results (Figs. 5-9 and 5-10) to estimate the ratio of the diesel engine *brake* fuel conversion efficiency to the spark-ignition engine *brake* fuel conversion efficiency.
(*b*) Estimate what percentage of the higher diesel brake fuel conversion efficiency comes from:
(1) The higher diesel compression ratio
(2) The leaner diesel equivalence ratio
(3) The lack of intake throttling in the diesel compared with the spark-ignition engine

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The values of fuel conversion efficiency and mean effective pressure given in the graphs are *gross* indicated values (i.e., values obtained from $\int p\,dV$ over the compression and expansion strokes only).

You may assume, if necessary, that for the real engines, the gross indicated efficiency and gross indicated mean effective pressure are 0.8 times the fuel-air cycle values. Also, the mechanical rubbing friction for each engine is 30 percent of the *net* indicated power or mep.

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CHAPTER 6

GAS EXCHANGE PROCESSES

This chapter deals with the fundamentals of the gas exchange processes—intake and exhaust in four-stroke cycle engines and scavenging in two-stroke cycle engines. The purpose of the exhaust and inlet processes or of the scavenging process is to remove the burned gases at the end of the power stroke and admit the fresh charge for the next cycle. Equation (2.38) shows that the indicated power of an internal combustion engine at a given speed is proportional to the mass flow rate of air. Thus, inducting the maximum air mass at wide-open throttle or full load and retaining that mass within the cylinder is the primary goal of the gas exchange processes. Engine gas exchange processes are characterized by overall parameters such as volumetric efficiency (for four-stroke cycles), and scavenging efficiency and trapping efficiency (for two-stroke cycles). These overall parameters depend on the design of engine subsystems such as manifolds, valves, and ports, as well as engine operating conditions. Thus, the flow through individual components in the engine intake and exhaust system has been extensively studied also. Supercharging and turbocharging are used to increase air flow through engines, and hence power density. Obviously, whether the engine is naturally aspirated or supercharged (or turbocharged) significantly affects the gas exchange processes. The above topics are the subject of this chapter.

For spark-ignition engines, the fresh charge is fuel, air, and (if used for emission control) recycled exhaust, so mixture preparation is also an important

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goal of the intake process. Mixture preparation includes both achieving the appropriate mixture composition and achieving equal distribution of air, fuel, and recycled exhaust amongst the different cylinders. In diesels, only air (or air plus recycled exhaust) is inducted. Mixture preparation and manifold flow phenomena are discussed in Chap. 7. A third goal of the gas exchange processes is to set up the flow field within the engine cylinders that will give a fast-enough combustion process for satisfactory engine operation. In-cylinder flows are the subject of Chap. 8.

6.1 INLET AND EXHAUST PROCESSES IN THE FOUR-STROKE CYCLE

In a spark-ignition engine, the intake system typically consists of an air filter, a carburetor and throttle or fuel injector and throttle or throttle with individual fuel injectors in each intake port, and intake manifold. During the induction process, pressure losses occur as the mixture passes through or by each of these components. There is an additional pressure drop across the intake port and valve. The exhaust system typically consists of an exhaust manifold, exhaust pipe, often a catalytic converter for emission control, and a muffler or silencer. Figure 6-1 illustrates the intake and exhaust gas flow processes in a conventional spark-ignition engine. These flows are pulsating. However, many aspects of these flows can be analysed on a quasi-steady basis, and the pressures indicated in the intake system in Fig. 6-1*a* represent time-averaged values for a multicylinder engine.

The drop in pressure along the intake system depends on engine speed, the flow resistance of the elements in the system, the cross-sectional area through which the fresh charge moves, and the charge density. Figure 6-1*d* shows the inlet and exhaust valve lifts versus crank angle. The usual practice is to extend the valve open phases beyond the intake and exhaust strokes to improve emptying and charging of the cylinders and make the best use of the inertia of the gases in the intake and exhaust systems. The exhaust process usually begins 40 to 60° before BC. Until about BC the burned cylinder gases are discharged due to the pressure difference between the cylinder and the exhaust system. After BC, the cylinder is scavenged by the piston as it moves toward TC. The terms *blowdown* and *displacement* are used to denote these two phases of the exhaust process. Typically, the exhaust valve closes 15 to 30° after TC and the inlet valve opens 10 to 20° before TC. Both valves are open during an *overlap period*, and when $p_i/p_e < 1$, backflow of exhausted gas into the cylinder and of cylinder gases into the intake will usually occur. The advantage of valve overlap occurs at high engine speeds when the longer valve-open periods improve volumetric efficiency. As the piston moves past TC and the cylinder pressure falls below the intake pressure, gas flows from the intake into the cylinder. The intake valve remains open until 50 to 70° after BC so that fresh charge may continue to flow into the cylinder after BC.

In a diesel engine intake system, the carburetor or EFI system and the throttle plate are absent. Diesel engines are more frequently turbocharged. A

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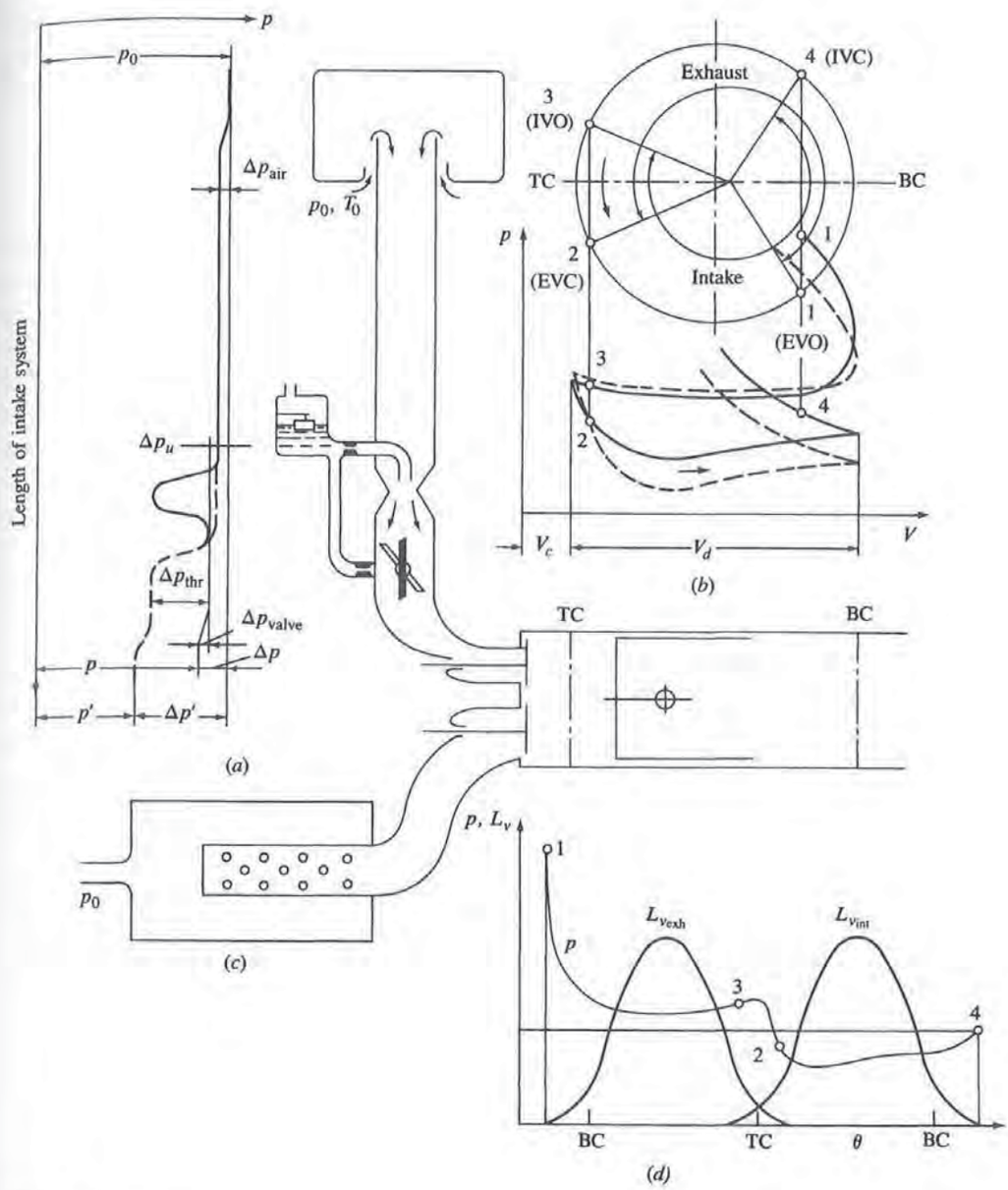


FIGURE 6-1
Intake and exhaust processes for four-stroke cycle spark-ignition engine: (*a*) intake system and average pressures within it; (*b*) valve timing and pressure-volume diagrams; (*c*) exhaust system; (*d*) cylinder pressure p and valve lift L_v versus crank angle θ. Solid lines are for wide-open throttle, dashed lines for part throttle; p_0, T_0, atmospheric conditions; Δp_{air} = pressure losses in air cleaner; Δp_u = intake losses upstream of throttle; Δp_{thr} = losses across throttle; Δp_{valve} = losses across the intake valve.[1]

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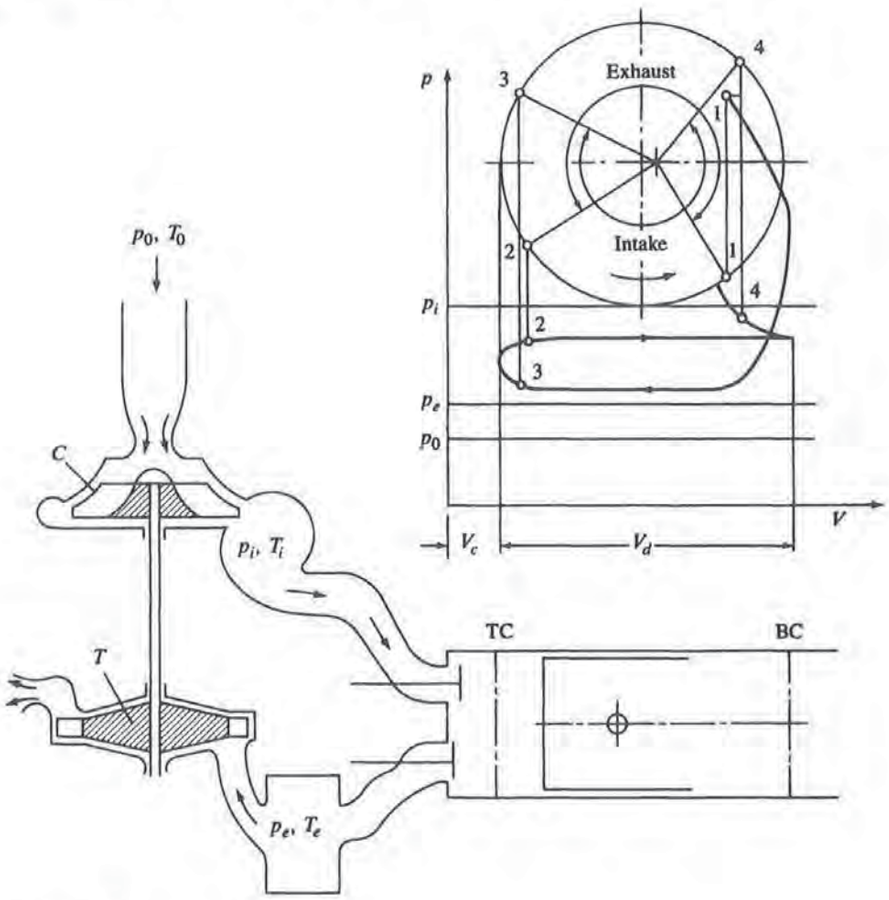


FIGURE 6-2
Intake and exhaust process for turbocharged four-stroke cycle engine. The turbocharger compressor C raises air pressure and temperature from ambient p_0, T_0 to p_i, T_i. Cylinder pressure during intake is less than p_i. During exhaust, the cylinder gases flow through the exhaust manifold to the turbocharger turbine T. Manifold pressure p_e may vary during the exhaust process and lies between cylinder pressure and ambient.[1]

similar set of diagrams illustrating the intake and exhaust processes for a turbocharged four-stroke diesel is shown in Fig. 6-2. When the exhaust valve opens, the burned cylinder gases are fed to a turbine which drives a compressor which compresses the air prior to entry to the cylinder.

Due to the time-varying valve open area and cylinder volume, gas inertia effects, and wave propagation in the intake and exhaust systems, the pressures in the intake, the cylinder, and the exhaust during these gas exchange processes vary in a complicated way. Analytical calculation of these processes is difficult (see Secs. 7.6.2 and 14.3 for a review of available methods). In practice, these processes are often treated empirically using overall parameters such as volumetric efficiency to define intake and exhaust system performance.

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6.2 VOLUMETRIC EFFICIENCY

Volumetric efficiency is used as an overall measure of the effectiveness of a four-stroke cycle engine and its intake and exhaust systems as an air pumping device. It is defined [see Sec. 2.10, Eq. (2.27)] as

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,0} V_d N} \tag{6.1}$$

The air density $\rho_{a,0}$ can be evaluated at atmospheric conditions; η_v is then the overall volumetric efficiency. Or it can be evaluated at inlet manifold conditions; η_v then measures the pumping performance of the cylinder, inlet port, and valve alone. This discussion will cover unthrottled (wide-open throttle) engine operation only. It is the air flow under these conditions that constrains maximum engine power. Lesser air flows in SI engines are obtained by restricting the intake system flow area with the throttle valve.

Volumetric efficiency is affected by the following fuel, engine design, and engine operating variables:

1. Fuel type, fuel/air ratio, fraction of fuel vaporized in the intake system, and fuel heat of vaporization
2. Mixture temperature as influenced by heat transfer
3. Ratio of exhaust to inlet manifold pressures
4. Compression ratio
5. Engine speed
6. Intake and exhaust manifold and port design
7. Intake and exhaust valve geometry, size, lift, and timings

The effects of several of the above groups of variables are essentially quasi steady in nature; i.e., their impact is either independent of speed or can be described adequately in terms of mean engine speed. However, many of these variables have effects that depend on the unsteady flow and pressure wave phenomena that accompany the time-varying nature of the gas exchange processes.

6.2.1 Quasi-Static Effects

VOLUMETRIC EFFICIENCY OF AN IDEAL CYCLE. For the ideal cycles of Fig. 5-2*d* and *e*, an expression for volumetric efficiency can be derived which is a function of the following variables: intake mixture pressure p_i, temperature T_i, and fuel/air ratio (F/A); compression ratio r_c; exhaust pressure p_e; and molecular weight M and γ for the cycle working fluid. The overall volumetric efficiency is

$$\eta_v = \frac{m_a}{\rho_{a,0} V_d} = \frac{m(1 - x_r)}{\rho_{a,0}[1 + (F/A)]} \frac{r_c}{(r_c - 1)V_1}$$

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where m is the mass in the cylinder at point 1 in the cycle. Since

$$p_i V_1 = m \frac{\tilde{R}}{M} T_1 \qquad \text{and} \qquad p_{a,0} = \rho_{a,0} \frac{\tilde{R}}{M_a} T_{a,0}$$

and Eq. (5.38) relates T_1 to T_i, the above expression for η_v can be written

$$\eta_v = \left(\frac{M}{M_a}\right)\left(\frac{p_i}{p_{a,0}}\right)\left(\frac{T_{a,0}}{T_i}\right) \frac{1}{[1 + (F/A)]} \left\{\frac{r_c}{r_c - 1} - \frac{1}{\gamma(r_c - 1)}\left[\left(\frac{p_e}{p_i}\right) + (\gamma - 1)\right]\right\} \tag{6.2}$$

For $(p_e/p_i) = 1$, the term in { } is unity.

EFFECT OF FUEL COMPOSITION, PHASE, AND FUEL/AIR RATIO. In a spark-ignition engine, the presence of gaseous fuel (and water vapor) in the intake system reduces the air partial pressure below the mixture pressure. For mixtures of air, water vapor, and gaseous or evaporated fuel we can write the intake manifold pressure as the sum of each component's partial pressure:

$$p_i = p_{a,i} + p_{f,i} + p_{w,i}$$

which with the ideal gas law gives

$$\frac{p_{a,i}}{p_i} = \left[1 + \left(\frac{\dot{m}_f}{\dot{m}_a}\right)\left(\frac{M_a}{M_f}\right) + \left(\frac{\dot{m}_w}{\dot{m}_a}\right)\left(\frac{M_a}{M_w}\right)\right]^{-1} \tag{6.3}$$

The water vapor correction is usually small (≤ 0.03). This ratio, $p_{a,i}/p_i$, for several common fuels as a function of $(\dot{m}_f/\dot{m}_a)$ is shown in Fig. 6-3. Note that $(\dot{m}_f/\dot{m}_a)$ only equals the engine operating fuel/air ratio if the fuel is fully vaporized.

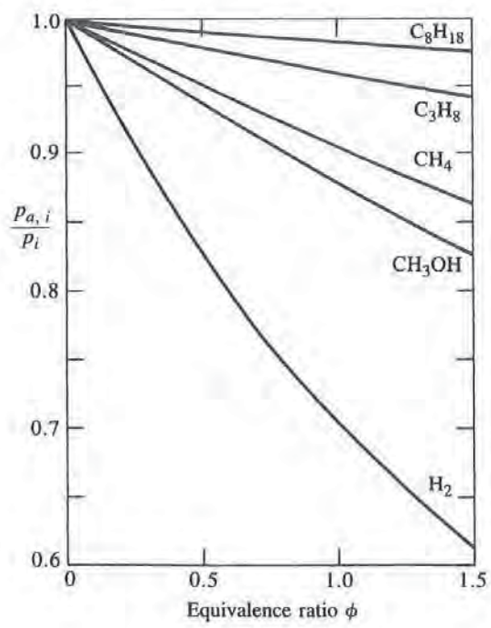


FIGURE 6-3
Effect of fuel (vapor) on inlet air partial pressure. Ratio of air inlet pressure $p_{a,i}$ to mixture inlet pressure p_i versus fuel/air equivalence ratio ϕ for isooctane vapor, propane, methane, methanol vapor, and hydrogen.

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For conventional liquid fuels such as gasoline the effect of fuel vapor, and therefore fuel/air ratio, is small. For gaseous fuels and for methanol vapor, the volumetric efficiency is significantly reduced by the fuel vapor in the intake mixture.

FRACTION FUEL VAPORIZED, HEAT OF VAPORIZATION, AND HEAT TRANSFER. For a constant-pressure flow with liquid fuel evaporation and with heat transfer, the steady-flow energy equation is

$$[\dot{m}_a h_a + (1 - x_e)\dot{m}_f h_{f,L} + x_e \dot{m}_f h_{f,V}]_A = \dot{Q} + (\dot{m}_a h_a + \dot{m}_f h_{f,L})_B \qquad (6.4)$$

where x_e is the mass fraction evaporated and the subscripts denote: a, air properties; f, fuel properties; L, liquid; V, vapor; B before evaporation; A after evaporation. Approximating the change in enthalpy per unit mass of each component of the mixture by $c_p \Delta T$, and with $h_{f,V} - h_{f,L} = h_{f,LV}$ (the enthalpy of vaporization), Eq. (6-4) becomes

$$T_A - T_B = \frac{(\dot{Q}/\dot{m}_a) - x_e(F/A)h_{f,LV}}{c_{p,a} + (F/A)c_{f,L}} \qquad (6.5)$$

Since $c_{f,L} \approx 2c_{p,a}$ the last term in the denominator can often be neglected.

If no heat transfer to the inlet mixture occurs, the mixture temperature decreases as liquid fuel is vaporized. For complete evaporation of isooctane, with $\phi = 1.0$, $T_A - T_B = -19°C$. For methanol under the same conditions, $T_A - T_B$ would be $-128°C$. In practice heating occurs; also, the fuel is not necessarily fully evaporated prior to entry to the cylinder. Experimental data show that the decrease in air temperature that accompanies liquid fuel evaporation more than offsets the reduction in air partial pressure due to the increased amount of fuel vapor: for the same heating rate, volumetric efficiency with fuel vaporization is higher by a few percent.[2]

The ideal cycle equation for volumetric efficiency [Eq. (6.2)] shows that the effect of gas temperature variations, measured at entry to the cylinder, is through the factor $(T_{a,0}/T_i)$. Engine test data indicate that a square root dependence of volumetric efficiency on temperature ratio is closer to real engine behavior. The square root dependence is a standard assumption in engine test data reduction (see Sec. 2.12).

EFFECT OF INLET AND EXHAUST PRESSURE RATIO AND COMPRESSION RATIO. As the pressure ratio (p_e/p_i) and the compression ratio are varied, the fraction of the cylinder volume occupied by the residual gas *at the intake pressure* varies. As this volume increases so volumetric efficiency decreases. These effects on ideal-cycle volumetric efficiency are given by the $\{\ \}$ term in Eq. (6.2). For $\gamma = 1.3$ these effects are shown in Fig. 6-4.

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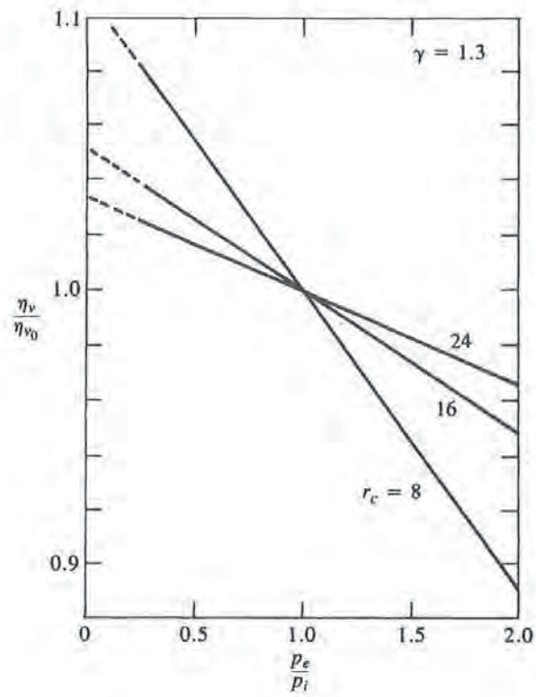


FIGURE 6-4
Effect of exhaust to inlet pressure ratio on ideal-cycle volumetric efficiency.

6.2.2 Combined Quasi-Static and Dynamic Effects

When gas flows unsteadily through a system of pipes, chambers, ports, and valves, both friction, pressure, and inertial forces are present. The relative importance of these forces depends on gas velocity and the size and shape of these passages and their junctions. Both quasi-steady and dynamic effects are usually significant. While the effects of changes in engine speed, and intake and exhaust manifold, port and valve design are interrelated, several separate phenomena which affect volumetric efficiency can be identified.

FRICTIONAL LOSSES. During the intake stroke, due to friction in each part of the intake system, the pressure in the cylinder p_c is less than the atmospheric pressure p_{atm} by an amount dependent on the square of the speed. This total pressure drop is the sum of the pressure loss in each component of the intake system: air filter, carburetor and throttle, manifold, inlet port, and inlet valve. Each loss is a few percent, with the port and valve contributing the largest drop. As a result, the pressure in the cylinder during the period in the intake process when the piston is moving at close to its maximum speed can be 10 to 20 percent lower than atmospheric. For each component in the intake (and the exhaust) system, Bernoulli's equation gives

$$\Delta p_j = \xi_j \rho v_j^2$$

where ξ_j is the resistance coefficient for that component which depends on its

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geometric details and v_j is the local velocity. Assuming the flow is quasi-steady, v_j is related to the mean piston speed $\bar{S}_p$ by

$$v_j A_j = \bar{S}_p A_p$$

where A_j and A_p are the component minimum flow area and the piston area, respectively. Hence, the total quasi-steady pressure loss due to friction is

$$p_{atm} - p_c = \sum \Delta p_j = \sum \xi_j \rho v_j^2 = \rho \bar{S}_p^2 \sum \xi_j \left(\frac{A_p}{A_j}\right)^2 \tag{6.6}$$

Equation (6.6) indicates the importance of large component flow areas for reducing frictional losses, and the dependence of these losses on engine speed. Figure 6-5 shows an example of the pressure losses due to friction across the air cleaner, carburetor, throttle, and manifold plenum of a standard four-cylinder

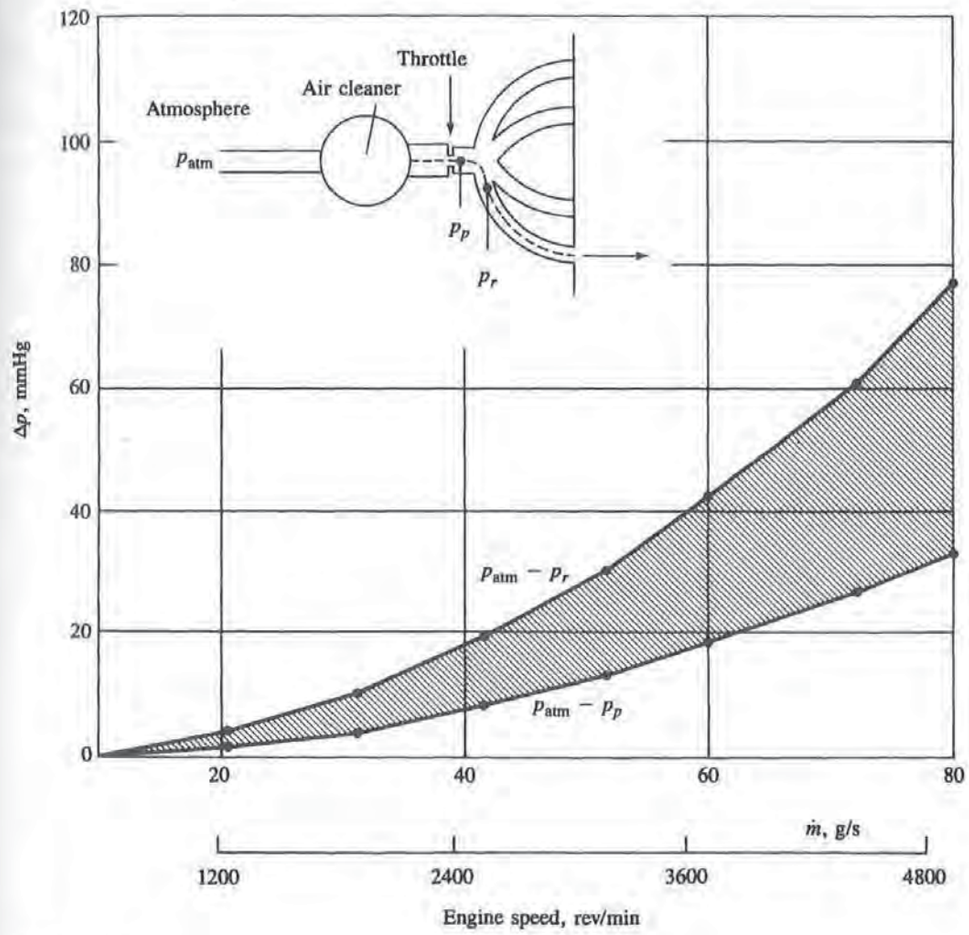


FIGURE 6-5
Pressure losses in the intake system of a four-stroke cycle spark-ignition engine determined under steady flow conditions.[3] Stroke = 89 mm. Bore = 84 mm.

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automobile engine intake system. These steady flow tests, conducted over the full engine speed range,[3] show that the pressure loss depends on speed squared.

Equivalent flow-dependent pressure losses in the exhaust system result in the exhaust port and manifold having average pressure levels that are higher than atmospheric. Figure 6-6 shows the time-averaged exhaust manifold gauge pressure as a function of inlet manifold vacuum (which varies inversely to load) and

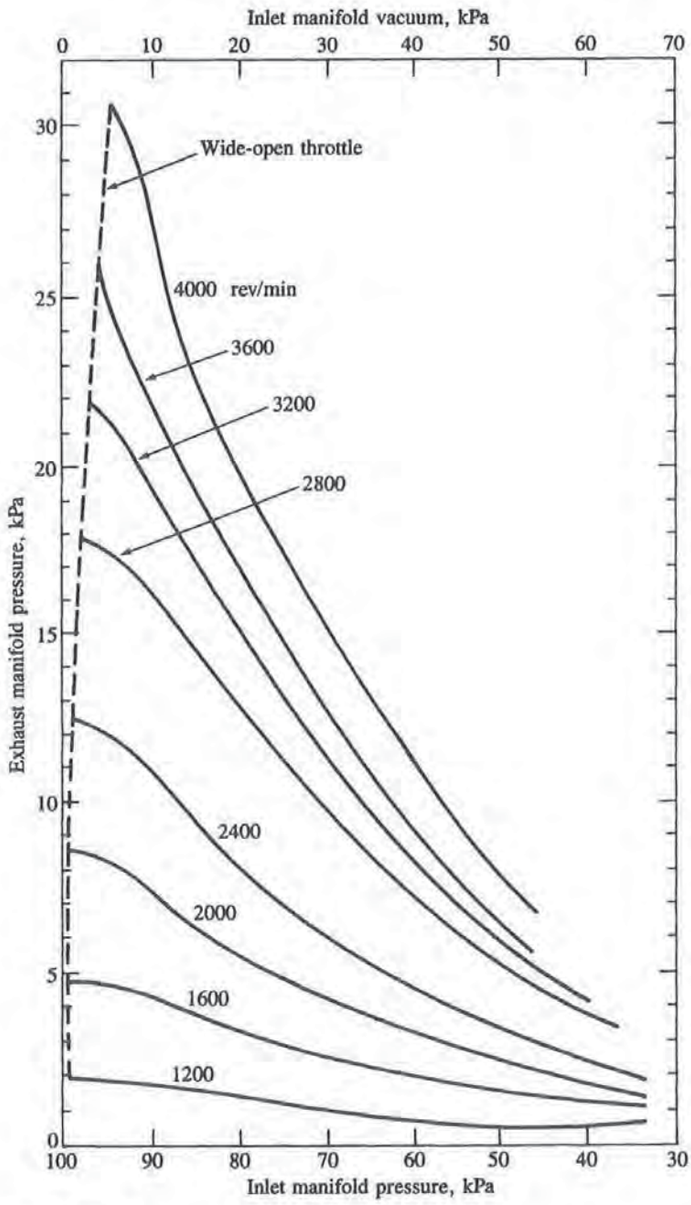


FIGURE 6-6
Exhaust manifold pressure as a function of load (defined by inlet manifold vacuum) and speed, four-stroke cycle four-cylinder spark-ignition engine.[4]

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speed for a four-cylinder automobile spark-ignition engine.[4] At high speeds and loads the exhaust manifold operates at pressures substantially above atmospheric.

RAM EFFECT. The pressure in the inlet manifold varies during each cylinder's intake process due to the piston velocity variation, valve open area variation, and the unsteady gas-flow effects that result from these geometric variations. The mass of air inducted into the cylinder, and hence the volumetric efficiency, is almost entirely determined by the pressure level in the inlet port during the short period before the inlet valve is closed.[5] At higher engine speeds, the inertia of the gas in the intake system as the intake valve is closing increases the pressure in the port and continues the charging process as the piston slows down around BC and starts the compression stroke. This effect becomes progressively greater as engine speed is increased. The inlet valve is closed some 40 to 60° after BC, in part to take advantage of this ram phenomenon.

REVERSE FLOW INTO THE INTAKE. Because the inlet valve closes after the start of the compression stroke, a reverse flow of fresh charge from the cylinder back into the intake can occur as the cylinder pressure rises due to piston motion toward TC. This reverse flow is largest at the lowest engine speeds. It is an inevitable consequence of the inlet valve closing time chosen to take advantage of the ram effect at high speeds.

TUNING. The pulsating flow from each cylinder's exhaust process sets up pressure waves in the exhaust system. These pressure waves propagate at the local sound speed relative to the moving exhaust gas. The pressure waves interact with the pipe junctions and ends in the exhaust manifold and pipe. These interactions cause pressure waves to be reflected back toward the engine cylinder. In multicylinder engines, the pressure waves set up by each cylinder, transmitted through the exhaust and reflected from the end, can interact with each other. These pressure waves may aid or inhibit the gas exchange processes. When they aid the process by reducing the pressure in the exhaust port toward the end of the exhaust process, the exhaust system is said to be *tuned*.[6]

The time-varying inlet flow to the cylinder causes expansion waves to be propagated back into the inlet manifold. These expansion waves can be reflected at the open end of the manifold (at the plenum) causing positive pressure waves to be propagated toward the cylinder. If the timing of these waves is appropriately arranged, the positive pressure wave will cause the pressure at the inlet valve at the end of the intake process to be raised above the nominal inlet pressure. This will increase the inducted air mass. Such an intake system is described as *tuned*.[6]

Methods which predict the unsteady flows in the intake and exhaust systems of internal combustion engines with good accuracy have been developed. These methods are complicated, however, so more detailed discussion is deferred to Chap. 14.

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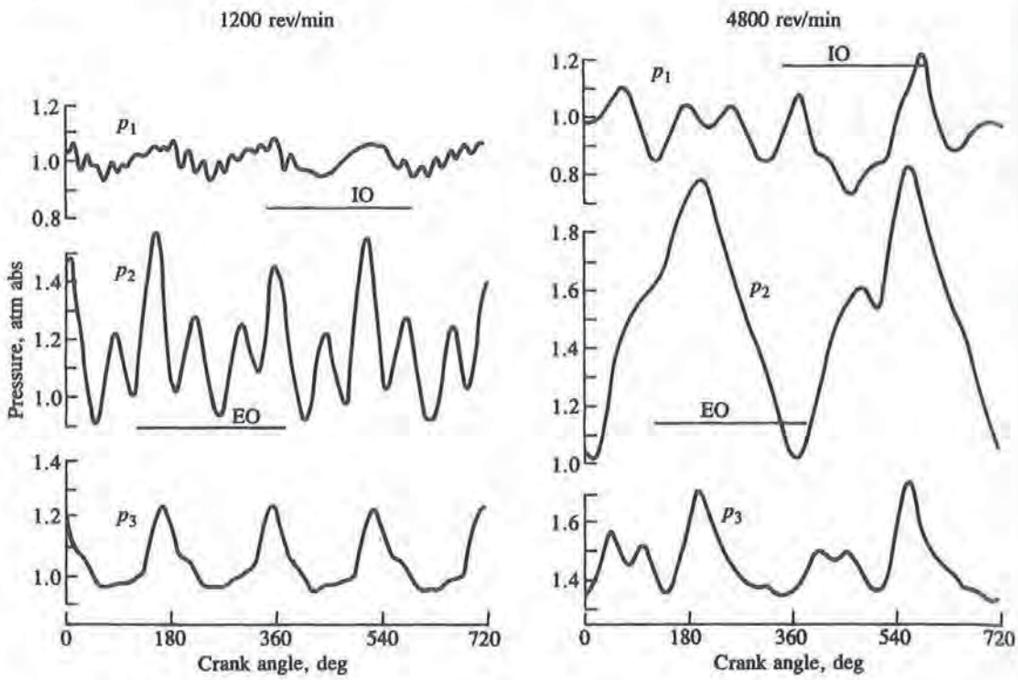


FIGURE 6-7
Instantaneous pressures in the intake and exhaust manifolds of a four-stroke cycle four-cylinder spark-ignition engine, at wide-open throttle. Locations: p_1, intake manifold runner 150 mm from cylinder 1; p_2, exhaust manifold runner 200 mm from cylinder 1; p_3, exhaust manifold runner 700 mm from cylinder 1. IO and EO, intake and exhaust valve open periods for that cylinder, respectively.[3] Stroke = 89 mm. Bore = 84 mm.

Examples of the pressure variations in the inlet and exhaust systems of a four-cylinder automobile spark-ignition engine at wide-open throttle are shown in Fig. 6-7. The complexity of the phenomena that occur is apparent. The amplitude of the pressure fluctuations increases substantially with increasing engine speed. The primary frequency in both the intake and exhaust corresponds to the frequency of the individual cylinder intake and exhaust processes. Higher harmonics that result from pressure waves in both the intake and exhaust are clearly important also.

6.2.3 Variation with Speed, and Valve Area, Lift, and Timing

Flow effects on volumetric efficiency depend on the velocity of the fresh mixture in the intake manifold, port, and valve. Local velocities for quasi-steady flow are equal to the volume flow rate divided by the local cross-sectional area. Since the intake system and valve dimensions scale approximately with the cylinder bore, mixture velocities in the intake system will scale with piston speed. Hence, volumetric efficiencies as a function of speed, for different engines, should be compared at the same mean piston speed.[7] Figure 6-8 shows typical curves of

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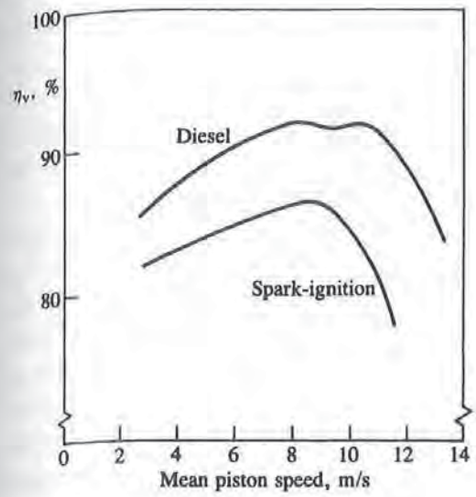


FIGURE 6-8
Volumetric efficiency versus mean piston speed for a four-cylinder automobile indirect-injection diesel[8] and a six-cylinder spark-ignition engine.[9]

volumetric efficiency versus mean piston speed for a four-cylinder automobile indirect-injection diesel engine[8] and a six-cylinder spark-ignition engine.[9] The volumetric efficiencies of spark-ignition engines are usually lower than diesel values due to flow losses in the carburetor and throttle, intake manifold heating, the presence of fuel vapor, and a higher residual gas fraction. The diesel curve with its double peak shows the effect of intake system tuning.

The shape of these volumetric efficiency versus mean piston speed curves can be explained with the aid of Fig. 6-9. This shows, in schematic form, how the

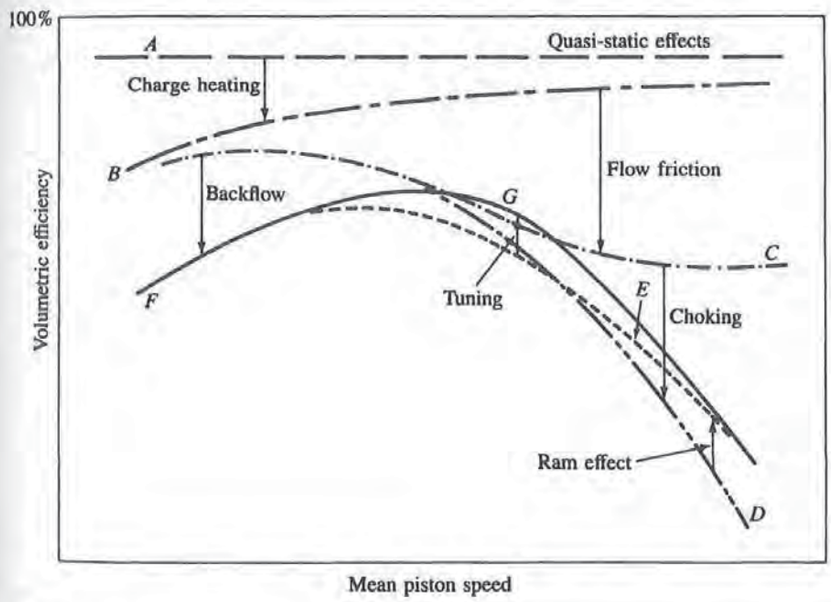


FIGURE 6-9
Effect on volumetric efficiency of different phenomena which affect the air flow rate as a function of speed. Solid line is final η_v versus speed curve.

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effect on volumetric efficiency of each of the different phenomena described in this section varies with speed. Non-speed-dependent effects (such as fuel vapor pressure) drop η_v below 100 percent (curve *A*). Charge heating in the manifold and cylinder drops curve *A* to curve *B*. It has a greater effect at lower engine speeds due to longer gas residence times. Frictional flow losses increase as the square of engine speed, and drop curve *B* to curve *C*. At higher engine speeds, the flow into the engine during at least part of the intake process becomes choked (see Sec. 6.3.2). Once this occurs, further increases in speed do not increase the flow rate significantly so volumetric efficiency decreases sharply (curve *C* to *D*). The induction ram effect, at higher engine speeds, raises curve *D* to curve *E*. Late inlet valve closing, which allows advantage to be taken of increased charging at higher speeds, results in a decrease in η_v at low engine speeds due to backflow (curves *C* and *D* to *F*). Finally, intake and/or exhaust tuning can increase the volumetric efficiency (often by a substantial amount) over part of the engine speed range, curve *F* to *G*.

An example of the effect on volumetric efficiency of tuning the intake manifold runner is shown in Fig. 6-10. In an unsteady flow calculation of the gas exchange processes of a four-cylinder spark-ignition engine, the length of the intake manifold runners was increased successively by factors of 2. The 34-cm length produces a desirable "tuned" volumetric efficiency curve with increased low-speed air flow and flat mid-speed characteristics. While the longest runner further increases low-speed air flow, the loss in η_v at high speed would be unacceptable.[10] Further discussion of intake system tuning can be found in Sec. 7.6.2.

Figure 6-11 shows data from a four-cylinder spark-ignition engine[3] which illustrates the effect of varying valve timing and valve lift on the volumetric efficiency versus speed curve. Earlier-than-normal inlet valve closing reduces backflow losses at low speed and increases η_v. The penalty is reduced air flow at high speed. Later-than-normal inlet valve closing is only advantageous at very high

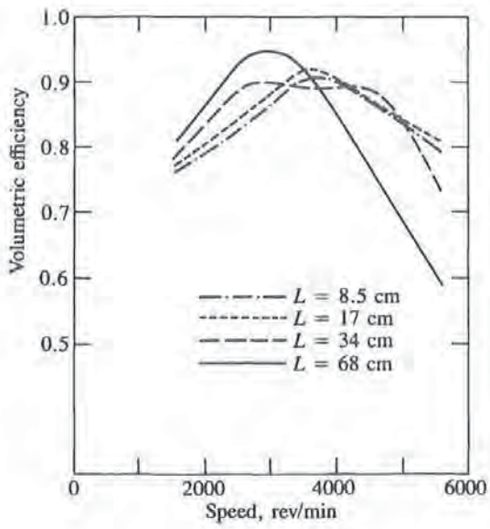


FIGURE 6-10
Effect of intake runner length on volumetric efficiency versus speed for 2.3-dm^3 four-cylinder spark-ignition engine.[10]

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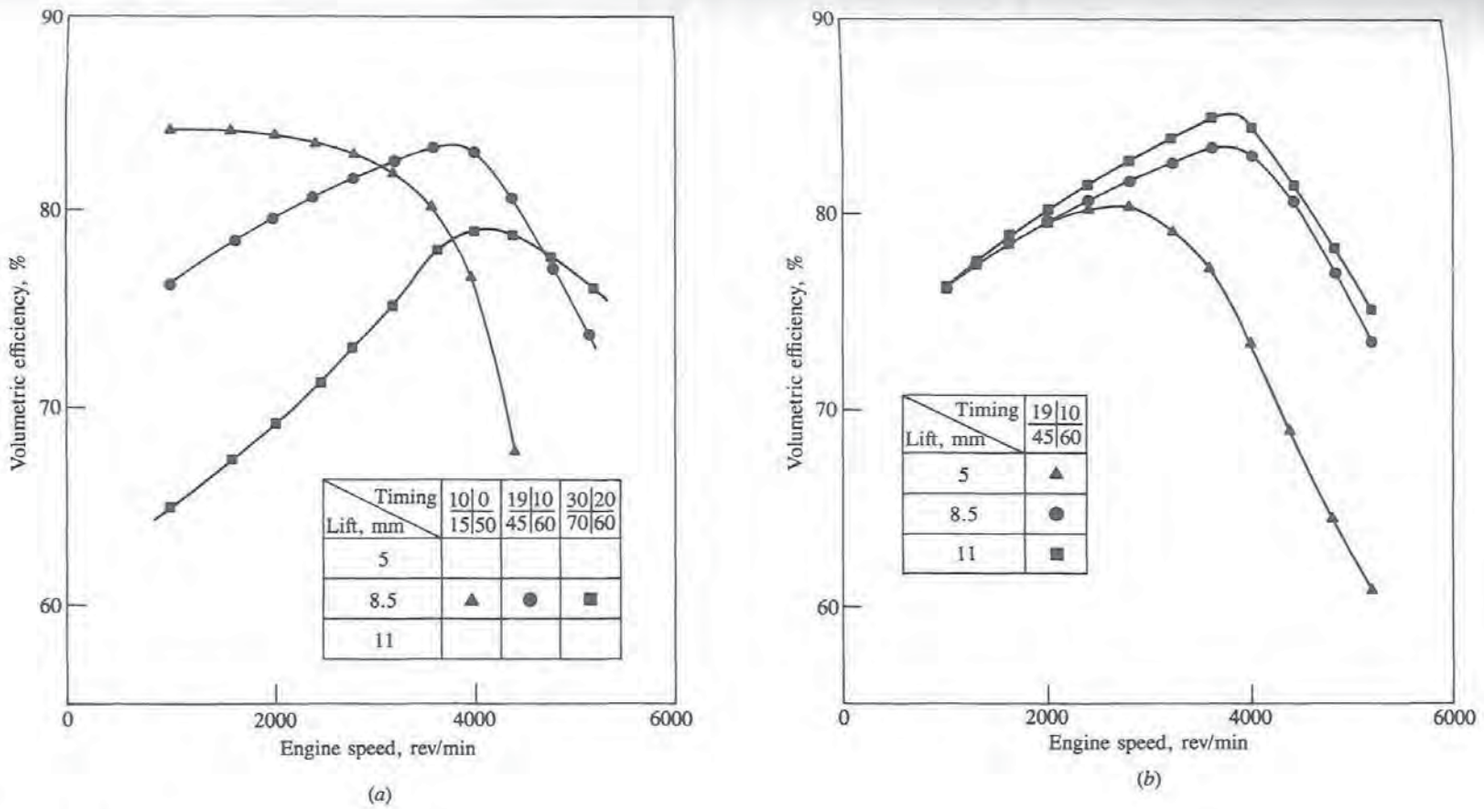


FIGURE 6-11
Effect of (*a*) valve timing and (*b*) valve lift on volumetric efficiency versus speed curves. Four-cylinder 1.6-dm^3 displacement spark-ignition engine at wide-open throttle, firing conditions, $(A/F) = 13$, MBT ignition timing. Timing numbers are: inlet valve opens (before TC) top left, closes (after BC) bottom left; exhaust valve opens (before BC) bottom right, closes (after TC) top right.[3] Stroke = 89 mm.

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speeds. Low valve lifts significantly restrict engine breathing over the mid-speed and high-speed operating ranges. Above a critical valve lift, lift is no longer a major constraint on effective valve open area (see Sec. 6.3).

6.3 FLOW THROUGH VALVES

The valve, or valve and port together, is usually the most important flow restriction in the intake and the exhaust system of four-stroke cycle engines. The characteristics of flows through poppet valves will now be reviewed.

6.3.1 Poppet Valve Geometry and Timing

Figure 6-12 shows the main geometric parameters of a poppet valve head and seat. Figure 6-13 shows the proportions of typical inlet and exhaust valves and ports, relative to the valve inner seat diameter D. The inlet port is generally circular, or nearly so, and the cross-sectional area is no larger than is required to achieve the desired power output. For the exhaust port, the importance of good valve seat and guide cooling, with the shortest length of exposed valve stem, leads to a different design. Although a circular cross section is still desirable, a rectangular or oval shape is often essential around the guide boss area. Typical valve head sizes for different shaped combustion chambers in terms of cylinder bore B are given in Table 6.1.[11] Each of these chamber shapes (see Secs. 10.2 and 15.4 for a discussion of spark-ignition and diesel combustion chamber design) imposes different constraints on valve size. Larger valve sizes (or four valves compared with two) allow higher maximum air flows for a given cylinder displacement.

Typical valve timing, valve-lift profiles, and valve open areas for a four-stroke cycle spark-ignition engine are shown in Fig. 6-14. There is no universally accepted criterion for defining valve timing points. Some are based upon a spe-

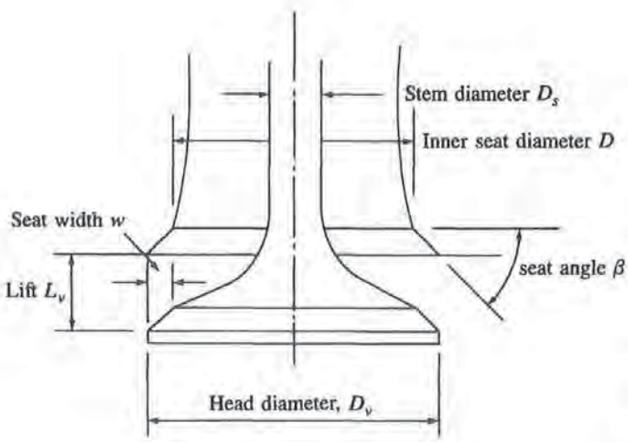
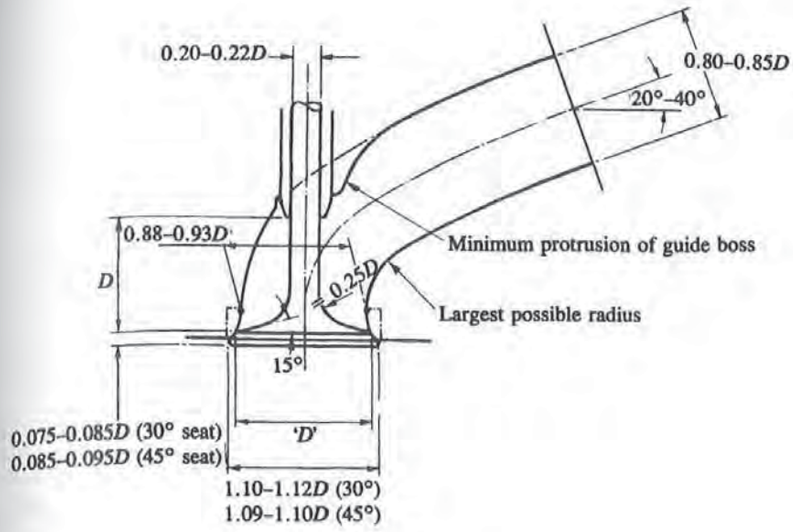
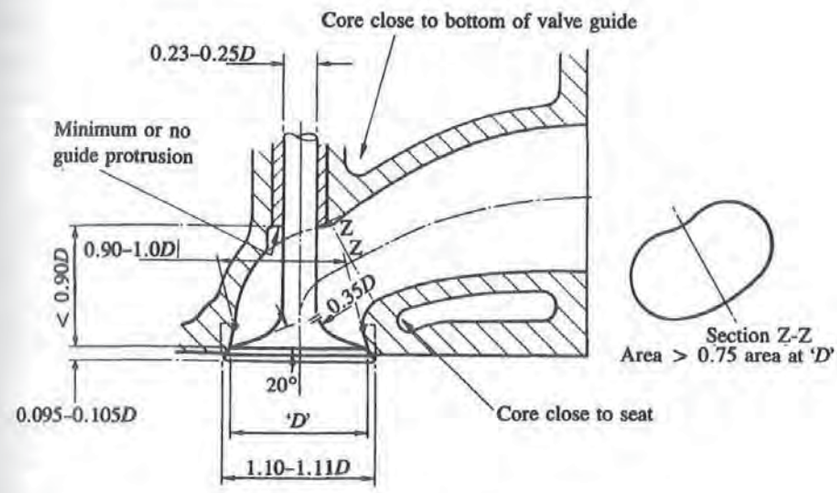


FIGURE 6-12
Parameters defining poppet valve geometry.

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(a)



(b)

FIGURE 6-13
Shape, proportions, and critical design areas of typical inlet (top) and exhaust (bottom) valves and ports.[11]

cific lift criterion. For example, SAE defines valve timing events based on reference valve-lift points:[13]

1. Hydraulic lifters. Opening and closing positions are the 0.15-mm (0.006-in) valve-lift points.
2. Mechanical lifters. Valve opening and closing positions are the points of 0.15-mm (0.006-in) lift plus the specified lash.

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TABLE 6.1
Valve head diameter in terms of cylinder bore B[11]

Combustion chamber shape†	Inlet	Exhaust	Approximate mean piston speed, max power, m/s
Wedge or bathtub	0.43–0.46B	0.35–0.37B	15
Bowl-in-piston	0.42–0.44B	0.34–0.37B	14
Hemispherical	0.48–0.5B	0.41–0.43B	18
Four-valve pent-roof	0.35–0.37B	0.28–0.32B	20

† See Fig. 15-15.

Alternatively, valve events can be defined based on angular criteria along the lift curve.[12] What is important is when significant gas flow through the valve-open area either starts or ceases.

The instantaneous valve flow area depends on valve lift and the geometric details of the valve head, seat, and stem. There are three separate stages to the flow area development as valve lift increases,[14] as shown in Fig. 6-14*b*. For low valve lifts, the minimum flow area corresponds to a frustrum of a right circular cone where the conical face between the valve and the seat, which is perpendicular to the seat, defines the flow area. For this stage:

$$\frac{w}{\sin \beta \cos \beta} > L_v > 0$$

and the minimum area is

$$A_m = \pi L_v \cos \beta \left(D_v - 2w + \frac{L_v}{2} \sin 2\beta \right) \tag{6.7}$$

where β is the valve seat angle, L_v is the valve lift, D_v is the valve head diameter (the outer diameter of the seat), and w is the seat width (difference between the inner and outer seat radii).

For the second stage, the minimum area is still the slant surface of a frustrum of a right circular cone, but this surface is no longer perpendicular to the valve seat. The base angle of the cone increases from $(90 - \beta)°$ toward that of a cylinder, 90°. For this stage:

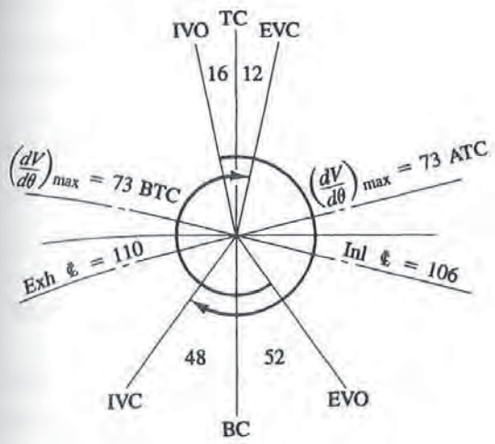
$$\left[\left(\frac{D_p^2 - D_s^2}{4D_m} \right) - w^2 \right]^{1/2} + w \tan \beta \geqslant L_v > \frac{w}{\sin \beta \cos \beta}$$

and

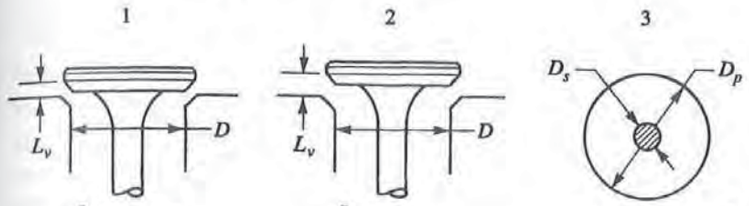
$$A_m = \pi D_m [(L_v - w \tan \beta)^2 + w^2]^{1/2} \tag{6.8}$$

where D_p is the port diameter, D_s is the valve stem diameter, and D_m is the mean seat diameter $(D_v - w)$.

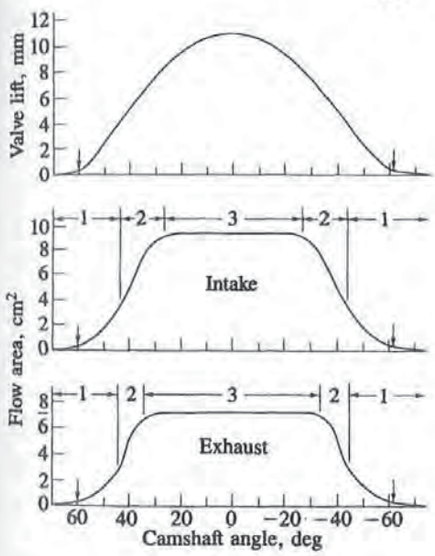
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(*a*)



(*b*)



(*c*)

FIGURE 6-14
(*a*) Typical valve timing diagram for high-speed 2.2-dm^3 four-cylinder spark-ignition engine. (*b*) Schematic showing three stages of valve lift. (*c*) Valve-lift curve and corresponding minimum intake and exhaust valve open areas as a function of camshaft angle. Inlet and exhaust valve diameters are 3.6 and 3.1 cm, respectively.[12]

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Finally, when the valve lift is sufficiently large, the minimum flow area is no longer between the valve head and seat; it is the port flow area minus the sectional area of the valve stem. Thus, for

$$L_v > \left[\left(\frac{D_p^2 - D_s^2}{4D_m}\right)^2 - w^2\right]^{1/2} + w \tan \beta$$

then

$$A_m = \frac{\pi}{4}(D_p^2 - D_s^2) \tag{6.9}$$

Intake and exhaust valve open areas corresponding to a typical valve-lift profile are plotted versus camshaft angle in Fig. 6-14*c*. These three different flow regimes are indicated. The maximum valve lift is normally about 12 percent of the cylinder bore.

Inlet valve opening (IVO) typically occurs 10 to 25° BTC. Engine performance is relatively insensitive to this timing point. It should occur sufficiently before TC so that cylinder pressure does not dip early in the intake stroke. Inlet valve closing (IVC) usually falls in the range 40 to 60° after BC, to provide more time for cylinder filling under conditions where cylinder pressure is below the intake manifold pressure at BC. IVC is one of the principal factors that determines high-speed volumetric efficiency; it also affects low-speed volumetric efficiency due to backflow into the intake (see Sec. 6.2.3). Exhaust valve opening (EVO) occurs 50 to 60° before BC, well before the end of the expansion stroke, so that blowdown can assist in expelling the exhaust gases. The goal here is to reduce cylinder pressure to close to the exhaust manifold pressure as soon as possible after BC over the full engine speed range. Note that the timing of EVO affects the cycle efficiency since it determines the effective expansion ratio. Exhaust valve closing (EVC) ends the exhaust process and determines the duration of the valve overlap period. EVC typically falls in the range 8 to 20° after TC. At idle and light load, in spark-ignition engines (which are throttled), it therefore regulates the quantity of exhaust gases that flow back into the combustion chamber through the exhaust valve under the influence of intake manifold vacuum. At high engine speeds and loads, it regulates how much of the cylinder burned gases are exhausted. EVC timing should occur sufficiently far after TC so that the cylinder pressure does not rise near the end of the exhaust stroke. Late EVC favors high power at the expense of low-speed torque and idle combustion quality. Note from the timing diagram (Fig. 6-14*a*) that the points of maximum valve lift and maximum piston velocity (Fig. 2-2) do not coincide.

The effect of valve geometry and timing on air flow can be illustrated conceptually by dividing the rate of change of cylinder volume by the instantaneous minimum valve flow area to obtain a *pseudo flow velocity* for each valve:[12]

$$v_{ps} = \frac{1}{A_m}\frac{dV}{d\theta} = \frac{\pi B^2}{4A_m}\frac{ds}{d\theta} \tag{6.10}$$

where V is the cylinder volume [Eq. (2.4)], B is the cylinder bore, s is the distance

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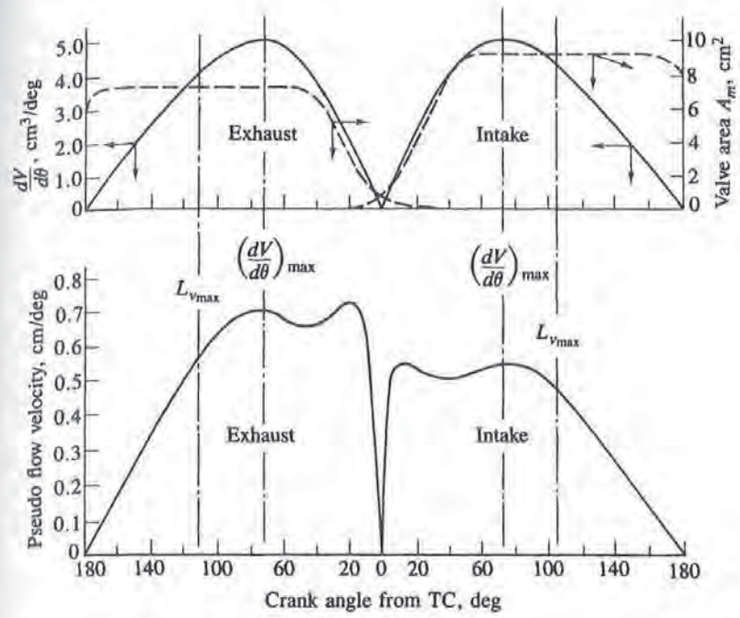


FIGURE 6-15
Rate of change of cylinder volume $dV/d\theta$, valve minimum flow area A_m, and pseudo flow velocity as function of crank angle for exhaust and inlet valves of Fig. 6-14.[12]

between the wrist pin and crank axis [see Fig. 2-1 and Eq. (2.5)] and A_m is the valve area given by Eqs. (6.7), (6.8), or (6.9). Instantaneous pseudo flow velocity profiles for the exhaust and intake strokes of a four-stroke four-cylinder engine are shown in Fig. 6-15. Note the appearance of two peaks in the pseudo flow velocity for both the exhaust and intake strokes. The broad peaks occurring at maximum piston velocity reflect the fact that valve flow area is constant at this point. The peaks close to TC result from the exhaust valve closing and intake valve opening profiles. The peak at the end of the exhaust stroke is important since it indicates a high pressure drop across the valve at this point, which will result in higher trapped residual mass. The magnitude of this exhaust stroke pseudo velocity peak depends strongly on the timing of exhaust valve closing. The pseudo velocity peak at the start of the intake stroke is much less important. That the pseudo velocities early in the exhaust stroke and late in the intake stroke are low indicates that phenomena other than quasi-steady flow govern the flow rate. These are the periods when exhaust blowdown and ram and tuning effects in the intake are most important.

6.3.2 Flow Rate and Discharge Coefficients

The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction [Eqs. (C.8) or (C.9) in App. C]. This equation is derived from a one-dimensional isentropic flow analysis, and

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real gas flow effects are included by means of an experimentally determined discharge coefficient C_D. The air flow rate is related to the upstream stagnation pressure p_0 and stagnation temperature T_0, static pressure just downstream of the flow restriction (assumed equal to the pressure at the restriction, p_T), and a reference area A_R characteristic of the valve design:

$$\dot{m} = \frac{C_D A_R p_0}{(RT_0)^{1/2}} \left(\frac{p_T}{p_0}\right)^{1/\gamma} \left\{ \frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{p_T}{p_0}\right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2} \tag{6.11}$$

When the flow is choked, i.e., $p_T/p_0 \leq [2/(\gamma + 1)]^{\gamma/(\gamma-1)}$, the appropriate equation is

$$\dot{m} = \frac{C_D A_R p_0}{(RT_0)^{1/2}} \gamma^{1/2} \left(\frac{2}{\gamma + 1}\right)^{(\gamma+1)/2(\gamma-1)} \tag{6.12}$$

For flow into the cylinder through an intake valve, p_0 is the intake system pressure p_i and p_T is the cylinder pressure. For flow out of the cylinder through an exhaust valve, p_0 is the cylinder pressure and p_T is the exhaust system pressure.

The value of C_D and the choice of reference area are linked together: their product, $C_D A_R$, is the effective flow area of the valve assembly A_E. Several different reference areas have been used. These include the valve head area $\pi D_v^2/4$,[7] the port area at the valve seat $\pi D_p^2/4$,[15] the geometric minimum flow area [Eqs. (6.7), (6.8), and (6.9)], and the curtain area $\pi D_v L_v$,[16] where L_v is the valve lift. The choice is arbitrary, though some of these choices allow easier interpretation than others. As has been shown above, the geometric minimum flow area is a complex function of valve and valve seat dimensions. The most convenient reference area in practice is the so-called valve curtain area:

$$A_C = \pi D_v L_v \tag{6.13}$$

since it varies linearly with valve lift and is simple to determine.

INLET VALVES. Figure 6-16 shows the results of steady flow tests on a typical inlet valve configuration with a sharp-cornered valve seat.[16] The discharge coefficient based on valve curtain area is a discontinuous function of the valve-lift/diameter ratio. The three segments shown correspond to different flow regimes as indicated. At very low lifts, the flow remains attached to the valve head and seat, giving high values for the discharge coefficient. At intermediate lifts, the flow separates from the valve head at the inner edge of the valve seat as shown. An abrupt decrease in discharge coefficient occurs at this point. The discharge coefficient then increases with increasing lift since the size of the separated region remains approximately constant while the minimum flow area is increasing. At high lifts, the flow separates from the inner edge of the valve seat as well.[14, 17] Typical maximum values of L_v/D_v are 0.25.

An important question is whether these steady flow data are representative of the dynamic flow behavior of the valve in an operating engine. There is some evidence that the "change points" between different flow regimes shown in Fig. 6-16 occur at slightly different valve lifts under dynamic operation than under

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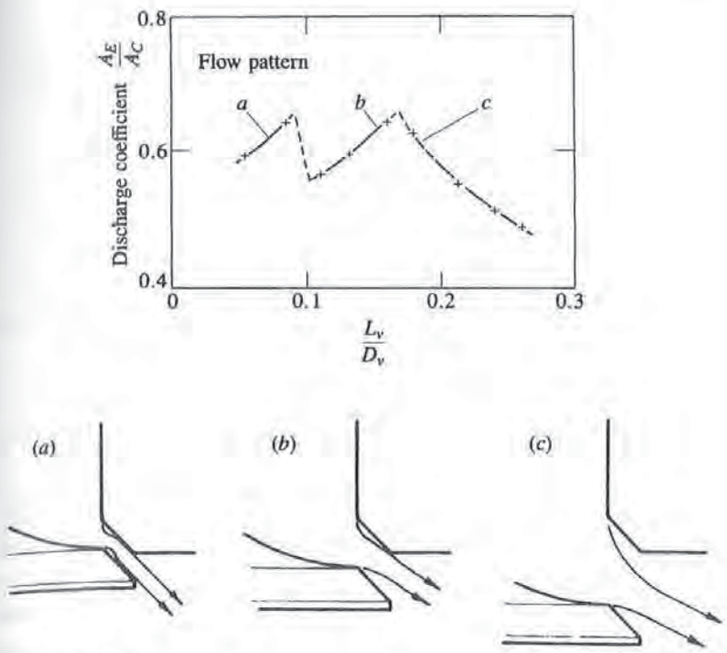


FIGURE 6-16
Discharge coefficient of typical inlet poppet valve (effective flow area/valve curtain area) as a function of valve lift. Different segments correspond to flow regimes indicated.[16]

steady flow operation. Also, as has been discussed in Sec. 6.2.2, the pressure upstream of the valve varies significantly during the intake process. However, it has been shown that over the normal engine speed range, steady flow discharge-coefficient results can be used to predict dynamic performance with reasonable precision.[14, 18]

In addition to valve lift, the performance of the inlet valve assembly is influenced by the following factors: valve seat width, valve seat angle, rounding of the seat corners, port design, cylinder head shape. In many engine designs the port and valve assembly are used to generate a rotational motion (swirl) inside the engine cylinder during the induction process, or the cylinder head can be shaped to restrict the flow through one side of the valve open area to generate swirl. Swirl production is discussed later, in Section 8.3. Swirl generation significantly reduces the valve (and port) flow coefficient. Changes in seat width affect the L_v/D_v at which the shifts in flow regimes illustrated in Fig. 6-16 occur. C_D increases as seat width decreases. The seat angle β affects the discharge coefficient in the low-lift regime in Fig. 6-16. Rounding the upstream corner of the valve seat reduces the tendency of the flow to break away, thus increasing C_D at higher lifts. At low valve lifts, when the flow remains attached, increasing the Reynolds number decreases the discharge coefficient. Once the flow breaks away from the wall, there is no Reynolds number dependence of C_D.[16]

For well-designed ports (e.g., Fig. 6-13) the discharge coefficient of the port and valve assembly need be no lower than that of the isolated valve (except when

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the port is used to generate swirl). However, if the cross-sectional area of the port is not sufficient or the radius of the surface at the inside of the bend is too small, a significant reduction in C_D for the assembly can result.[16]

At high engine speeds, unless the inlet valve is of sufficient size, the inlet flow during part of the induction process can become choked (i.e., reach sonic velocity at the minimum valve flow area). Choking substantially reduces volumetric efficiency. Various definitions of inlet Mach number have been used to identify the onset of choking. Taylor and coworkers[7] correlated volumetric efficiencies measured on a range of engine and inlet valve designs with an *inlet Mach index Z* formed from an average gas velocity through the inlet valve:

$$Z = \frac{A_p \bar{S}_p}{C_i A_i a} \tag{6.14}$$

where A_i is the nominal inlet valve area ($\pi D_v^2/4$), C_i is a mean valve discharge coefficient based on the area A_i, and a is the sound speed. From the method used to determine C_i, it is apparent that $C_i A_i$ is the average effective open area of the valve (it is the average value of $C_D \pi D_v L_v$). Z corresponds closely, therefore, to the mean Mach number in the inlet valve throat. Taylor's correlations show that η_v decreases rapidly for $Z \geq 0.5$. An alternative equivalent approach to this problem has been developed, based on the average flow velocity through the valve during the period the valve is open.[19] A *mean inlet Mach number* was defined:

$$\bar{M}_i = \frac{\bar{v}_i}{a} \tag{6.15}$$

where $\bar{v}_i$ is the mean inlet flow velocity during the valve open period. $\bar{M}_i$ is related to Z via

$$\bar{M}_i = \frac{Z(\eta_v/100)180}{\theta_{\text{IVC}} - \theta_{\text{IVO}}} \tag{6.16}$$

This mean inlet Mach number correlates volumetric efficiency characteristics better than the Mach index. For a series of modern small four-cylinder engines, when $\bar{M}_i$ approaches 0.5 the volumetric efficiency decreases rapidly. This is due to the flow becoming choked during part of the intake process. This relationship can be used to size the inlet valve for the desired volumetric efficiency at maximum engine speed. Also, if the inlet valve is closed too early, volumetric efficiency will decrease gradually with increasing $\bar{M}_i$, for $\bar{M}_i < 0.5$, even if the valve open area is sufficiently large.[19]

EXHAUST VALVES. In studies of the flow from the cylinder through an exhaust poppet valve, different flow regimes at low and high lift occur, as shown in Fig. 6-17. Values of C_D based on the valve curtain area, for several different exhaust valve and port combinations, are given in Fig. 6-18. A sharp-cornered isolated poppet valve (i.e., straight pipe downstream, no port) gives the best performance.

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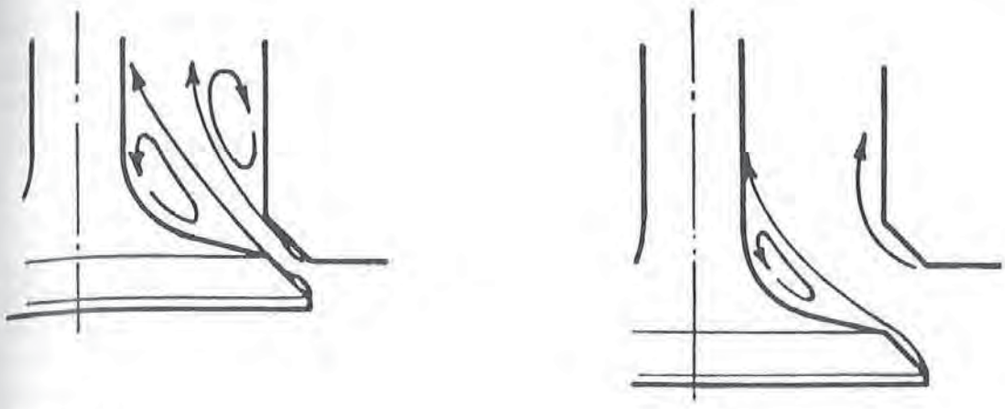


FIGURE 6-17
Flow pattern through exhaust valve at low and high lift.[16]

At high lifts, $L_v/D_v \geq 0.2$, the breakaway of the flow reduces the discharge coefficient. (At $L_v/D_v = 0.25$ the effective area is about 90 percent of the minimum geometric area. For $L_v/D_v < 0.2$ it is about 95 percent.[16]) The port design significantly affects C_D at higher valve lifts, as indicated by the data from four port designs in Fig. 6-18. Good designs can approach the performance of isolated

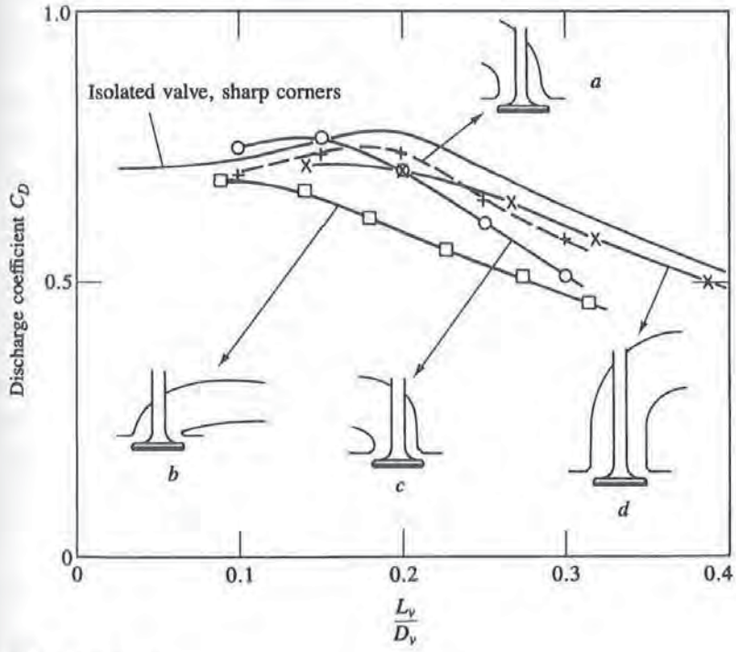


FIGURE 6-18
Discharge coefficient as function of valve lift for several exhaust valve and port designs.[16] *a*,[20] *b*,[15] *c*,[20] *d*.[21]

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valves, however. Exhaust valves operate over a wide range of pressure ratios (1 to 5). For pressure ratios greater than about 2 the flow will be choked, but the effect of pressure ratio on discharge coefficient is small and confined to higher lifts (e.g., ± 5 percent at $L_v/D_v = 0.3$).[15]

6.4 RESIDUAL GAS FRACTION

The residual gas fraction in the cylinder during compression is determined by the exhaust and inlet processes. Its magnitude affects volumetric efficiency and engine performance directly, and efficiency and emissions through its effect on working-fluid thermodynamic properties. The residual gas fraction is primarily a function of inlet and exhaust pressures, speed, compression ratio, valve timing, and exhaust system dynamics.

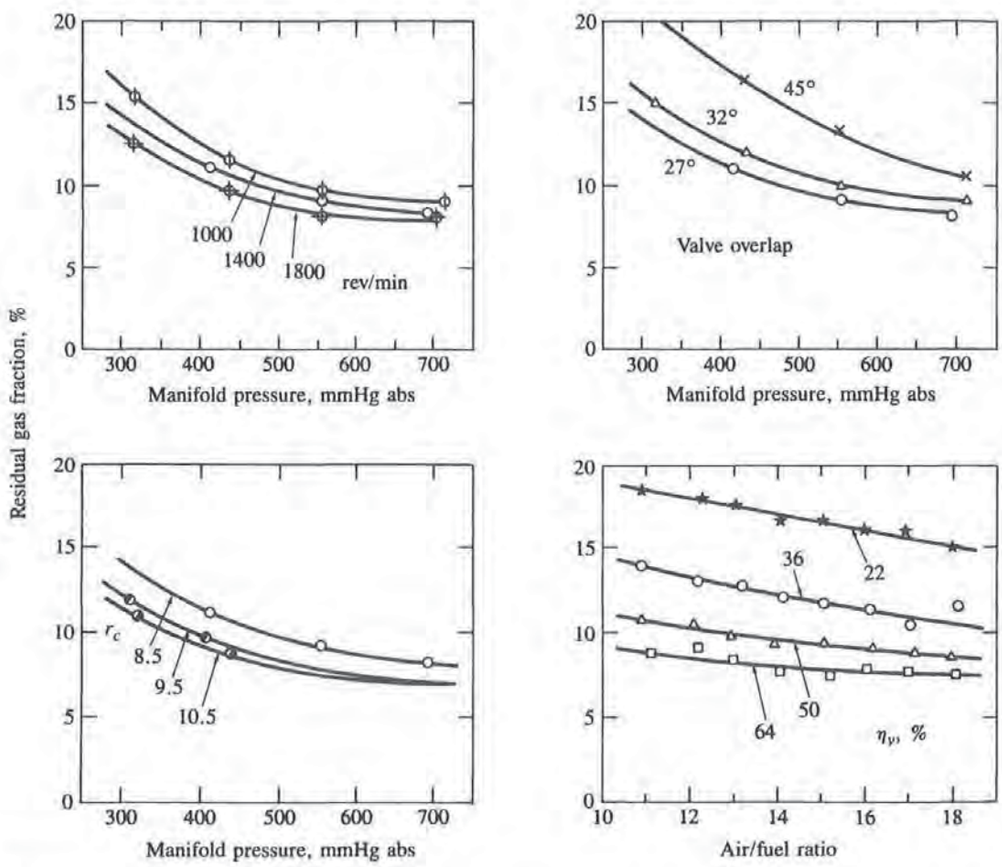


FIGURE 6-19
Residual gas fraction for 2-dm^3 four-cylinder spark-ignition engine as a function of intake manifold pressure for a range of speeds, compression ratios, and valve overlaps: also as a function of air/fuel ratio for a range of volumetric efficiencies. Operating conditions, unless noted: speed = 1400 rev/min, $A/F = 14.5$, spark timing set to give 0.95 maximum torque, compression ratio = 8.5.[22]

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The residual gas mass fraction x_r (or burned gas fraction if EGR is used) is usually determined by measuring the CO_2 concentration in a sample of gas extracted from the cylinder during the compression stroke. Then

$$x_r = \frac{(\tilde{x}_{CO_2})_C}{(\tilde{x}_{CO_2})_e} \tag{6.17}$$

where the subscripts C and e denote compression and exhaust, and $\tilde{x}_{CO_2}$ are mole fractions in the wet gas. Usually CO_2 volume or mole fractions are measured in dry gas streams (see Sec. 4.9). A correction factor K,

$$K = \frac{(\tilde{x}_i)_{wet}}{(\tilde{x}_i)_{dry}} = \frac{1}{1 + 0.5[\, y(\tilde{x}^*_{CO_2} + x^*_{CO}) - 0.74\tilde{x}^*_{CO}]} \tag{6.18}$$

where y is the molar hydrogen/carbon ratio of the fuel and $\tilde{x}^*_{CO_2}$, $\tilde{x}^*_{CO}$ are dry mole fractions, can be used to convert the dry mole fraction measurements.

Residual gas measurements in a spark-ignition engine are given in Fig. 6-19, which shows the effect of changes in speed, valve overlap, compression ratio, and air/fuel ratio for a range of inlet manifold pressures for a 2-dm^3, 88.5-mm bore, four-cylinder engine.[22] The effect of variations in spark timing was negligible. Inlet pressure, speed, and valve overlap are the most important variables, though the exhaust pressure also affects the residual fraction.[23] Normal settings for inlet valve opening (about 15° before TC) and exhaust valve closing (about 12° after TC) provide sufficient overlap for good scavenging, but avoid excessive backflow from the exhaust port into the cylinder.

Residual gas fractions in diesel engines are substantially lower than in SI engines because inlet and exhaust pressures are comparable in magnitude and the compression ratio is 2 to 3 times as large. Also, a substantial fraction of the residual gas is air.

6.5 EXHAUST GAS FLOW RATE AND TEMPERATURE VARIATION

The exhaust gas mass flow rate and the properties of the exhaust gas vary significantly during the exhaust process. The origin of this variation for an ideal exhaust process is evident from Fig. 5-3. The thermodynamic state (pressure, temperature, etc.) of the gas in the cylinder varies continually during the exhaust blowdown phase, until the cylinder pressure closely approaches the exhaust manifold pressure. In the real exhaust process, the exhaust valve restricts the flow out of the cylinder, the valve lift varies with time, and the cylinder volume changes during the blowdown process, but the principles remain the same.

Measurements have been made of the variation in mass flow rate through the exhaust valve and gas temperature at the exhaust port exit during the exhaust process of a spark-ignition engine.[24] Figure 6-20 shows the instantaneous mass flow rate data at three different engine speeds. The blowdown and displacement

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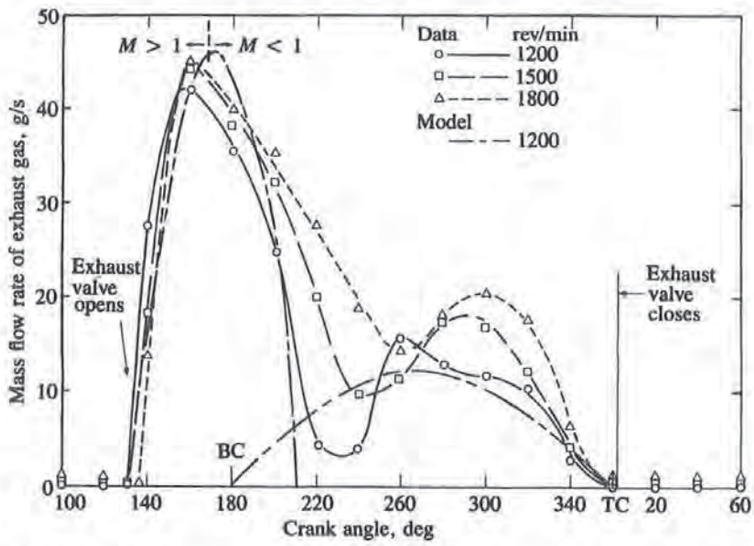


FIGURE 6-20
Instantaneous mass flow rate of exhaust gas through the valve versus crank angle: equivalence ratio = 1.2, wide-open throttle, compression ratio = 7. Dash-dot line is one-dimensional compressible flow model for blowdown and incompressible displacement model for exhaust stroke.[24]

phases of the exhaust process are evident. Simple quasi-steady models of these phases give good agreement with the data at lower engine speeds. The blowdown model shown applies orifice flow equations to the flow across the exhaust valve using the measured cylinder pressure and estimated gas temperature for upstream stagnation conditions. Equation (C.9) is used when the pressure ratio across the valve exceeds the critical value. Equation (C.8) is used when the pressure ratio is less than the critical value. The displacement model assumes the gas in the cylinder is incompressible as the piston moves through the exhaust stroke. As engine speed increases, the crank angle duration of the blowdown phase increases. There is evidence of dynamic effects occurring at the transition between the two phases. The peak mass flow rate during blowdown does not vary substantially with speed since the flow is choked. The mass flow rate at the time of maximum piston speed during displacement scales approximately with piston speed. As the inlet manifold pressure is reduced below the wide-open throttle value, the proportion of the charge which exits the cylinder during the blowdown phase decreases but the mass flow rate during displacement remains essentially constant.

The exhaust gas temperature varies substantially through the exhaust process, and decreases due to heat loss as the gas flows past the exhaust valve and through the exhaust system.

Figure 6-21 shows the measured cylinder pressure, calculated cylinder gas temperature and exhaust mass flow rate, and measured gas temperature at the exhaust port exit for a single-cylinder spark-ignition engine at mid-load and low speed.[25] The average cylinder-gas temperature falls rapidly during blowdown, and continues to fall during the exhaust stroke due to heat transfer to the cylin-

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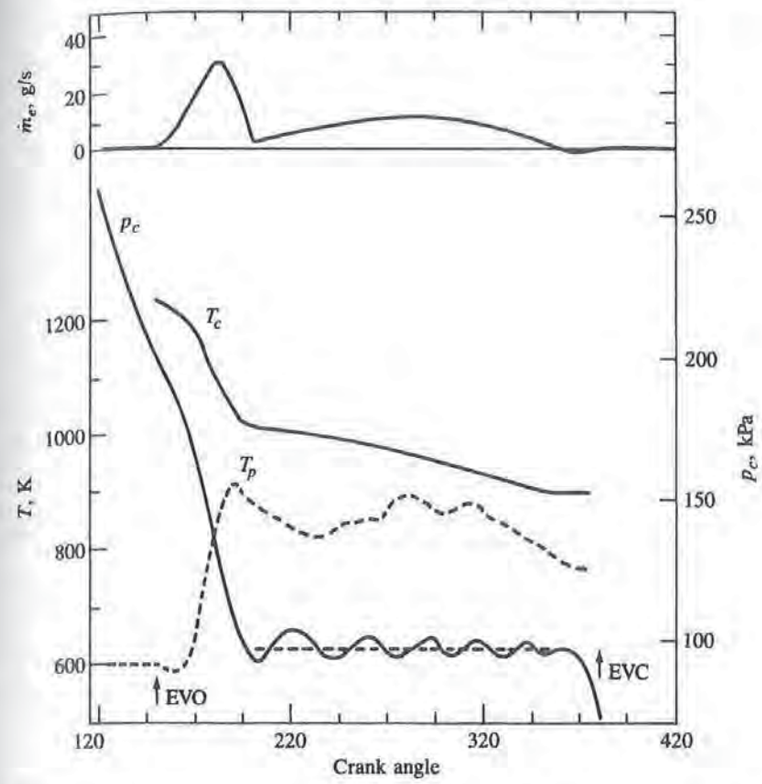


FIGURE 6-21
Measured cylinder pressure p_c, calculated cylinder-gas temperature T_c, exhaust mass flow rate $\dot{m}_e$, and measured gas temperature at exhaust port exit T_p, for single-cylinder spark-ignition engine. Speed = 1000 rev/min, imep = 414 kPa, equivalence ratio = 1.2, spark timing = 10° BTC, r_c = 7.2.[25]

der walls. The gas temperature at the port exit at the start of the exhaust flow pulse is a mixture of hotter gas which has just left the cylinder and cooler gas which left the cylinder at the end of the previous exhaust process and has been stationary in the exhaust port while the valve has been closed. The port exit temperature has a minimum where the transition from blowdown flow to displacement occurs, and the gas comes momentarily to rest and loses a substantial fraction of its thermal energy to the exhaust port walls.

Figure 6-22 shows the effect of varying load and speed on exhaust port exit temperatures. Increasing load ($A \rightarrow B \rightarrow C$) increases the mass and temperature in the blowdown pulse. Increasing speed ($B \rightarrow D$) raises the gas temperature throughout the exhaust process. These effects are the result of variations in the relative importance of heat transfer in the cylinder and heat transfer to the exhaust valve and port. The time available for heat transfer, which depends on engine speed and exhaust gas flow rate, is the most critical factor. The exhaust temperature variation with equivalence ratio follows from the variation in expansion stroke temperatures, with maximum values at $\phi = 1.0$ and lower values for leaner and richer mixtures.[24] Diesel engine exhaust temperatures are significantly

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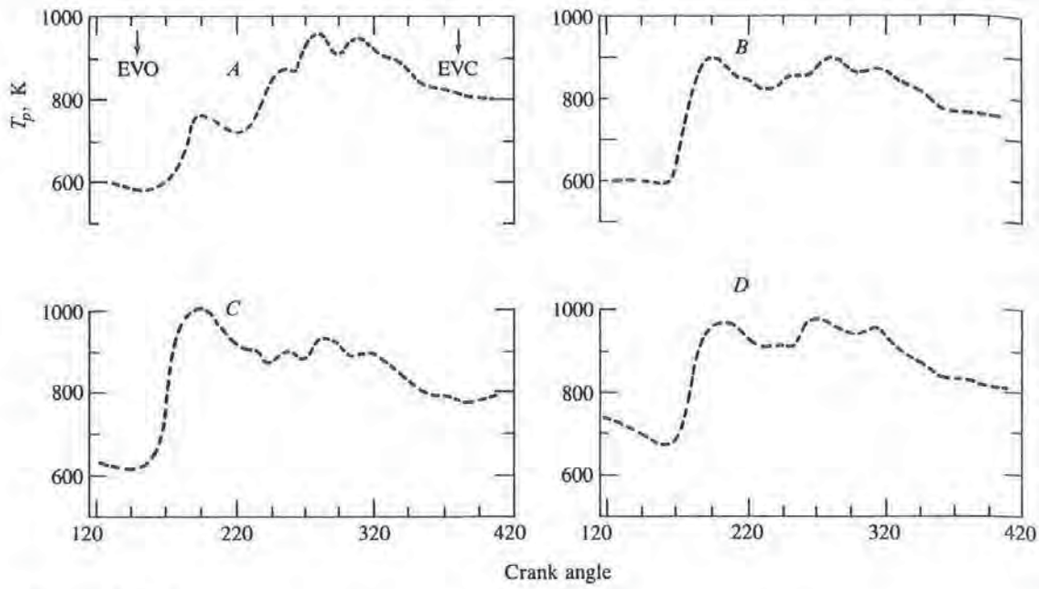


FIGURE 6-22
Measured gas temperature at exhaust port exit as a function of crank angle, single-cylinder spark-ignition engine, for different loads and speeds. Curve A: imep = 267 kPa, 1000 rev/min; curve B: imep = 414 kPa, 1000 rev/min; curve C: imep = 621 kPa, 1000 rev/min; curve D: imep = 414 kPa, 1600 rev/min. Equivalence ratio = 1.2, spark timing = 10° BTC, compression ratio = 7.2.[25]

lower than spark-ignition engine exhaust temperatures because of the lean operating equivalence ratio and their higher expansion ratio during the power stroke.

The average exhaust gas temperature is an important quantity for determining the performance of turbochargers, catalytic converters, and particulate traps. The time-averaged exhaust temperature does not correspond to the average energy of the exhaust gas because the flow rate varies substantially. An enthalpy-averaged temperature

$$\bar{T}_h = \left(\int_{\text{EVO}}^{\text{EVC}} \dot{m} c_p T_g \, d\theta \right) \Big/ \left(\int_{\text{EVO}}^{\text{EVC}} \dot{m} c_p \, d\theta \right) \tag{6.19}$$

is the best indicator of exhaust thermal energy. Average exhaust gas temperatures are usually measured with a thermocouple. Thermocouple-averaged temperatures are close to time-averaged temperatures. Mass-averaged exhaust temperatures (which are close to $\bar{T}_h$ if c_p variations are small) for a spark-ignition engine at the exhaust port exit are about 100 K higher than time-averaged or thermocouple-determined temperatures. Mass-average temperatures in the cylinder during the exhaust process are about 200 to 300 K higher than mass-averaged port temperatures. All these temperatures increase with increasing speed, load, and spark retard, with speed being the variable with the largest impact.[26]

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6.6 SCAVENGING IN TWO-STROKE CYCLE ENGINES

6.6.1 Two-Stroke Engine Configurations

In two-stroke cycle engines, each outward stroke of the piston is a power stroke. To achieve this operating cycle, the fresh charge must be supplied to the engine cylinder at a high-enough pressure to displace the burned gases from the previous cycle. Raising the pressure of the intake mixture is done in a separate pump or blower or compressor. The operation of clearing the cylinder of burned gases and filling it with fresh mixture (or air)—the combined intake and exhaust process—is called *scavenging*. However, air capacity is just as important as in the four-stroke cycle; usually, a greater air mass flow rate must be achieved to obtain the same output power. Figures 1-12, and 1-5 and 1-24 show sectioned drawings of a two-stroke spark-ignition engine and two two-stroke diesels.

The different categories of two-stroke cycle scavenging flows and the port (and valve) arrangements that produce them are illustrated in Figs. 6-23 and 6-24. Scavenging arrangements are classified into: (*a*) *cross-scavenged*, (*b*) *loop-scavenged*, and (*c*) *uniflow-scavenged configurations*. The location and orientation of the scavenging ports control the scavenging process, and the most common arrangements are indicated. Cross- and loop-scavenging systems use exhaust and inlet ports in the cylinder wall, uncovered by the piston as it approaches BC.[27] The uniflow system may use inlet ports with exhaust valves in the cylinder head,

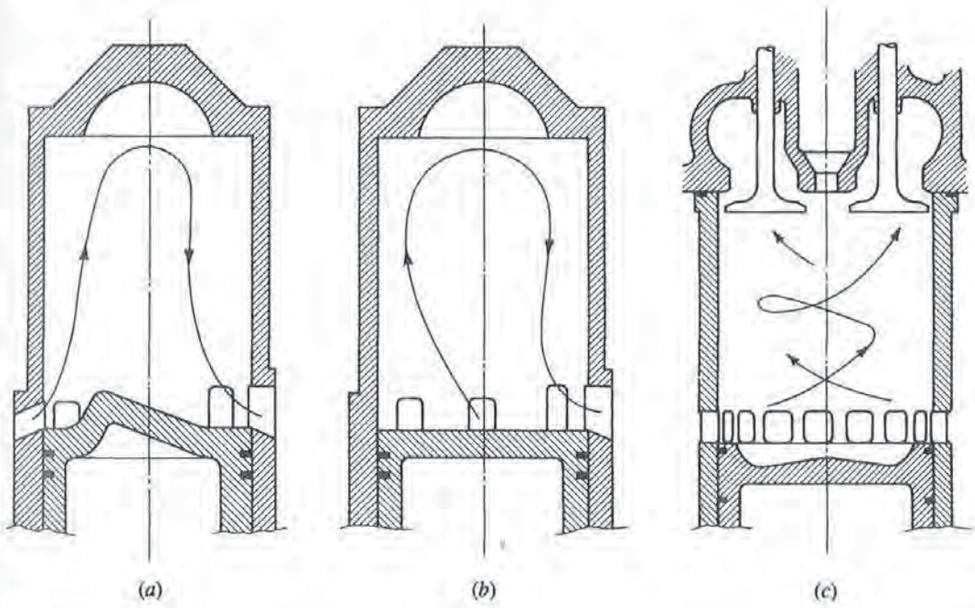


FIGURE 6-23
(*a*) Cross-scavenged, (*b*) loop-scavenged, and (*c*) uniflow-scavenged two-stroke cycle flow configurations.

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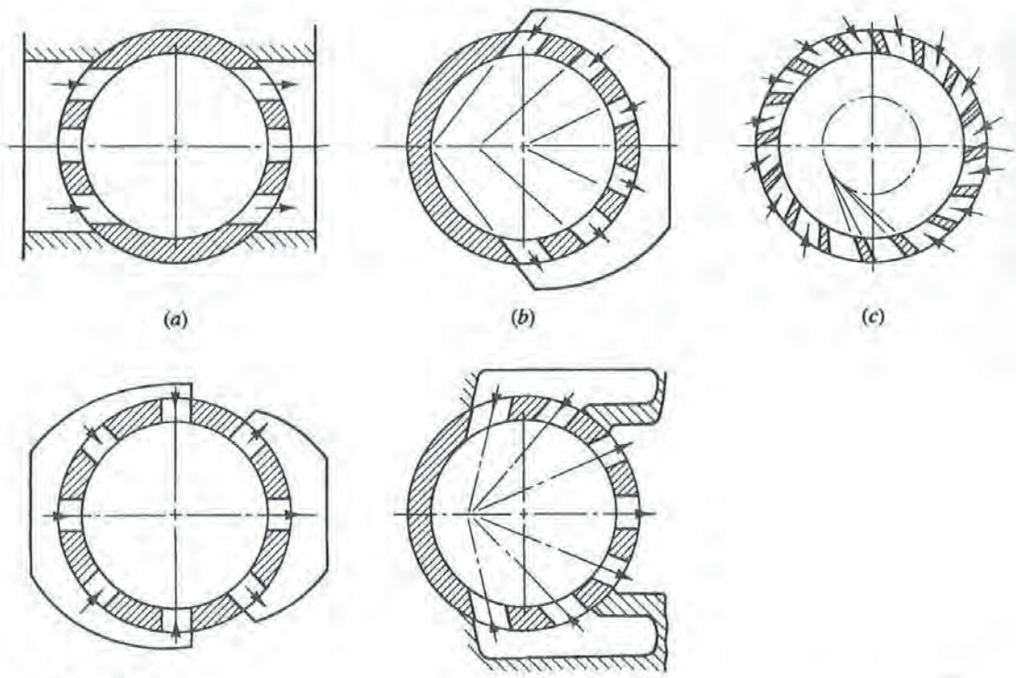


FIGURE 6-24
Common porting arrangements that go with (*a*) cross-scavenged, (*b*) loop-scavenged, and (*c*) uniflow-scavenged configurations.

or inlet and exhaust ports with opposed pistons. Despite the different flow patterns obtained with each cylinder geometry, the general operating principles are similar. Air in a diesel, or fuel-air mixture in a spark-ignition engine, must be supplied to the inlet ports at a pressure higher than the exhaust system pressure.

Figure 6-25 illustrates the principles of the scavenging process for a uniflow engine design. Between 100 and 110° after TC, the exhaust valve opens and a blowdown discharge process commences. Initially, the pressure ratio across the exhaust valve exceeds the critical value (see App. C) and the flow at the valve will be sonic: as the cylinder pressure decreases, the pressure ratio drops below the critical value. The discharge period up to the time of the scavenging port opening is called the blowdown (or free exhaust) period. The scavenging ports open between 60 and 40° before BC when the cylinder pressure slightly exceeds the scavenging pump pressure. Because the burned gas flow is toward the exhaust valves, which now have a large open area, the exhaust flow continues and no backflow occurs. When the cylinder pressure falls below the inlet pressure, air enters the cylinder and the scavenging process starts. This flow continues as long as the inlet ports are open and the inlet total pressure exceeds the pressure in the cylinder. As the cylinder pressure rises above the exhaust pressure, the fresh charge flowing into the cylinder displaces the burned gases: a part of the fresh charge mixes with the burned gases and is expelled with them. The exhaust valves usually close after the inlet ports close. Since the flow in the cylinder is toward the exhaust valve, additional scavenging is obtained. Figure 1-16 illustrates the

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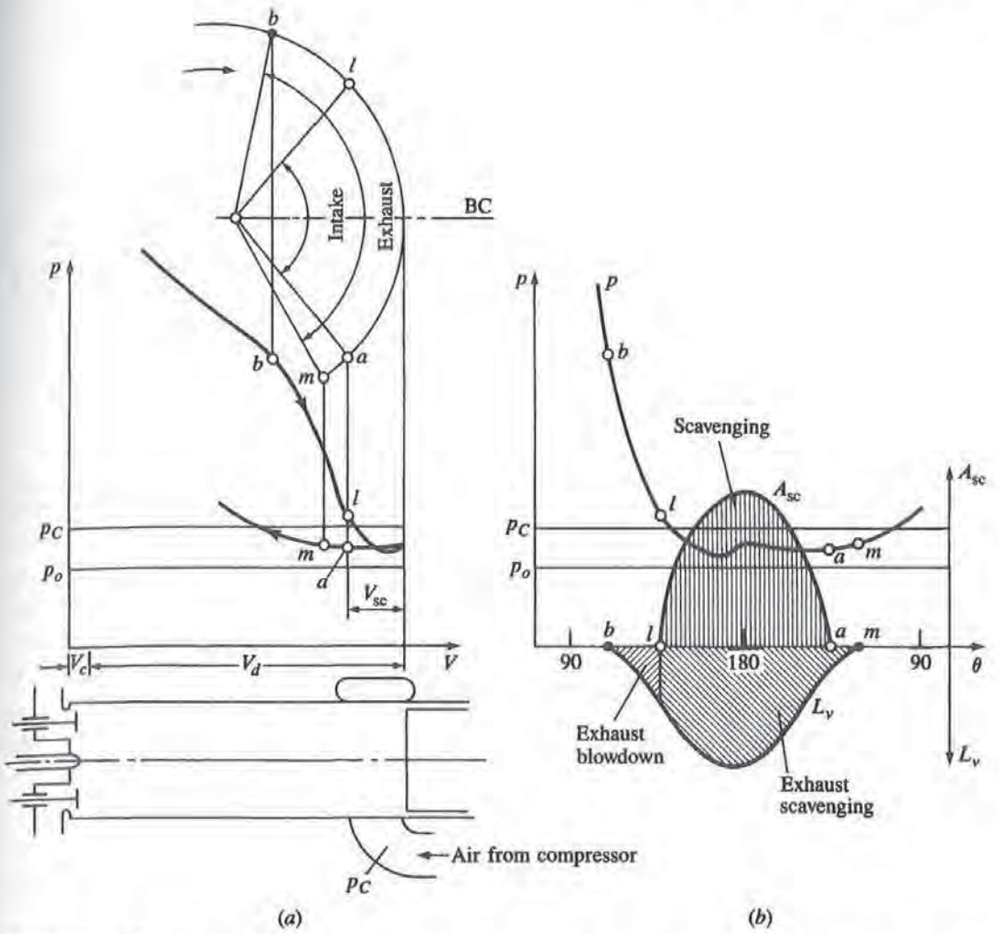


FIGURE 6-25
Gas exchange process in two-stroke cycle uniflow-scavenged diesel engine: (*a*) valve and port timing and pressure-volume diagram; (*b*) pressure, scavenging port open area A_{sc}, and exhaust valve lift L_v as functions of crank angle.[1]

similar sequence of events for a loop-scavenged engine. Proper flow patterns for the fresh charge are extremely important for good scavenging and charging of the cylinder.

Common methods for supercharging or pressurizing the fresh charge are shown in Fig. 6-26. In large two-stroke cycle engines, more complex combinations of these approaches are often used, as shown in Fig. 1-24.

6.6.2 Scavenging Parameters and Models

The following overall parameters are used to describe the scavenging process.[13]

The *delivery ratio* Λ:

$$\Lambda = \frac{\text{mass of delivered air (or mixture) per cycle}}{\text{reference mass}} \tag{6.20}$$

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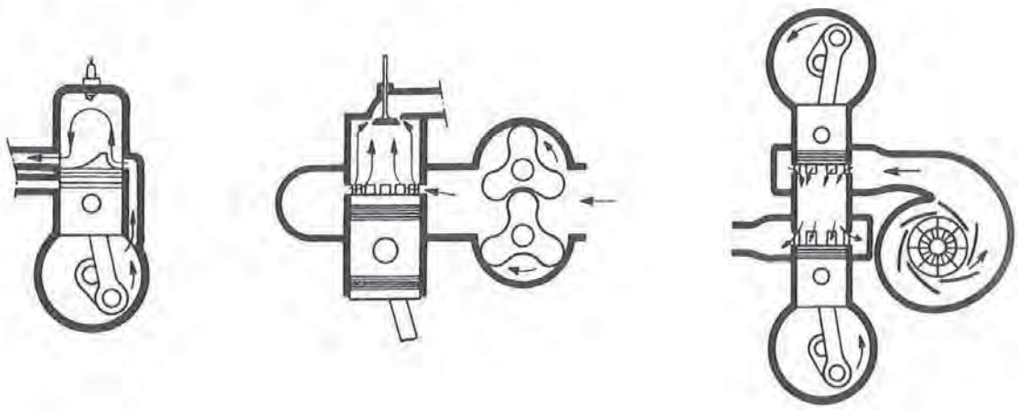


FIGURE 6-26
Common methods of pressurizing the fresh charge in two-stroke cycle engines: left, crankcase compression; center, roots blower; right, centrifugal compressor.[7]

compares the actual scavenging air mass (or mixture mass) to that required in an ideal charging process.† The reference mass is defined as displaced volume × ambient air (or mixture) density. Ambient air (or mixture) density is determined at atmospheric conditions or at intake conditions. This definition is useful for experimental purposes. For analytical work, it is often convenient to use the trapped cylinder mass m_{tr} as the reference mass.

The *trapping efficiency* η_{tr}:

$$\eta_{tr} = \frac{\text{mass of delivered air (or mixture) retained}}{\text{mass of delivered air (or mixture)}} \tag{6.21}$$

indicates what fraction of the air (or mixture) supplied to the cylinder is retained in the cylinder.

The *scavenging efficiency* η_{sc}:

$$\eta_{sc} = \frac{\text{mass of delivered air (or mixture) retained}}{\text{mass of trapped cylinder charge}} \tag{6.22}$$

indicates to what extent the residual gases in the cylinder have been replaced with fresh air.

The *purity* of the charge:

$$\text{Purity} = \frac{\text{mass of air in trapped cylinder charge}}{\text{mass of trapped cylinder charge}} \tag{6.23}$$

indicates the degree of dilution, with burned gases, of the unburned mixture in the cylinder.

† If scavenging is done with fuel-air mixture, as in spark-ignition engines, then mixture mass is used instead of air mass.

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The *charging efficiency* η_{ch}:

$$\eta_{ch} = \frac{\text{mass of delivered air (or mixture) retained}}{\text{displaced volume} \times \text{ambient density}} \tag{6.24}$$

indicates how effectively the cylinder volume has been filled with fresh air (or mixture). Charging efficiency, trapping efficiency, and delivery ratio are related by

$$\eta_{ch} = \Lambda \eta_{tr} \tag{6.25}$$

When the reference mass in the definition of delivery ratio is the trapped cylinder mass m_{tr} (or closely approximated by it) then

$$\eta_{sc} = \Lambda \eta_{tr} \tag{6.26}$$

In real scavenging processes, mixing occurs as the fresh charge displaces the burned gases and some of the fresh charge may be expelled. Two limiting ideal models of this process are: (1) perfect displacement and (2) complete mixing. Perfect displacement or scavenging would occur if the burned gases were pushed out by the fresh gases without any mixing. Complete mixing occurs if entering fresh mixture mixes instantaneously and uniformly with the cylinder contents.

For *perfect displacement* (with m_{tr} as the reference mass in the delivery ratio),

$$\begin{aligned} \eta_{sc} &= \Lambda \quad \text{and} \quad \eta_{tr} = 1 \quad \text{for } \Lambda \leqslant 1 \\ \eta_{sc} &= 1 \quad \text{and} \quad \eta_{tr} = \Lambda^{-1} \quad \text{for } \Lambda > 1 \end{aligned} \tag{6.27}$$

For the *complete mixing* limit, consider the scavenging process as a quasi-steady flow process. Between time t and $t + dt$, a mass element dm_{ad} of air is delivered to the cylinder and is uniformly mixed throughout the cylinder volume. An equal amount of fluid, with the same proportions of air and burned gas as the cylinder contents at time t, leaves the cylinder during this time interval. Thus the mass of air delivered between t and $t + dt$ which is retained, dm_{ar}, is given by

$$dm_{ar} = dm_{ad}\left(1 - \frac{m_{ar}}{m_{tr}}\right)$$

Assuming m_{tr} is constant, this integrates over the duration of the scavenging process to give

$$\frac{m_{ar}}{m_{tr}} = 1 - \exp\left(\frac{m_{ad}}{m_{tr}}\right) \tag{6.28}$$

Thus, for *complete mixing*, with the above definitions,

$$\begin{aligned} \eta_{sc} &= 1 - e^{-\Lambda} \\ \eta_{tr} &= \frac{1}{\Lambda}(1 - e^{-\Lambda}) \end{aligned} \tag{6.29}$$

Figure 6-27 shows η_{sc} and η_{tr} for the perfect displacement and complete mixing assumptions as a function of Λ, the delivery ratio.

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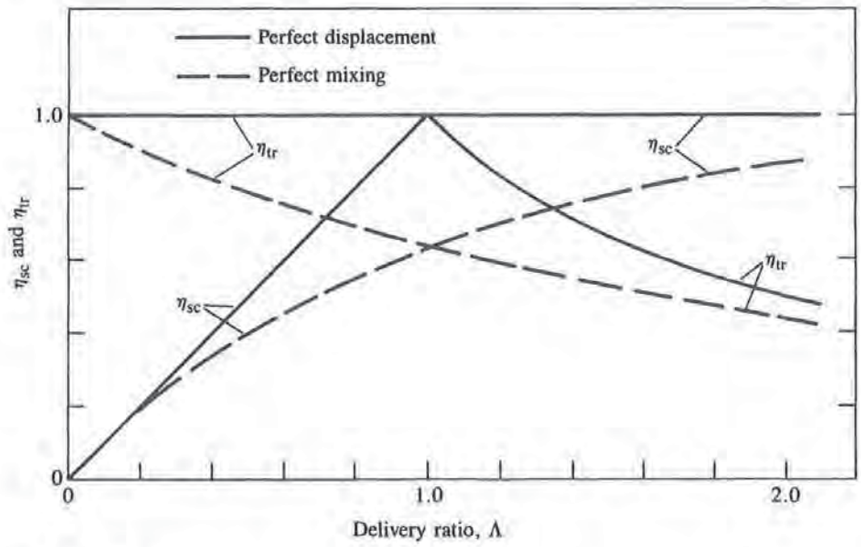


FIGURE 6-27
Scavenging efficiency η_{sc} and trapping efficiency η_{tr} versus delivery ratio Λ for perfect displacement and complete mixing models.

An additional possibility is the direct flow of fresh mixture through the cylinder into the exhaust without entraining burned gases. This is called *short-circuiting*; it is obviously undesirable since some fresh air or mixture is wasted. There is no simple model for this process. When short-circuiting occurs, lower scavenging efficiencies result even though the volume occupied by the short-circuiting flow through the cylinder does displace an equal volume of the burned gases. Another phenomenon which reduces scavenging efficiency is the formation of *pockets* or *dead zones* in the cylinder volume where burned gases can become trapped and escape displacement or entrainment by the fresh scavenging flow. These unscavenged zones are most likely to occur in regions of the cylinder that remain secluded from the main fresh mixture flow path.

6.6.3 Actual Scavenging Processes

Several methods have been developed for determining what occurs in actual cylinder scavenging processes.[27] Accurate measurement of scavenging efficiency is difficult due to the problem of measuring the trapped air mass. Estimation of η_{sc} from indicated mean effective pressure and from gas sampling are the most reliable methods.[7] Flow visualization experiments[28–30] in liquid analogs of the cylinder and flow velocity mapping techniques[31] have proved useful in providing a qualitative picture of the scavenging flow field and identifying problems such as short-circuiting and dead volumes.

Flow visualization studies indicate the key features of the scavenging process. Figure 6-28 shows a sequence of frames from a movie of one liquid scavenging another in a model of a large two-stroke cycle loop-scavenged

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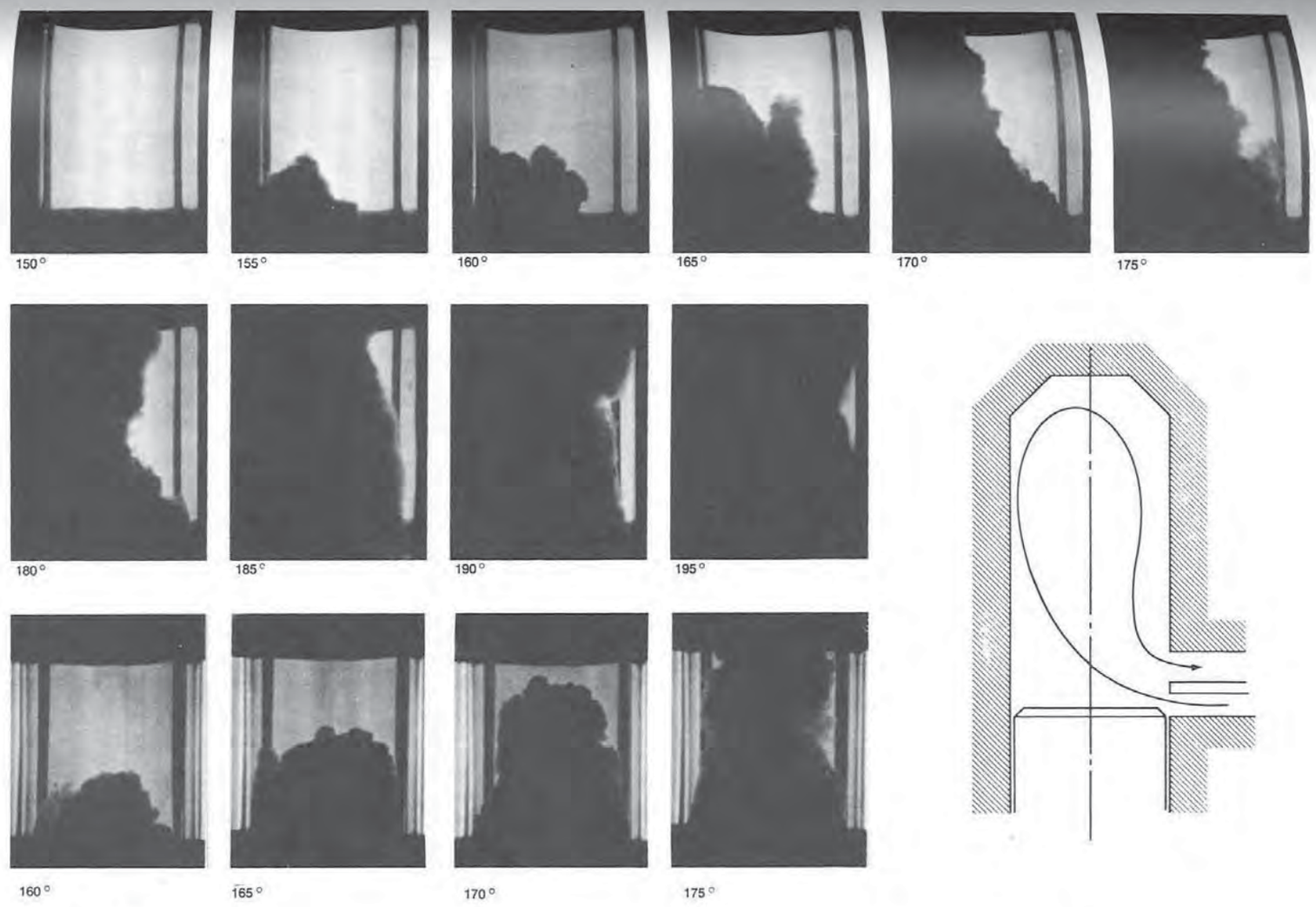


FIGURE 6-28
Photos of one fluid scavenging another in liquid analog experiments in model loop-scavenged engine cylinder. Top two rows: View perpendicular to scavenging loop. Bottom row: Orthogonal views. Dark denser fluid displacing light less dense fluid.[29]

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diesel.[29] The physical variables were scaled to maintain the same values of the appropriate dimensionless numbers for the liquid analog flow and the real engine flow. The density of the liquid representing air (which is dark) was twice the density of the liquid representing burned gas (which is clear). Early in the scavenging process, the fresh air jets penetrate into the burned gas and displace it first toward the cylinder head and then toward the exhaust ports (the schematic gives the location of the ports). During this initial phase, the outflowing gas contains no air; pure displacement of the burned gas from the cylinder is being achieved. Then short-circuiting losses start to occur, due to the damming-up or buildup of fresh air on the cylinder wall opposite the exhaust ports. The short-circuiting fluid flows directly between the scavenge ports and the exhaust ports above them. Since this damming-up of the inflowing fresh air back toward the exhaust ports continues, short-circuiting losses will also continue. While the scavenging front remains distinct as it traverses the cylinder, its turbulent character indicates that mixing between burned gas and air across the front is taking place. For both these reasons (short-circuiting and short-range mixing), the outflowing gas, once the "displacement" phase is over, contains an increasing amount of fresh air.

Outflowing fluid composition measurements from this model study of a Sulzer two-stroke loop-scavenged diesel engine confirm this sequence of events. At 24 crank angle degrees after the onset of scavenging, fresh air due to short-circuiting was detected in the exhaust. At the time the displacement front reached the exhaust port (65° after the onset of scavenging), loss of fresh air due to scavenging amounted to 13 percent of the scavenge air flow. The actual plot of degree of purity (or η_{sc}) versus delivery ratio (Λ) closely followed the perfect displacement line for $\Lambda < 0.4$. For $\Lambda > 0.4$, the shape of the actual curve was similar in shape to the complete mixing curve.

Engine tests confirm these results from model studies. Initially, the exhausted gas contains no fresh air or mixture; only burned gas is being displaced from the cylinder. However, within the cylinder both displacement *and* mixing at the interface between burned gas and fresh gas are occurring. The departure from perfect scavenging behavior is evident when fresh mixture first appears in the exhaust. For loop-scavenged engines this is typically when $\Lambda \approx 0.4$. For uniflow scavenging this perfect scavenging phase lasts somewhat longer; for cross-scavenging it is over sooner (because the short-circuiting path is shorter).

The mixing that occurs is short-range mixing, not mixing throughout the cylinder volume. The jets of scavenging mixture, on entering the cylinder, mix readily with gases in the immediate neighborhood of the jet efflux. More efficient scavenging—i.e., less mixing—is obtained by reducing the size of the inlet ports while increasing their number.[32] It is important that the jets from the inlet ports slow down significantly once they enter the cylinder. Otherwise the scavenging front will reach the exhaust ports or valves too early. The jets are frequently directed to impinge on each other or against the cylinder wall. Swirl in uniflow-scavenged systems may be used to obtain an equivalent result.

The most desirable loop-scavenging flow is illustrated in Fig. 6-29. The

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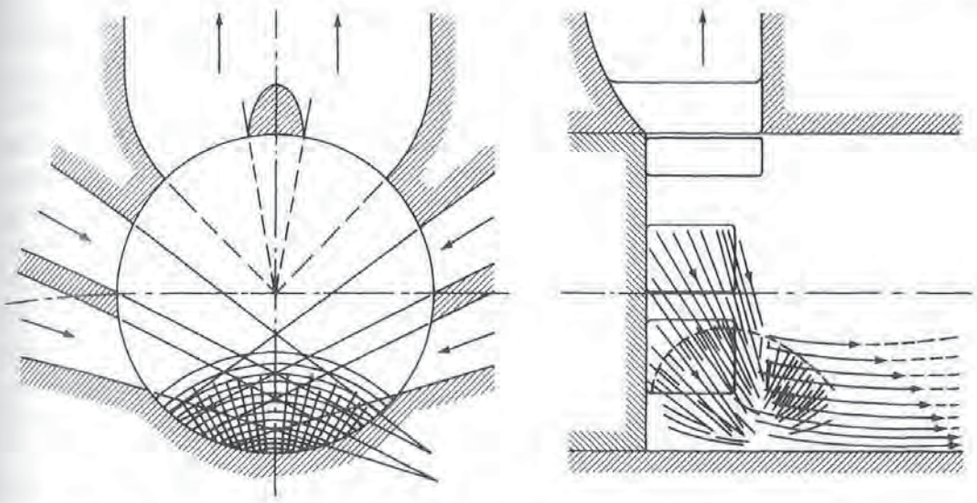


FIGURE 6-29
Desirable air flow in loop-scavenged engine: air from the entering jets impinges on far cylinder wall and flows toward the cylinder head.[31]

scavenging jets enter symmetrically with sufficient velocity to fill up about half the cylinder cross section, and thereafter flow at lower velocity along the cylinder wall toward the cylinder head. By proper direction of the scavenging jets it is possible to achieve almost no outflow in the direction of the exhaust from the cross-hatched stagnation zone on the opposite cylinder wall. In fact, measurement of the velocity profile in this region is a good indicator of the effectiveness of the scavenging flow. If the flow along the cylinder wall toward the head is stable, i.e., if its maximum velocity occurs near the wall and the velocity is near zero on the plane perpendicular to the axis of symmetry of the ports (which passes through the cylinder axis), the scavenging flow will follow the desired path. If there are "tongues" of scavenging flow toward the exhaust port, either in the center of the cylinder or along the walls, then significant short-circuiting will occur.[31]

In uniflow-scavenged configurations, the inlet ports are evenly spaced around the full circumference of the cylinder and are usually directed so that the scavenging jets create a swirling flow within the cylinder (see Fig. 6-24). Results of measurements of scavenging front location in rig flow tests of a valved uniflow two-stroke diesel cylinder, as the inlet port angle was varied to give a wide range of swirl, showed that inlet jets directed tangentially to a circle of half the cylinder radius gave the most stable scavenging front profile over a wide range of conditions.[33]

Though the scavenging processes in spark-ignition and diesel two-stroke engines are similar, these two types of engine operate with quite different delivery ratios. In mixture-scavenged spark-ignition engines, any significant expulsion of fresh fluid with the burned gas results in a significant loss of fuel and causes high hydrocarbon emissions as well as loss of the energy expended in pumping the

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flow which passes straight through the cylinder. In diesels the scavenging medium is air, so only the pumping work is lost. One consequence of this is that two-stroke spark-ignition engines are usually crankcase pumped. This approach provides the maximum pressure and thus also the maximum velocity in the scavenging medium at the start of the scavenging process just after the cylinder pressure has blown down; as the crankcase pressure falls during the scavenging process, the motion of the scavenging front within the cylinder also slows down. Figure 6-30 shows the delivery ratio and trapping, charging, and scavenging efficiencies of two crankcase-scavenged spark-ignition engines as a function of engine speed. These quantities depend significantly on intake and exhaust port design and open period and the exhaust system configuration.[34–36] For two-stroke cycle spark-ignition engines, which use crankcase pumping, delivery ratios vary between about 0.5 and 0.8.

Figure 6-31 shows scavenging data typical of large two-stroke diesels.[37] The purity (mass of air in trapped cylinder charge/mass of trapped cylinder charge) is shown as a function of the delivery ratio. The different scavenging configurations have different degrees of effectiveness, with uniflow scavenging being the most efficient. These diesel engines normally operate with delivery ratios in the range 1.2 to 1.4.

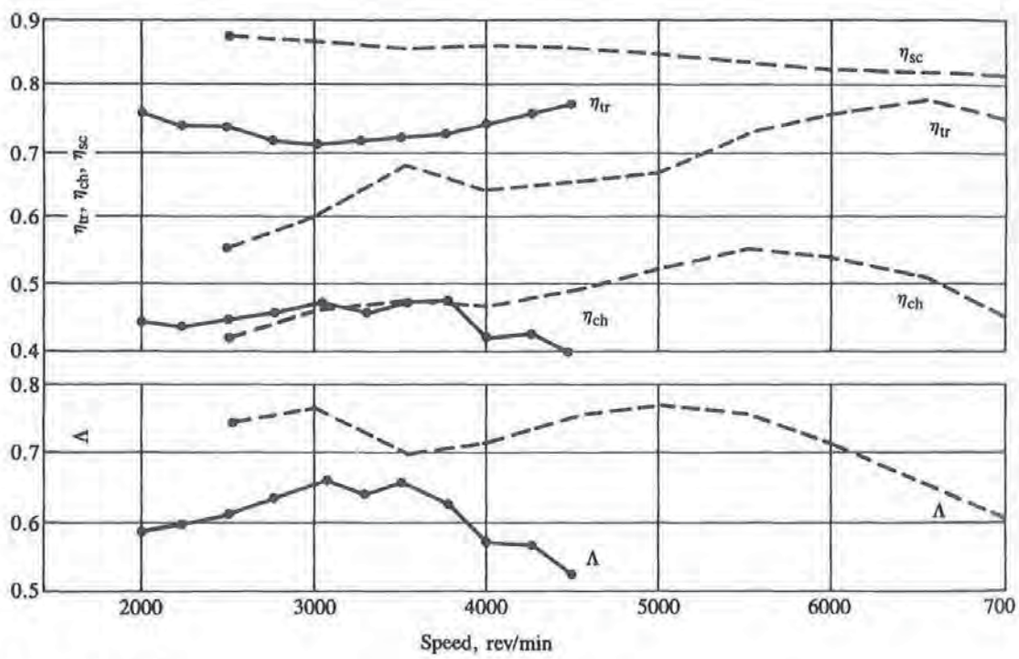


FIGURE 6-30
Delivery ratio Λ, trapping efficiency η_{tr}, charging efficiency η_{ch}, and scavenging efficiency η_{sc}, at full load, as functions of speed for two single-cylinder two-stroke cycle spark-ignition engines. Solid line is 147 cm^3 displacement engine.[34] Dashed line is loop-scavenged 246 cm^3 displacement engine.[35]

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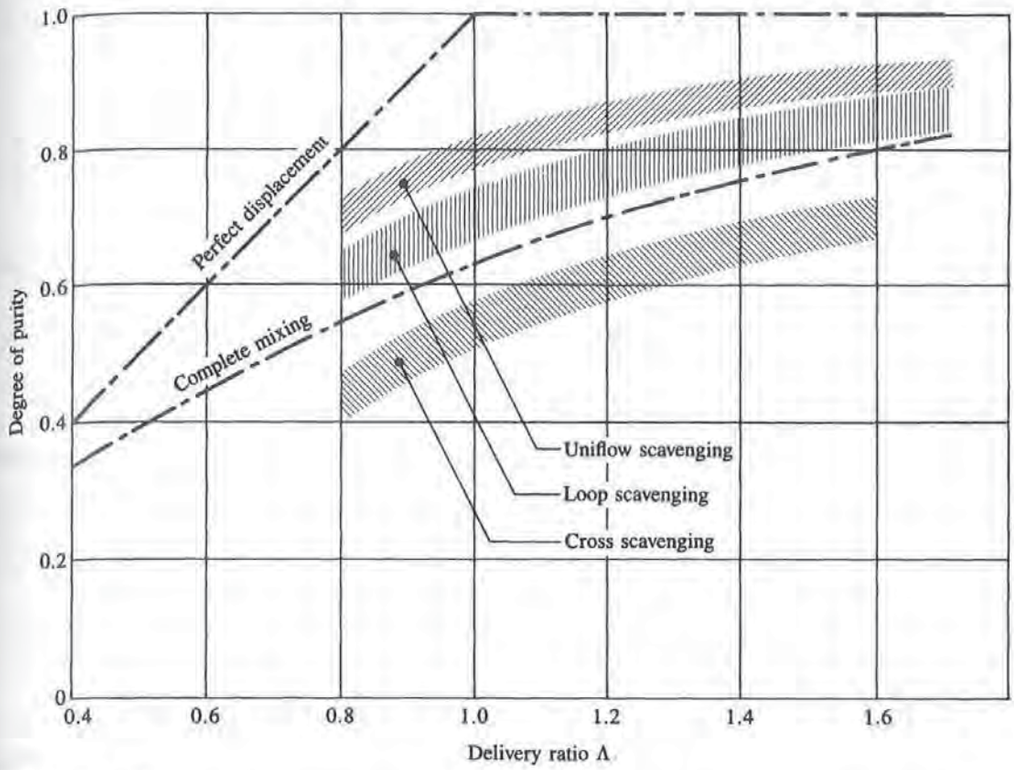


FIGURE 6-31
Purity as a function of delivery ratio Λ for different types of large marine two-stroke diesel engines.[37]

6.7 FLOW THROUGH PORTS

The importance of the intake and exhaust ports to the proper functioning of the two-stroke cycle scavenging process is clear from the discussion in Sec. 6.6. The crank angle at which the ports open, the size, number, geometry, and location of the ports around the cylinder circumference, and the direction and velocity of the jets issuing from the ports into the cylinder all affect the scavenging flow. A summary of the information available on flow through piston-controlled ports can be found in Annand and Roe.[16] Both the flow resistance of the inlet and exhaust port configurations, as well as the details of the flow pattern produced by the port system inside the cylinder during scavenging, are important. Figure 6-32 defines the essential geometrical characteristics of inlet ports. Rectangular ports make best use of the cylinder wall area and give precise timing control. Ports can be tapered, and may have axial and tangential inclination as shown.

Figure 6-33 illustrates the flow patterns expected downstream of piston-controlled inlet ports. For small openings, the flow remains attached to the port walls. For fully open ports with sharp corners the flow detaches at the upstream corners. Both a rounded entry and converging taper to the port help prevent flow detachment within the port. Discharge coefficients for ports have been measured as a function of the open fraction of the port, the pressure ratio across the port,

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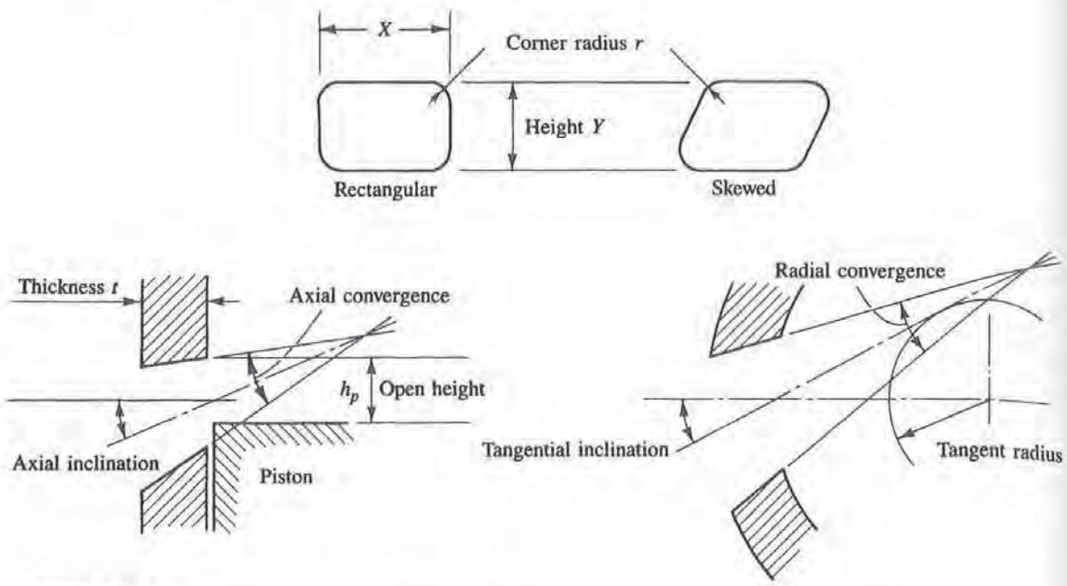


FIGURE 6-32
Parameters which define geometry of inlet ports.[16]

and port geometry and inclination (see Ref. 16 for a detailed summary). The most appropriate reference area for evaluating the discharge coefficient is the open area of the port (see Fig. 6-32). For the open height h_p less than $(Y - r)$ but greater than r this is

$$A_R = Xh_p - 0.43r^2 \tag{6.30}$$

where Y is the port height, X the port width, and r the corner radius. For $h_p = Y$, the reference area is

$$A_R = XY - 0.86r^2 \tag{6.31}$$

The effect of variations in geometry and operating conditions on the discharge coefficient C_D can usually be interpreted by reference to the flow patterns illustrated in Fig. 6-33. The effects of inlet port open fraction and port geometry on C_D are shown in Fig. 6-34: geometry effects are most significant at small and large open fractions.[20] C_D varies with pressure ratio, increasing as the pressure

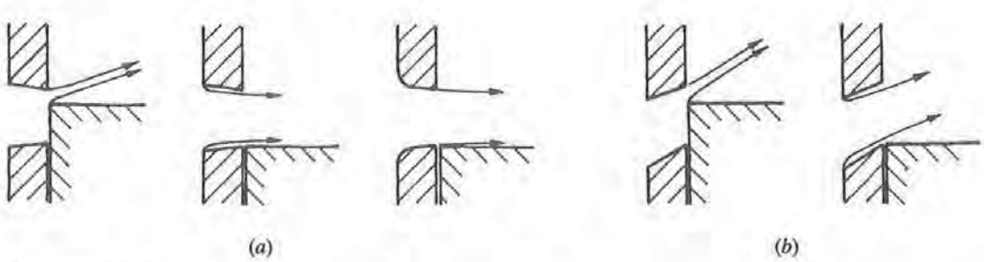


FIGURE 6-33
Flow pattern through piston-controlled inlet ports: (*a*) port axis perpendicular to wall; small opening and large opening with sharp and rounded entry; (*b*) port axis inclined.[16]

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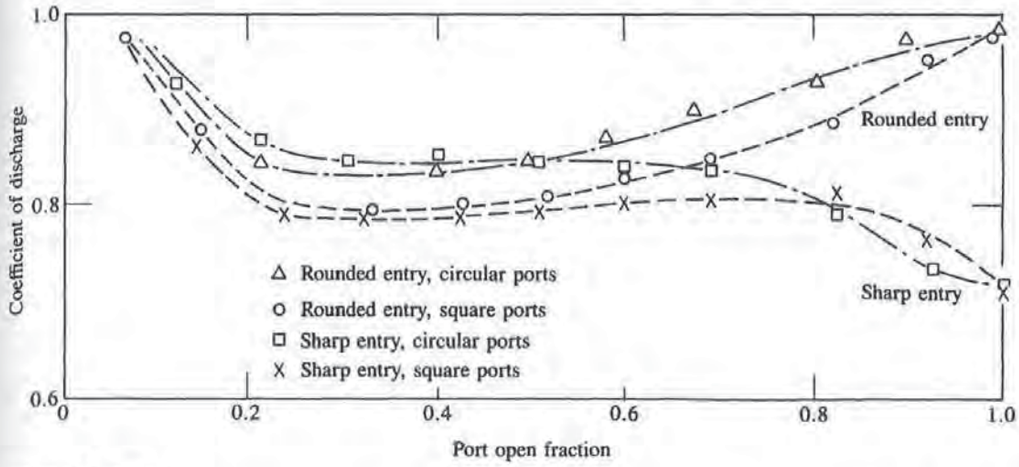


FIGURE 6-34
Discharge coefficients as a function of port open fraction (uncovered height/port height) for different inlet port designs. Pressure ratio across port = 2.35.[20]

ratio increases. Empirical relations that predict this variation with pressure ratio have been developed.[38]

Tangentially inclined inlet ports are used when swirl is desired to improve scavenging or when jet focusing or impingement within the cylinder off the cylinder axis is required (see Sec. 6.6.3). The discharge coefficient decreases as the jet tangential inclination increases. The jet angle and the port angle can deviate significantly from each other depending on the details of the port design and the open fraction.[31]

In piston-controlled exhaust ports, the angle of the jet from a thin-walled exhaust port increases as indicated in Fig. 6-35.[31] In thick ports, the walls are

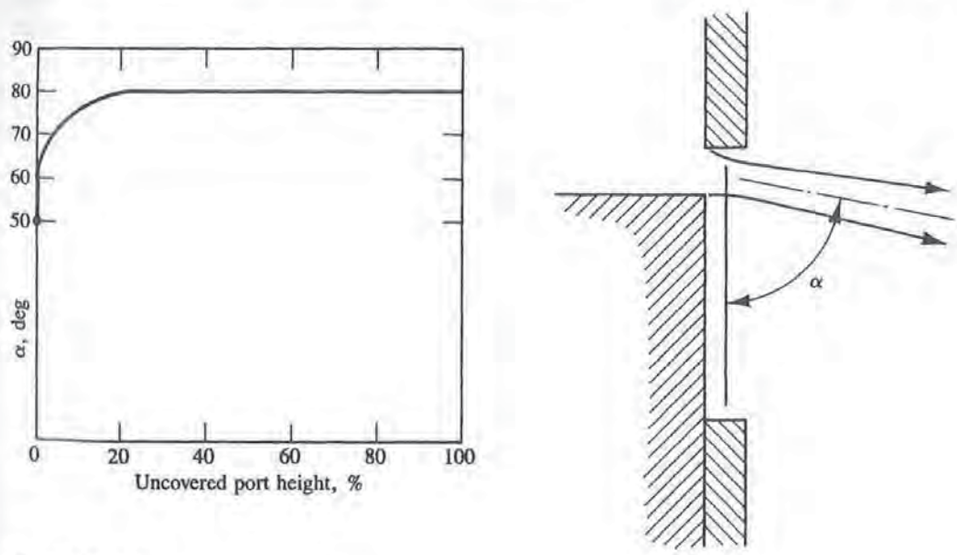


FIGURE 6-35
Angle of jet exiting exhaust port as a function of open port height.[31]

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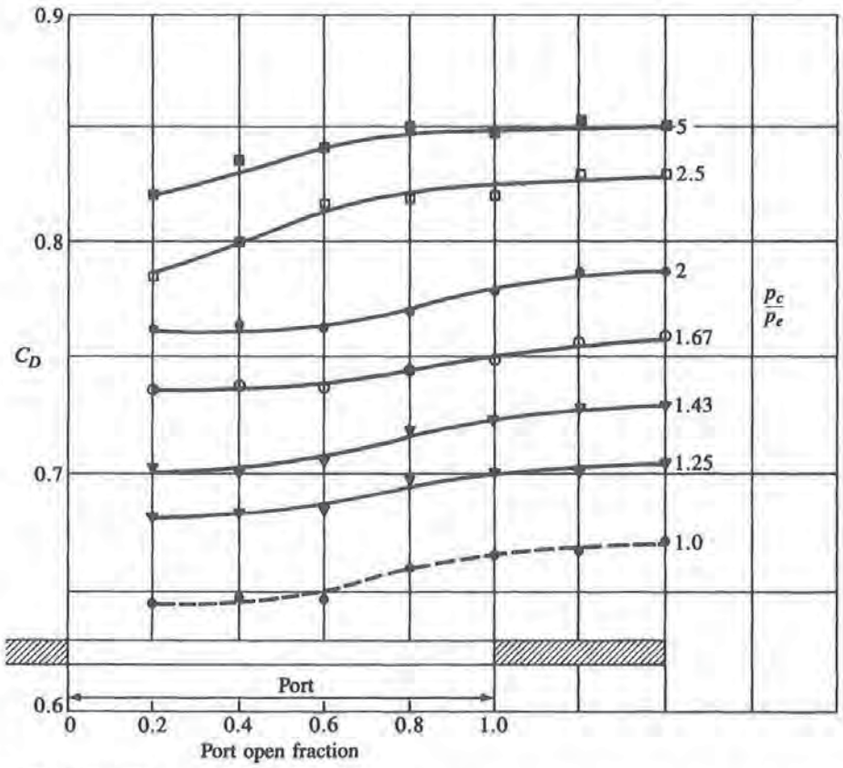


FIGURE 6-36
Discharge coefficient of a single rectangular exhaust port (7.6 mm wide × 12.7 mm high) in the wall of a 51-mm bore cylinder as a function of open fraction and pressure ratio. Steady-flow rig tests at 21°C. p_c = cylinder pressure, p_e = exhaust system pressure.[39]

usually tapered to allow the outward flow to diffuse. The pressure ratio across the exhaust ports varies substantially during the exhaust process. The pressure ratio has a significant effect on the exhaust port discharge coefficient, as shown in Fig. 6-36. The changes in exit jet angle and separation point explain the effects of increasing open fraction and pressure ratio. The discharge coefficient also increases modestly with increasing gas temperature.[39]

6.8 SUPERCHARGING AND TURBOCHARGING

6.8.1 Methods of Power Boosting

The maximum power a given engine can deliver is limited by the amount of fuel that can be burned efficiently inside the engine cylinder. This is limited by the amount of air that is introduced into each cylinder each cycle. If the inducted air is compressed to a higher density than ambient, prior to entry into the cylinder, the maximum power an engine of fixed dimensions can deliver will be increased. This is the primary purpose of supercharging; Eqs. (2.39) to (2.41) show how power, torque, and mean effective pressure are proportional to inlet air density.

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The term *supercharging* refers to increasing the air (or mixture) density by increasing its pressure prior to entering the engine cylinder. Three basic methods are used to accomplish this. The first is *mechanical supercharging* where a separate pump or blower or compressor, usually driven by power taken from the engine, provides the compressed air. The second method is *turbocharging*, where a turbocharger—a compressor and turbine on a single shaft—is used to boost the inlet air (or mixture) density. Energy available in the engine's exhaust stream is used to drive the turbocharger turbine which drives the turbocharger compressor which raises the inlet fluid density prior to entry to each engine cylinder. The third method—*pressure wave supercharging*—uses wave action in the intake and exhaust systems to compress the intake mixture. The use of intake and exhaust manifold tuning to increase volumetric efficiency (see Sec. 6.2.2) is one example of this method of increasing air density. An example of a pressure wave supercharging device is the Comprex, which uses the pressure available in the exhaust gas stream to compress the inlet mixture stream by direct contact of the fluids in narrow flow channels (see Sec. 6.8.5). Figure 6-37 shows typical arrangements of the different supercharging and turbocharging systems. The most common arrangements use a mechanical supercharger (Fig. 6-37*a*) or turbocharger (Fig. 6-37*b*). Combinations of an engine-driven compressor and a turbocharger (Fig. 6-37*c*) are used (e.g., in large marine engines; Fig. 1-24). Two-stage turbocharging (Fig. 6-37*d*) is one viable approach for providing very high boost pressures (4 to 7 atm) to obtain higher engine brake mean effective pressures. Turbocompounding, i.e., use of a second turbine in the exhaust directly geared to the engine drive shaft (Fig. 6-37*e*), is an alternative method of increasing engine power (and efficiency). Charge cooling with a heat exchanger (often called an aftercooler or intercooler) after compression, prior to entry to the cylinder, can be used to increase further the air or mixture density as shown in Fig. 6-37*f*.

Supercharging is used in four-stroke cycle engines to boost the power per unit displaced volume. Some form of supercharging is necessary in two-stroke cycle engines to raise the fresh air (or mixture) pressure above the exhaust pressure so that the cylinder can be scavenged effectively. With additional boost in two-stroke cycle engines, the power density is also raised. This section reviews the operating characteristics of the blowers, compressors, turbines, and wave-compression devices used to increase inlet air or mixture density or convert exhaust-gas availability to work. The operating characteristics of supercharged and turbocharged engine systems are discussed in Chap. 15.

6.8.2 Basic Relationships

Expressions for the work required to drive a blower or compressor and the work produced by a turbine are obtained from the first and second laws of thermodynamics. The first law, in the form of the steady flow energy equation, applied to a control volume around the turbomachinery component is

$$\dot{Q} - \dot{W} = \dot{m}\left[\left(h + \frac{C^2}{2} + gz\right)_{\text{out}} - \left(h + \frac{C^2}{2} + gz\right)_{\text{in}}\right] \tag{6.32}$$

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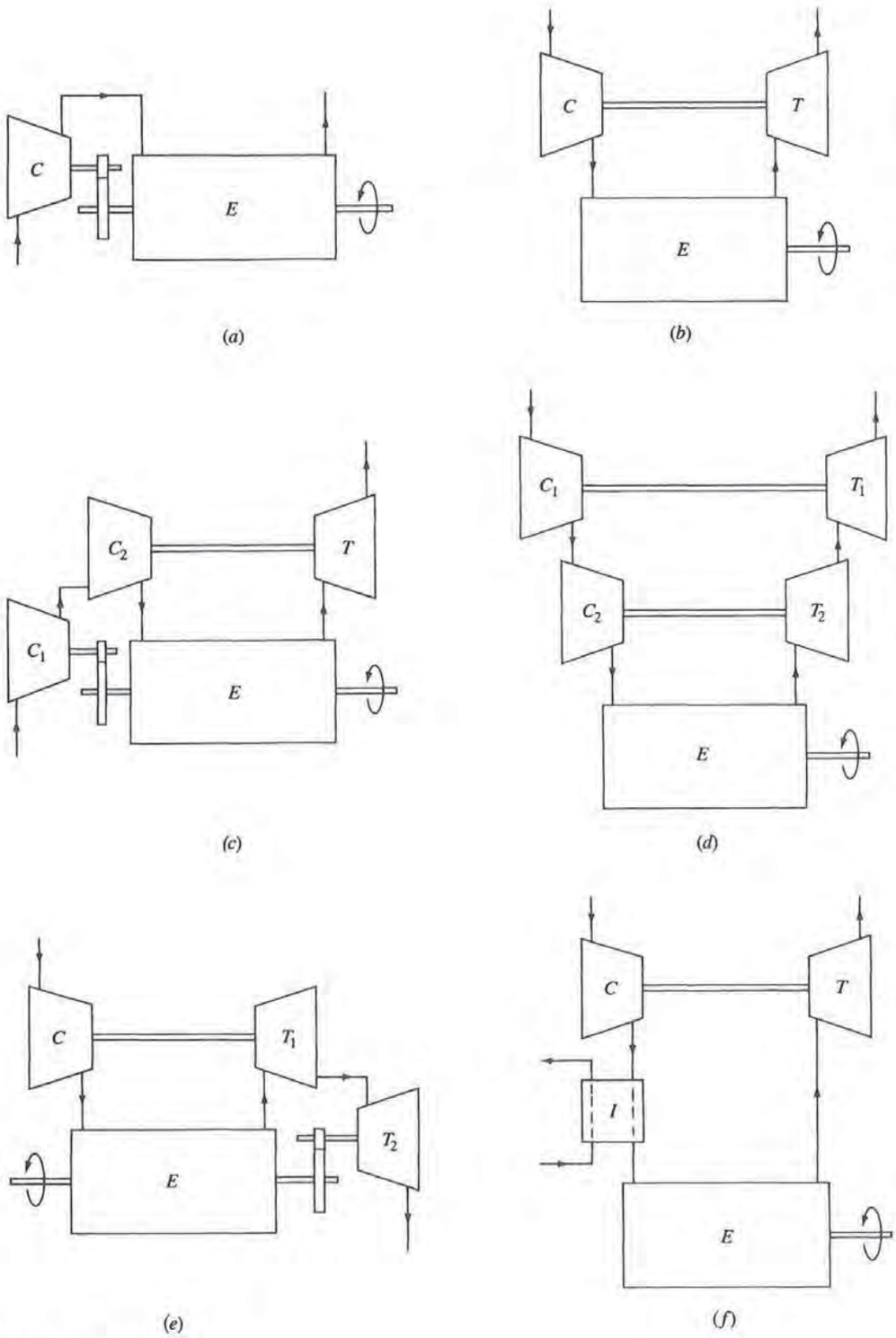


FIGURE 6-37
Supercharging and turbocharging configurations: (*a*) mechanical supercharging; (*b*) turbocharging; (*c*) engine-driven compressor and turbocharger; (*d*) two-stage turbocharging; (*e*) turbocharging with turbocompounding; (*f*) turbocharger with intercooler. *C* Compressor, *E* Engine, *I* Inter-cooler, *T* Turbine.

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where $\dot{Q}$ is the heat-transfer rate into the control volume, $\dot{W}$ is the shaft work-transfer rate out of the control volume, $\dot{m}$ is the mass flow, h is the specific enthalpy, $C^2/2$ is the specific kinetic energy, and gz is the specific potential energy (which is not important and can be omitted).

A *stagnation* or *total enthalpy*, h_0, can be defined as

$$h_0 = h + \frac{C^2}{2} \tag{6.33}$$

For an ideal gas, with constant specific heats, a *stagnation* or *total temperature* follows from Eq. (6.33):

$$T_0 = T + \frac{C^2}{2c_p} \tag{6.34}$$

A *stagnation* or *total pressure* is also defined: it is the pressure attained if the gas is isentropically brought to rest:

$$p_0 = p\left(\frac{T_0}{T}\right)^{\gamma/(\gamma-1)} \tag{6.35}$$

$\dot{Q}$ in Eq. (6.32) for pumps, blowers, compressors, and turbines is usually small enough to be neglected. Equation (6.32) then gives the work-transfer rate as

$$-\dot{W} = \dot{m}(h_{0,\,\text{out}} - h_{0,\,\text{in}}) \tag{6.36}$$

A component efficiency is used to relate the actual work-transfer rate to the work-transfer rate required (or produced) by an equivalent reversible adiabatic device operating between the same pressures. The second law is then used to determine this reversible adiabatic work-transfer rate, which is that occurring in an isentropic process.

For a compressor, the *compressor isentropic efficiency* η_C is

$$\eta_C = \frac{\text{reversible power requirement}}{\text{actual power requirement}} \tag{6.37}$$

Figure 6-38 shows the end states of the gas passing through a compressor on an h-s diagram. Both static (p_1, p_2) and stagnation (p_{01}, p_{02}) constant-pressure lines are shown. The total-to-total isentropic efficiency is, from Eq. (6.37),

$$\eta_{\text{CTT}} = \frac{h_{02s} - h_{01}}{h_{02} - h_{01}} \tag{6.38}$$

which, since c_p is essentially constant for air, or fuel-air mixture, becomes

$$\eta_{\text{CTT}} = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} \tag{6.39}$$

Since the process 01 to 02s is isentropic,

$$T_{02s} = T_{01}\left(\frac{p_{02}}{p_{01}}\right)^{(\gamma-1)/\gamma}$$

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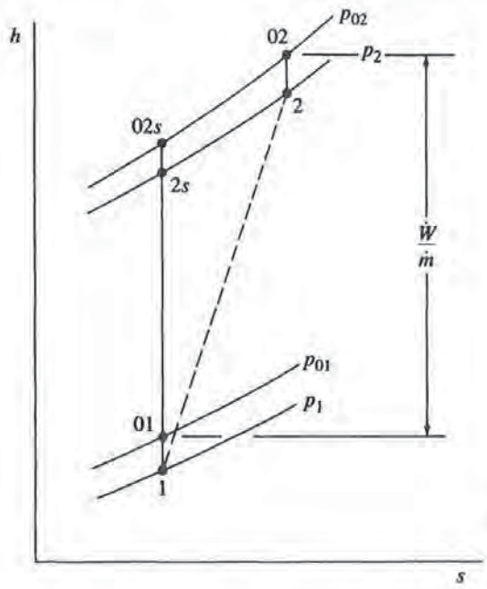


FIGURE 6-38
Enthalpy-entropy diagram for compressor. Inlet state 01, exit state 2; equivalent isentropic compressor exit state 2*s*.

Equation (6.39) becomes

$$\eta_{CTT} = \frac{(p_{02}/p_{01})^{(\gamma-1)/\gamma} - 1}{(T_{02}/T_{01}) - 1} \tag{6.40}$$

In deriving Eq. (6.40) it has been tacitly assumed that the kinetic energy pressure head $(p_{02} - p_2)$ can be recovered. In internal combustion engine applications the compressor feeds the engine via a large manifold, and much of this kinetic energy will be dissipated. The blower or compressor should be designed for effective recovery of this kinetic energy before the exit duct. Since the kinetic energy of the gas leaving the compressor is not usually recovered, a more realistic definition of efficiency is based on exit static conditions:[40]

$$\eta_{CTS} = \frac{T_{2s} - T_{01}}{T_{02} - T_{01}} = \frac{(p_2/p_{01})^{(\gamma-1)/\gamma} - 1}{(T_{02}/T_{01}) - 1} \tag{6.41}$$

This is termed the total-to-static efficiency. The basis on which the efficiency is calculated should always be clearly stated.

The work-transfer rate or power required to drive the compressor is obtained by combining Eq. (6.36), the ideal gas model, and Eq. (6.40):

$$-\dot{W}_C = \dot{m}_i c_{p,i}(T_{02} - T_{01}) = \frac{\dot{m}_i c_{p,i} T_{01}}{\eta_{CTT}} \left[\left(\frac{p_{02}}{p_{01}} \right)^{(\gamma-1)/\gamma} - 1 \right] \tag{6.42}$$

where the subscript i denotes inlet mixture properties. If η_{CTS} is used to define the compressor performance, then p_2 replaces p_{02} in Eq. (6.42). Equation (6.42) gives the thermodynamic power requirement. There will also be mechanical losses in

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the blower or compressor. Thus the power required to drive the device, $-\dot{W}_{C,D}$, will be

$$-\dot{W}_{C,D} = -\frac{\dot{W}_C}{\eta_m} \tag{6.43}$$

where η_m is the blower or compressor mechanical efficiency.

Figure 6-39 shows the gas states at inlet and exit to a turbine on an h-s diagram. State 03 is the inlet stagnation state; 4 and 04 are the exit static and stagnation states, respectively. States 4s and 04s define the static and stagnation exit states of the equivalent reversible adiabatic turbine. The *turbine isentropic efficiency* is defined as

$$\eta_T = \frac{\text{actual power output}}{\text{reversible power output}} \tag{6.44}$$

Thus, the total-to-total turbine efficiency is

$$\eta_{TTT} = \frac{h_{03} - h_{04}}{h_{03} - h_{04s}} \tag{6.45}$$

If the exhaust gas is modeled as an ideal gas with constant specific heats, then Eq. (6.45) can be written

$$\eta_{TTT} = \frac{T_{03} - T_{04}}{T_{03} - T_{04s}} = \frac{1 - (T_{04}/T_{03})}{1 - (p_{04}/p_{03})^{(\gamma-1)/\gamma}} \tag{6.46}$$

Note that for exhaust gas over the temperature range of interest, c_p may vary significantly with temperature (see Figs. 4-10 and 4-17).

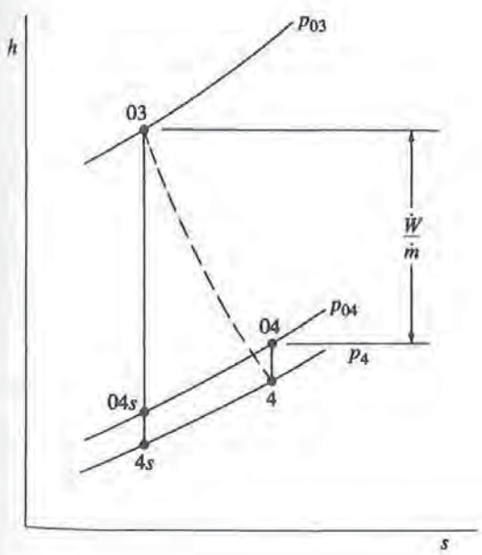


FIGURE 6-39
Enthalpy-entropy diagram for a turbine. Inlet state 03, exit state 4; equivalent isentropic turbine exit state 4s.

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Since the kinetic energy at the exit of a turbocharger turbine is usually wasted, a total-to-static turbine isentropic efficiency, where the reversible adiabatic power output is that obtained between inlet stagnation conditions and the exit static pressure, is more realistic:[40]

$$\eta_{TTS} = \frac{h_{03} - h_{04}}{h_{03} - h_{4s}} = \frac{T_{03} - T_{04}}{T_{03} - T_{4s}} = \frac{1 - (T_{04}/T_{03})}{1 - (p_4/p_{03})^{(\gamma-1)/\gamma}} \tag{6.47}$$

The power delivered by the turbine is given by [Eqs. (6.36) and (6.46)]

$$\dot{W}_T = \dot{m}_e(h_{03} - h_{04}) = \dot{m}_e c_{p,e}(T_{03} - T_{04}) = \dot{m}_e c_{p,e} \eta_{TTT} T_{03}\left[1 - \left(\frac{p_{04}}{p_{03}}\right)^{(\gamma_e-1)/\gamma_e}\right] \tag{6.48}$$

where the subscript e denotes exhaust gas properties. If the total-to-static turbine efficiency (η_{TTS}) is used in the relation for $\dot{W}_T$, then p_4 replaces p_{04} in Eq. (6.48). With a turbocharger, the turbine is mechanically linked to the compressor. Hence, at constant turbocharger speed,

$$-\dot{W}_C = \eta_m \dot{W}_T \tag{6.49}$$

where η_m is the mechanical efficiency of the turbocharger. The mechanical losses are mainly bearing friction losses. The mechanical efficiency is usually combined with the turbine efficiency since these losses are difficult to separate out.

It is advantageous if the operating characteristics of blowers, compressors, and turbines can be expressed in a manner that allows easy comparison between different designs and sizes of devices. This can be done by describing the performance characteristics in terms of dimensionless numbers.[40] The most important dependent variables are: mass flow rate $\dot{m}$, component isentropic efficiency η, and temperature difference across the device ΔT_0. Each of these are a function of the independent variables: $p_{0,\text{in}}$, $p_{0,\text{out}}$ (or p_{out}), $T_{0,\text{in}}$, N(speed), D(characteristic dimension), R(gas constant), γ (c_p/c_v),and μ(viscosity); i.e.,

$$\dot{m},\ \eta,\ \Delta T_0 = f(p_{0,\text{in}},\ p_{0,\text{out}},\ T_{0,\text{in}},\ N,\ D,\ R,\ \gamma,\ \mu) \tag{6.50}$$

By dimensional analysis, these eight independent variables can be reduced to four dimensionless groups:

$$\frac{\dot{m}\sqrt{RT_{0,\text{in}}}}{p_{0,\text{in}}D^2},\ \eta,\ \frac{\Delta T_0}{T_{0,\text{in}}} = f\left(\frac{ND}{\sqrt{RT_{0,\text{in}}}},\ \frac{p_{0,\text{out}}}{p_{0,\text{in}}},\ \frac{\dot{m}}{\mu D},\ \gamma\right) \tag{6.51}$$

The Reynolds number, $\dot{m}/(\mu D)$, has little effect on performance and γ is fixed by the gas. Therefore these variables can be omitted and Eq. (6.51) becomes

$$\frac{\dot{m}\sqrt{RT_{0,\text{in}}}}{p_{0,\text{in}}D^2},\ \eta,\ \frac{\Delta T_0}{T_{0,\text{in}}} = f\left(\frac{ND}{\sqrt{RT_{0,\text{in}}}},\ \frac{p_{0,\text{out}}}{p_{0,\text{in}}}\right) \tag{6.52}$$

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For a particular device, the dimensions are fixed and the value of R is fixed. So it has become the convention to plot

$$\frac{\dot{m}\sqrt{T_{0,\text{in}}}}{p_{0,\text{in}}},\ \eta,\ \frac{\Delta T_0}{T_{0,\text{in}}} = f\left(\frac{N}{\sqrt{T_{0,\text{in}}}},\ \frac{p_{0,\text{out}}}{p_{0,\text{in}}}\right) \tag{6.53}$$

$\dot{m}\sqrt{T_{0,\text{in}}}/p_{0,\text{in}}$ is referred to as the corrected mass flow; $N/\sqrt{T_{0,\text{in}}}$ is referred to as the corrected speed. The disadvantage of this convention of removing D and R is that the groups of variables are no longer dimensionless, and performance plots or maps relate to a specific machine.

Compressor characteristics are usually plotted in terms of the pressure ratio (p_{02}/p_{01}) or (p_2/p_{01}) against the corrected mass flow $(\dot{m}\sqrt{T_{01}}/p_{01})$ along lines of constant corrected speed $(N/\sqrt{T_{01}})$. Contours of constant efficiency are superposed. Similar plots are used for turbines: p_{03}/p_4 against $\dot{m}\sqrt{T_{03}}/p_{03}$ along lines of constant $N/\sqrt{T_{03}}$. Since these occupy a narrow region of the turbine performance map, other plots are often used (see Sec. 6.8.4).

6.8.3 Compressors

Practical mechanical supercharging devices can be classified into: (1) sliding vane compressors, (2) rotary compressors, and (3) centrifugal compressors. The first two types are positive displacement compressors; the last type is an aerodynamic compressor. Four different types of positive displacement compressors are illustrated in Fig. 6-40.

In the sliding vane compressor (Fig. 6-40*a*), deep slots are cut into the rotor to accommodate thin vanes which are free to move radially. The rotor is mounted eccentrically in the housing. As the rotor rotates, the centrifugal forces acting on the vanes force them outward against the housing, thereby dividing the crescent-shaped space into several compartments. Ambient air is drawn through the intake port into each compartment as its volume increases to a maximum. The trapped air is compressed as the compartment volume decreases, and is then discharged through the outlet port. The flow capacity of the sliding vane compressor depends on the maximum induction volume which is determined by the housing cylinder bore, rotor diameter and length, eccentricity, number of vanes, dimensions of the inlet and outlet ports. The actual flow rate and pressure rise at constant speed will be reduced by leakage. Also, heat transfer from the moving vanes and rotor and stator surfaces will reduce compression efficiency unless cooling is used to remove the thermal energy generated by friction between the vanes, and the rotor and stator. The volumetric efficiency can vary between 0.6 and 0.9 depending on the size of the machine, the quality of the design, and the method of lubrication and cooling employed. The displaced volume V_D is given by

$$V_D = \pi\varepsilon l(2r + \varepsilon) \tag{6.54}$$

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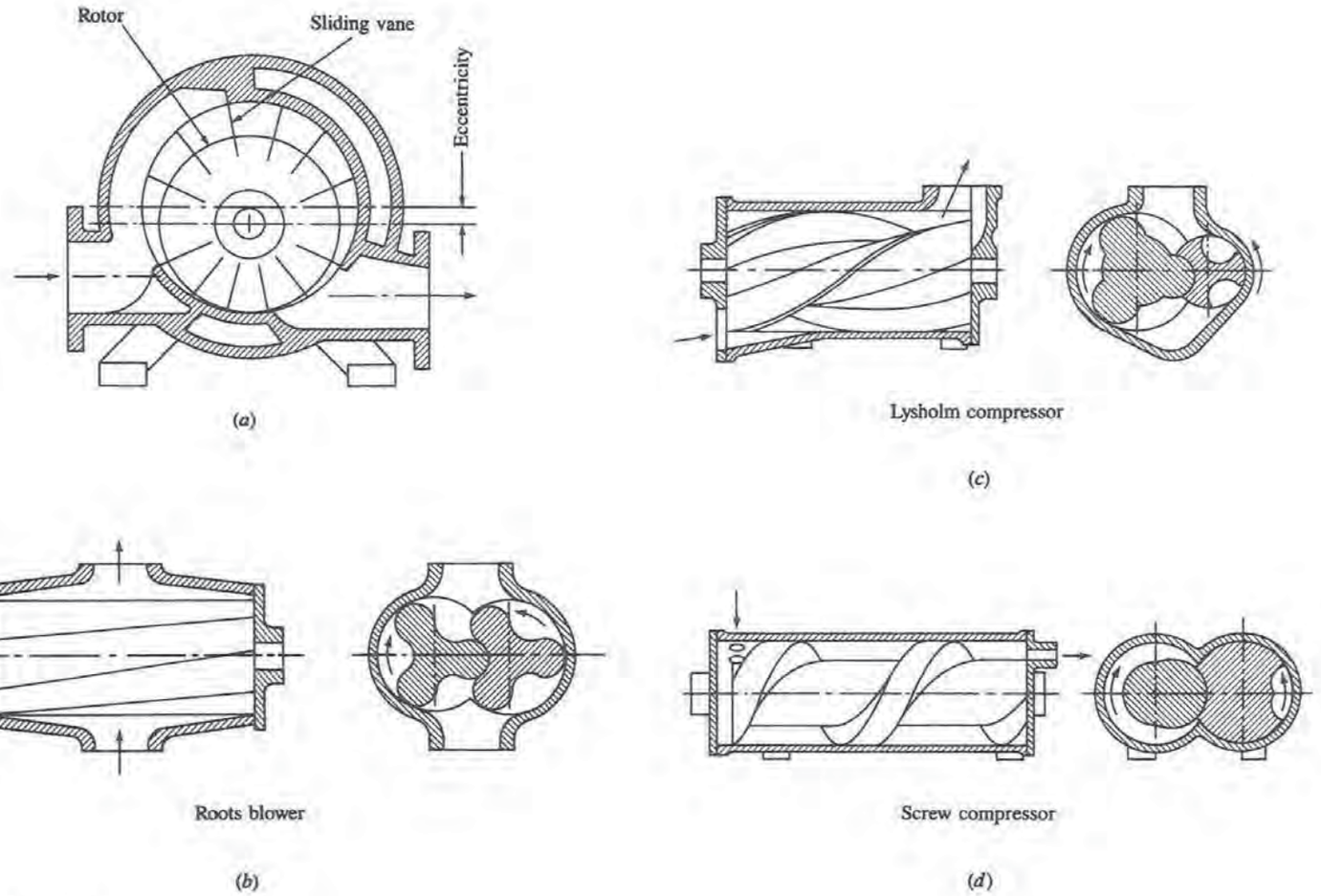


FIGURE 6-40
Positive displacement compressors: (*a*) sliding vane compressor; (*b*) roots blower; (*c*) Lysholm compressor; (*d*) screw compressor.[41]

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where r is the rotor radius, ε the eccentricity, and l the axial length of the compressor. The mass flow rate parameter is

$$\frac{\dot{m}\sqrt{T_0/T_{\text{std}}}}{p_0/p_{\text{std}}} = \text{constant} \times \rho_i \eta_v N \varepsilon l(2r + \varepsilon) \tag{6.55}$$

where η_v is the device volumetric efficiency, N its speed, and the subscripts i, 0 and std refer to inlet, inlet stagnation, and standard atmospheric conditions, respectively. Figure 6-41 shows the performance characteristics of a typical sliding vane compressor. The mass flow rate at constant speed depends on the pressure ratio only through its (weak) effect on volumetric efficiency. The isentropic efficiency is relatively low.[41]

An alternative positive displacement supercharger is the roots blower (Fig. 6-40*b*). The two rotors are connected by gears. The working principles are as follows. Air trapped in the recesses between the rotor lobes and the housing is carried toward the delivery port without significant change in volume. As these recesses open to the delivery line, since the suction side is closed, the trapped air is suddenly compressed by the backflow from the higher-pressure delivery line. This intermittent delivery produces nonuniform torque on the rotor and pressure pulses in the delivery line. Roots blowers are most suitable for small pressure ratios (about 1.2). The volumetric efficiency depends on the running clearances, rotor length, rotational speed, and pressure ratio. In the three-lobe machines shown (two lobes are sometimes used) the volume of each recess V_R is

$$V_R = 0.546R^2 l$$

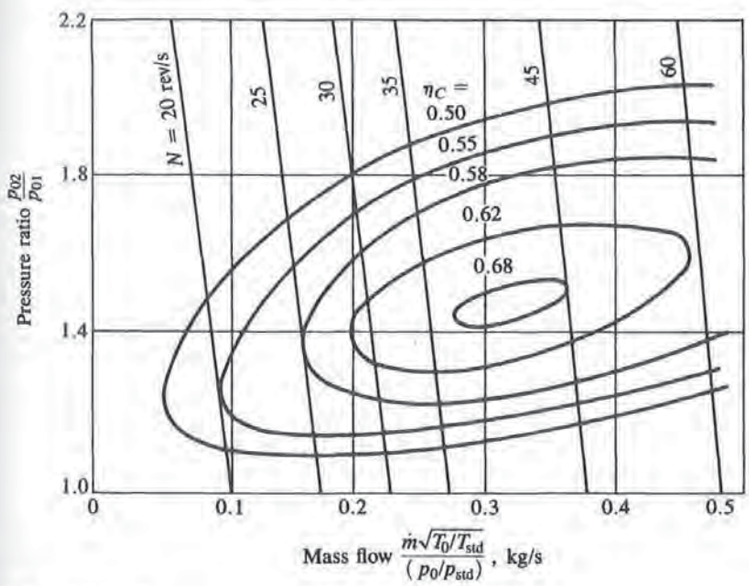


FIGURE 6-41
Performance map for sliding vane compressor.[41]

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where R is the rotor radius and l the blower length. The mass flow parameter is

$$\frac{\dot{m}\sqrt{T_0/T_{std}}}{p_0/p_{std}} = \text{constant} \times \rho_i \eta_v N R^2 l \tag{6.56}$$

A performance map of a typical small roots blower is shown in Fig. 6-42. It is similar in character to that of the sliding vane compressor. At constant speed, the flow rate depends on increasing pressure ratio only through the resulting decrease in volumetric efficiency (Eq. 6.56).[41] The advantage of the roots blower is that its performance range is not limited by surge and choking as is the centrifugal compressor (see below). Its disadvantages are its high noise level, poor efficiency, and large size.[42]

Screw compressors (Fig. 6-40*c* and *d*) must be precision machined to achieve close tolerances between rotating and stationary elements for satisfactory operation. They run at speeds between 3000 and 30,000 rev/min. It is usually necessary to cool the rotors internally. High values of volumetric and isentropic efficiency are claimed.[41]

A *centrifugal compressor* is primarily used to boost inlet air or mixture density coupled with an exhaust-driven turbine in a turbocharger. It is a single-stage radial flow device, well suited to the high mass flow rates at the relatively low pressure ratios (up to about 3.5) required by the engine. To operate efficiently it must rotate at high angular speed. It is therefore much better suited to direct coupling with the exhaust-driven turbine of the turbocharger than to mechanical coupling through a gearbox to the engine for mechanical supercharging.

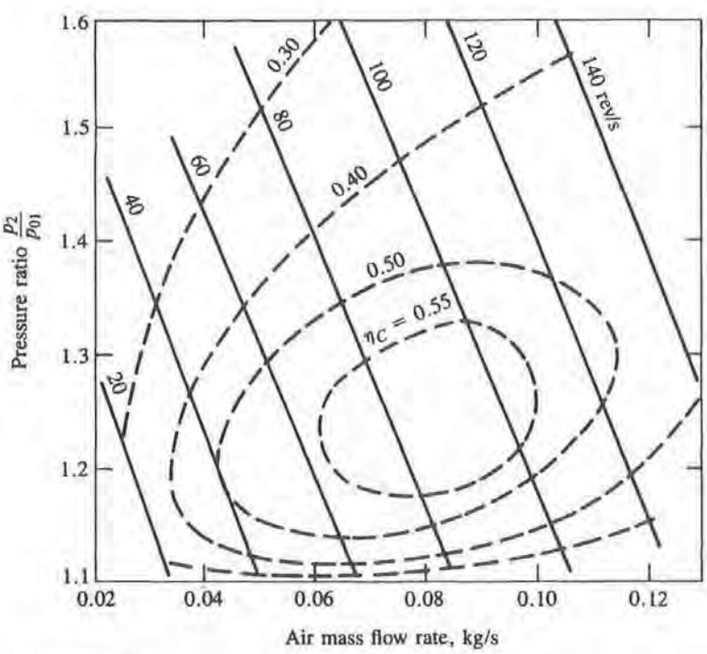


FIGURE 6-42
Performance map at standard inlet conditions for roots blower.[42]

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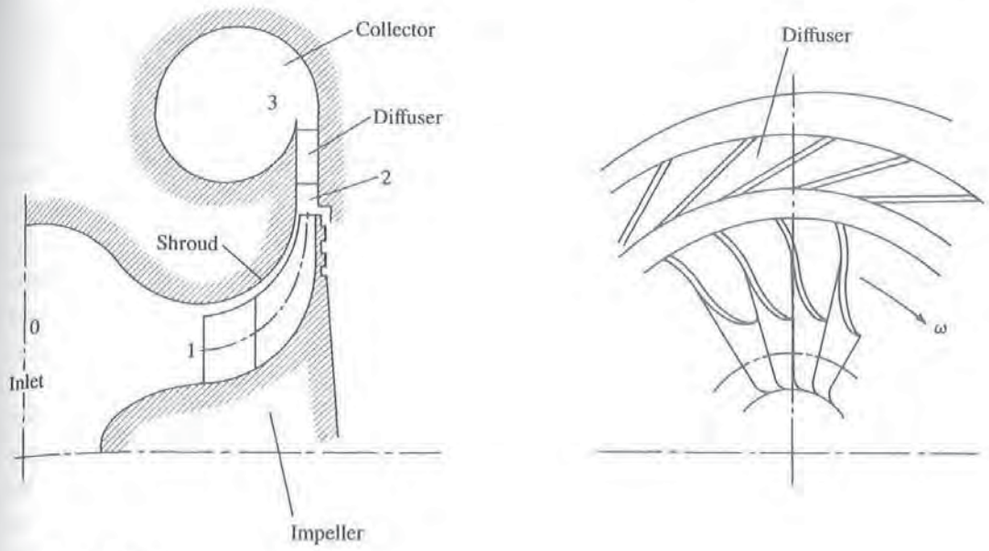


FIGURE 6-43
Schematic of centrifugal compressor.[40]

The centrifugal compressor consists of a stationary inlet casing, a rotating bladed impellor, a stationary diffuser (with or without vanes), and a collector or volute to bring the compressed air leaving the diffuser to the engine intake system (see Fig. 6-43). Figure 6-44 indicates, on an *h-s* diagram, how each component contributes to the overall pressure rise across the compressor. Air at stagnation

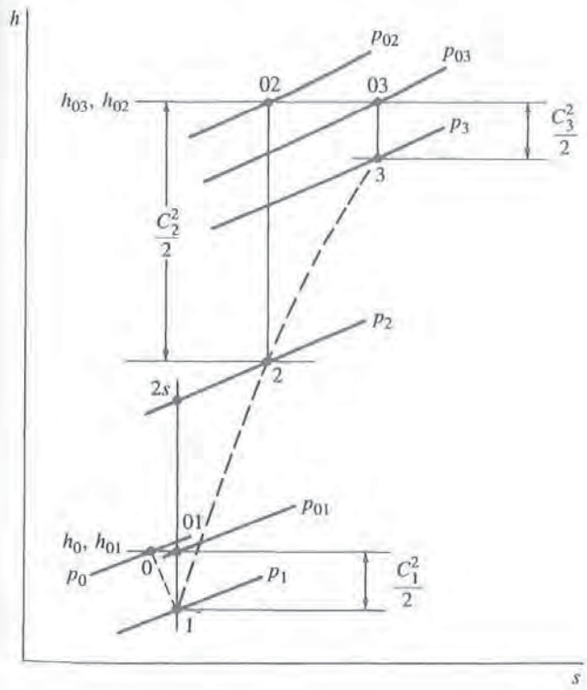


FIGURE 6-44
Enthalpy-entropy diagram for flow through centrifugal compressor.

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state 0 is accelerated in the inlet to pressure p_1 and velocity C_1. The enthalpy change 01 to 1 is $C_1^2/2$. Compression in the impeller flow passages increases the pressure to p_2 and velocity to C_2, corresponding to a stagnation state 02 if all the exit kinetic energy were recovered. The isentropic equivalent compression process has an exit static state 2*s*. The diffuser, 2 to 3, converts as much as practical of the air kinetic energy at exit to the impeller ($C_2^2/2$) to a pressure rise ($p_3 - p_2$) by slowing down the gas in carefully shaped expanding passages. The final state, in the collector, has static pressure p_3, low kinetic energy $C_3^2/2$, and a stagnation pressure p_{03} which is less than p_{02} since the diffusion process is incomplete as well as irreversible.[40]

The work transfer to the gas occurs in the impeller. It can be related to the change in gas angular momentum via the velocity components at the impeller entry and exit, which are shown in Fig. 6-45. Here C_1 and C_2 are the absolute gas velocities, U_1 and U_2 are the tangential blade velocities, and w_1 and w_2 are the gas velocities relative to the impeller all at inlet (1) and exit (2), respectively. The torque T exerted on the gas by the impeller equals the rate of change of angular momentum:

$$T = \dot{m}(r_2 C_{\theta_2} - r_1 C_{\theta_1}) \tag{6.57}$$

The rate of work transfer to the gas is given by

$$-\dot{W}_C = T\omega = \dot{m}\omega(r_2 C_{\theta_2} - r_1 C_{\theta_1}) = \dot{m}(U_2 C_{\theta_2} - U_1 C_{\theta_1}) \tag{6.58}$$

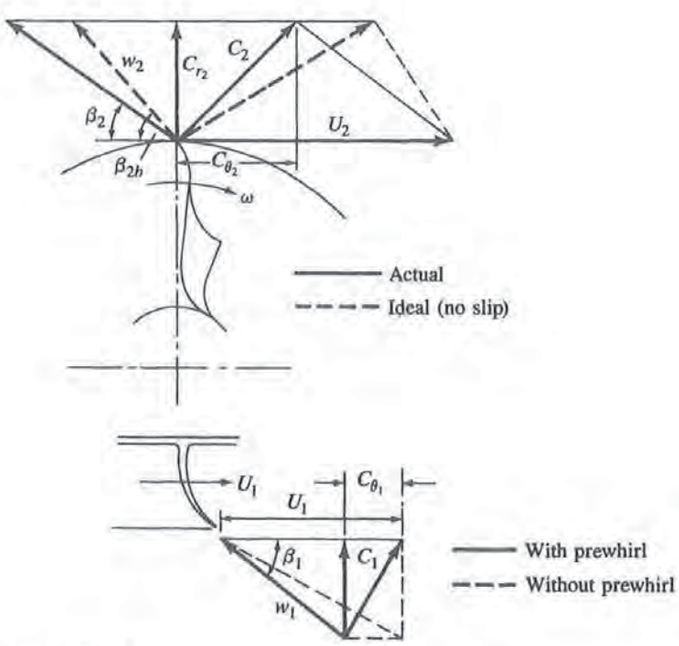


FIGURE 6-45
Velocity diagrams at inlet (1) and exit (2) to centrifugal compressor rotor or impeller.[40]

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This is often called the Euler equation. Normally in compressors the inlet flow is axial so $C_{\theta_1} = 0$. Thus Eq. (6.58) can be written:

$$-\frac{\dot{W}_C}{\dot{m}} = U_2 C_{\theta_2} = U_2\left(1 - \frac{C_{r2}}{U_2}\cot\beta_2\right) \tag{6.59}$$

where β_2 is the backsweep angle. In the ideal case with no slip, β_2 is the blade angle, β_{2b}. In practice, there is slip and β_2 is less than β_{2b}. Many compressors have radial vanes (i.e., $\beta_{2b} = 90°$). A recent trend is backswept vanes ($\beta_{2b} < 90°$) which give higher efficiency. Since work transfer to the gas occurs only in the impeller, the work-transfer rate given by Eq. (6.59) equals the change in stagnation enthalpy ($h_{03} - h_{01}$) in Fig. 6-44 [see Eq. (6.36)].

The operating characteristics of the centrifugal compressor are usually described by a *performance map*. This shows lines of constant compressor efficiency η_C, and constant corrected speed $N/\sqrt{T_{0,\text{in}}}$, on a plot of pressure ratio $p_{0,\text{out}}/p_{0,\text{in}}$ against corrected mass flow $\dot{m}\sqrt{T_{0,\text{in}}}/p_{0,\text{in}}$ [see Eq. (6.53)]. Figure 6-46 indicates the form of such a map. The stable operating range in the center of the map is separated from an unstable region on the left by the *surge line*. When the mass flow is reduced at a constant pressure ratio, local flow reversal eventually occurs in the boundary layer. Further reductions in mass flow cause the flow to reverse completely, causing a drop in pressure. This relieves the adverse pressure gradient. The flow reestablishes itself, builds up again, and the process repeats. Compressors should not be operated in this unstable regime. The

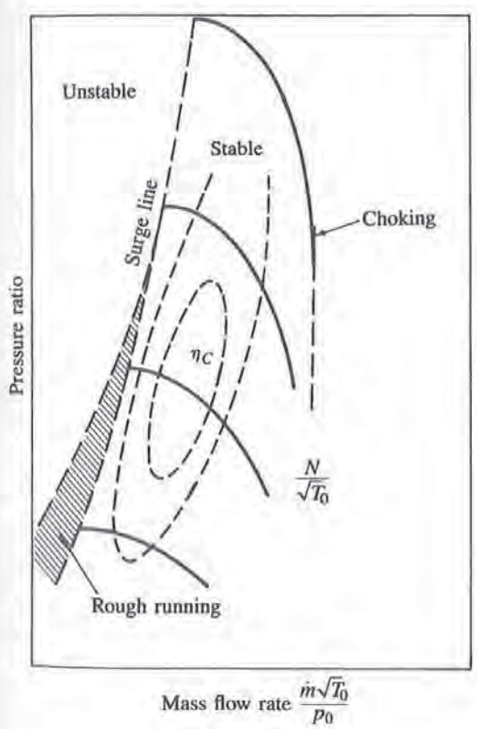


FIGURE 6-46
Schematic of compressor operating map showing stable operating range.[40]

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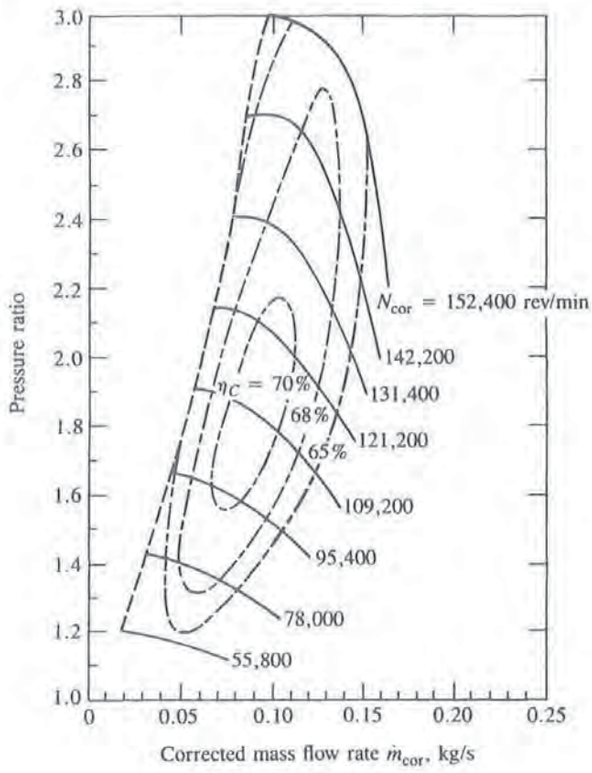


FIGURE 6-47
Centrifugal compressor operating map. Lines of constant corrected speed and compressor efficiency are plotted on a graph of pressure ratio against corrected mass flow.[43]

stable operating regime is limited on the right by choking. The velocities increase as $\dot{m}$ increases, and eventually the flow becomes sonic in the limiting area of the machine. Extra mass flow through the compressor can only be obtained by higher speed. When the diffuser is choked, compressor speed may rise substantially with only a limited increase in the mass flow rate.[40]

Figure 6-47 shows an actual turbocharger compressor performance map. In practice, the map variables corrected speed and mass flow rate are usually defined as[44]

$$N_{cor} = N\left(\frac{T_{ref}}{T_{0,\,in}}\right)^{1/2}$$
$$\dot{m}_{cor} = \dot{m}\left(\frac{T_{0,\,in}}{T_{ref}}\right)^{1/2}\left(\frac{p_{ref}}{p_{0,\,in}}\right) \tag{6.60}$$

where T_{ref} and p_{ref} are standard atmospheric temperature and pressure, respectively. Though the details of different compressor maps vary, their general characteristics are similar. The high efficiency region runs parallel to the surge line (and close to it for vaneless diffusers). A wide flow *range* for the compressor (see Fig. 6-46) is important in turbochargers used for transportation applications.

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6.8.4 Turbines

The turbocharger turbine is driven by the energy available in the engine exhaust. The ideal energy available is shown in Fig. 6-48. It consists of the blowdown work transfer produced by expanding the gas in the cylinder at exhaust valve opening to atmospheric pressure (area *abc*) and (for the four-stroke cycle engine) the work done by the piston displacing the gases remaining in the cylinder after blowdown (area *cdef*).

The reciprocating internal combustion engine is inherently an unsteady pulsating flow device. Turbines can be designed to accept such an unsteady flow, but they operate more efficiently under steady flow conditions. In practice, two approaches for recovering a fraction of the available exhaust energy are commonly used: constant-pressure turbocharging and pulse turbocharging. In *constant-pressure turbocharging*, an exhaust manifold of sufficiently large volume to damp out the mass flow and pressure pulses is used so that the flow to the turbine is essentially steady. The disadvantage of this approach is that it does not make full use of the high kinetic energy of the gases leaving the exhaust port; the losses inherent in the mixing of this high-velocity gas with a large volume of low-velocity gas cannot be recovered. With *pulse turbocharging*, short small-cross-section pipes connect each exhaust port to the turbine so that much of the kinetic energy associated with the exhaust blowdown can be utilized. By suitably grouping the different cylinder exhaust ports so that the exhaust pulses are sequential and have minimum overlap, the flow unsteadiness can be held to an acceptable level. The turbine must be specifically designed for this pulsating flow to achieve adequate efficiencies. The combination of increased energy available at the turbine, with reasonable turbine efficiencies, results in the pulse system being more commonly used for larger diesels.[40] For automotive engines, constant-pressure turbocharging is used.

Two types of turbines are used in turbochargers: radial and axial flow turbines. The radial flow turbine is similar in appearance to the centrifugal compressor; however, the flow is radially inward not outward. Radial flow turbines are

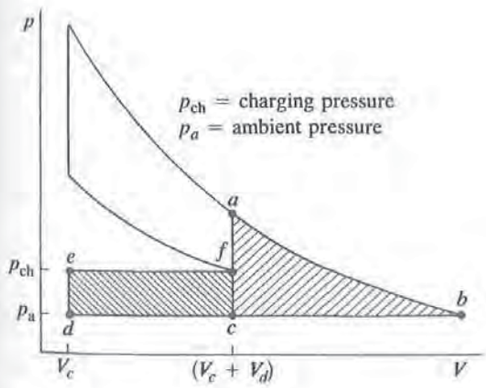


FIGURE 6-48
Constant-volume cycle *p-V* diagram showing available exhaust energy.

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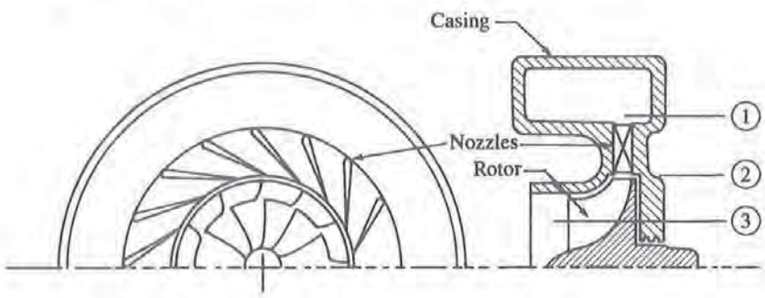


FIGURE 6-49
Schematic of radial flow turbine.

normally used in automotive or truck applications. Larger engines—locomotive, stationary, or marine—use axial flow turbines.

A drawing of a radial flow turbine is shown in Fig. 6-49. It consists of an inlet casing or scroll, a set of inlet nozzles (often omitted with small turbines), and the turbine rotor or wheel. The function of each component is evident from the *h-s* diagram and velocity triangles in Fig. 6-50. The nozzles (01-2) accelerate the flow, with modest loss in stagnation pressure. The drop in stagnation enthalpy, and hence the work transfer, occurs solely in the rotor passages, 2-3: hence, the rotor is designed for minimum kinetic energy $C_3^2/2$ at exit. The velocity triangles at inlet and exit relate the work transfer to the change in angular momentum via the Euler equation:

$$\dot{W}_T = T\omega = \dot{m}\omega(r_2 C_{\theta_2} - r_3 C_{\theta_3}) = \dot{m}(U_2 C_{\theta_2} - U_3 C_{\theta_3}) \tag{6.61}$$

where T is the torque and ω the rotor angular speed. For maximum work transfer the exit velocity should be axial. The work-transfer rate relates to the change in stagnation enthalpy via

$$\dot{W}_T = \dot{m}(h_{02} - h_{03}) = \dot{m}(h_{01} - h_{03}) \tag{6.62}$$

The turbine isentropic efficiency is given by Eqs. (6.44) to (6.47).

Many different types of plots have been used to define radial flow turbine characteristics. Figure 6-51 shows lines of constant corrected speed and efficiency on a plot of pressure ratio versus corrected mass flow rate. As flow rate increases at a given speed, it asymptotically approaches a limit corresponding to the flow becoming choked in the stator nozzle blades or the rotor. For turbines, efficiency is usually presented on a different diagram because the operating regime in Fig. 6-51 is narrow. Figure 6-52 shows an alternative plot for a radial turbine: corrected mass flow rate against corrected rotor speed. On this map, the operating regime appears broader.

A schematic of a turbocharger axial flow turbine is shown in Fig. 6-53. Usually a single stage is sufficient to expand the exhaust gas efficiently through the pressure ratios associated with engine turbocharging. This turbine consists of an annular flow passage, a single row of nozzles or stator blades, and a rotating blade ring. The changes in gas state across each component are similar to those

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of the radial turbine shown in the h-s diagram of Fig. 6-50. The velocity triangles at entry and exit to the rotor, shown in Fig. 6-54, relate the work transfer from the gas to the rotor to the change in angular momentum:

$$\dot{W}_T = \omega T = \dot{m}\omega(r_2 C_{\theta_2} + r_3 C_{\theta_3})$$

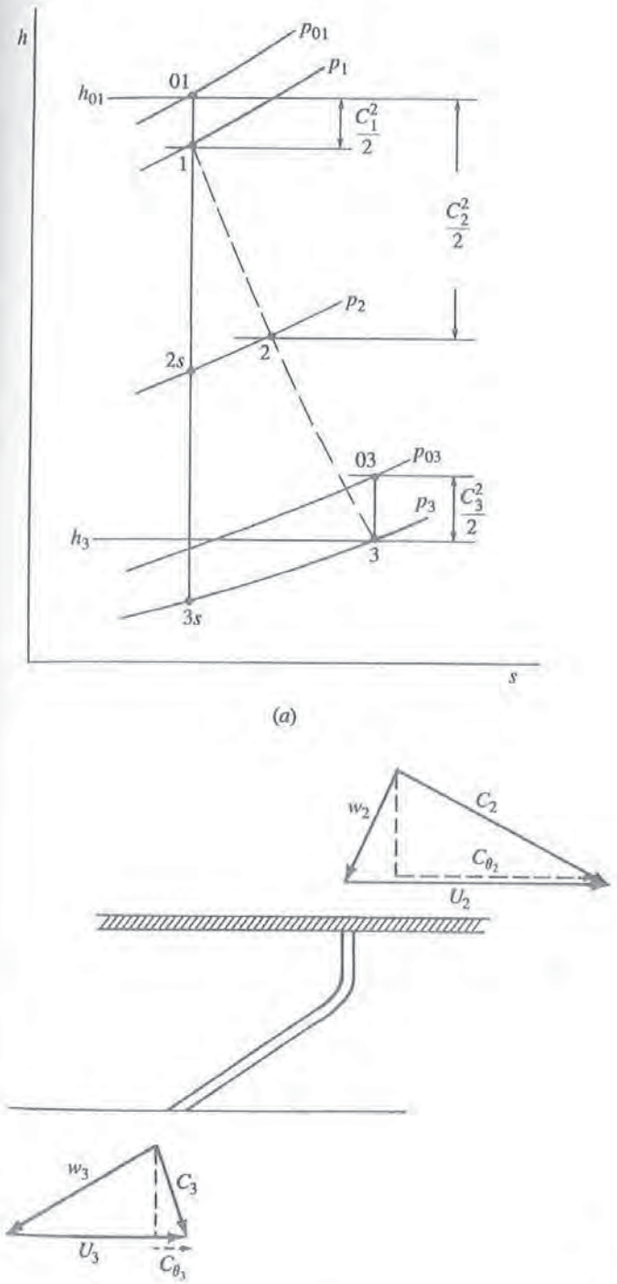


FIGURE 6-50
(*a*) Enthalpy-entropy diagram for radial turbine. (*b*) Velocity diagrams at turbine rotor inlet (2) and exit (3).

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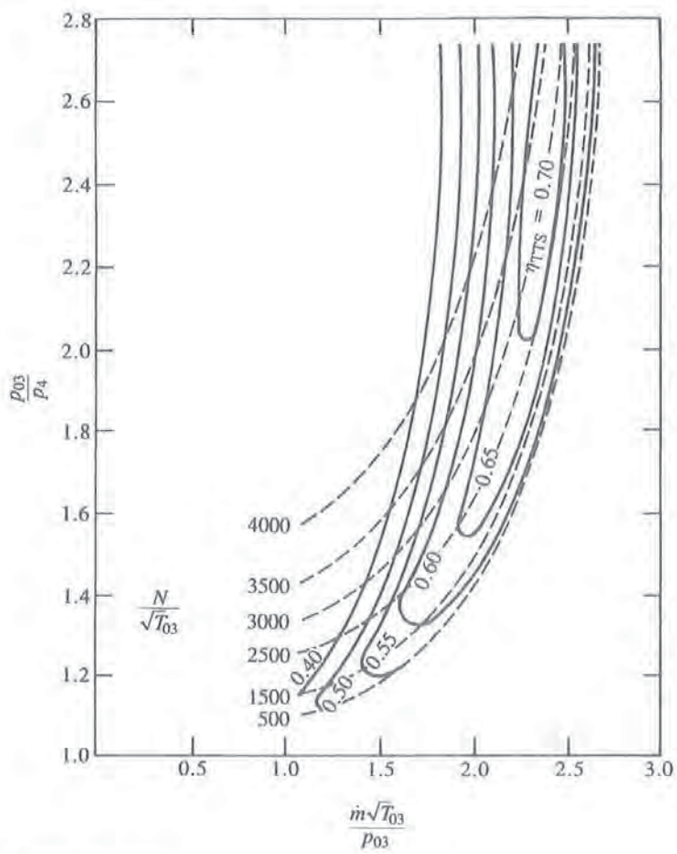


FIGURE 6-51
Radial turbine performance map showing lines of constant corrected speed and efficiency on a plot of pressure ratio versus corrected mass flow rate. T_{03} = turbine inlet temperature (K), p_{03} = turbine inlet pressure (bar), p_4 = turbine exit pressure (bar), $\dot{m}$ = mass flow rate (kg/s), N = speed (rev/min).[40]

Since the mid-radius r_2 usually equals the mid-radius r_3,

$$\dot{W}_T = \dot{m}U(C_{\theta_2} + C_{\theta_3}) = \dot{m}U(C_2 \sin \alpha_2 + C_3 \sin \alpha_3)$$
$$= \dot{m}U(C_{z_2} \tan \beta_2 + C_{z_3} \tan \beta_3) \qquad (6.63)$$

Equation (6.62) relates the work-transfer rate to the stagnation enthalpy change as in the radial turbine.

Figure 6-55 shows axial turbine performance characteristics on the standard dimensionless plot of pressure ratio versus corrected mass flow rate. Here the constant speed lines converge to a single choked flow limit as the mass flow is increased. In the radial turbine, the variation in centrifugal effects with speed cause a noticeable spread in the constant speed lines (Fig. 6-51).

An alternative performance plot for turbines is efficiency versus *blade speed ratio*. This ratio is the blade speed U (at its mean height) for an axial flow turbine

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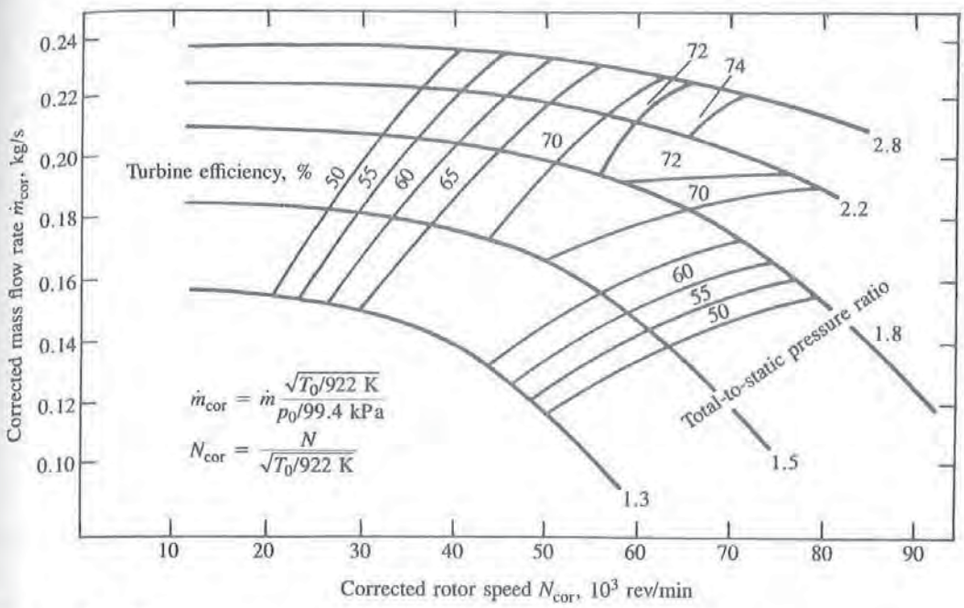


FIGURE 6-52
Alternative radial turbine performance map: corrected mass flow rate is plotted against corrected rotor speed.[45]

or the wheel tip speed for a radial flow turbine, divided by the velocity equivalent of the isentropic enthalpy drop across the turbine stage, C_s; i.e.,

$$\text{Blade speed ratio} = \frac{U}{C_s}$$

where

$$C_s = [2(h_{03} - h_{4s})]^{1/2} \tag{6.64}$$

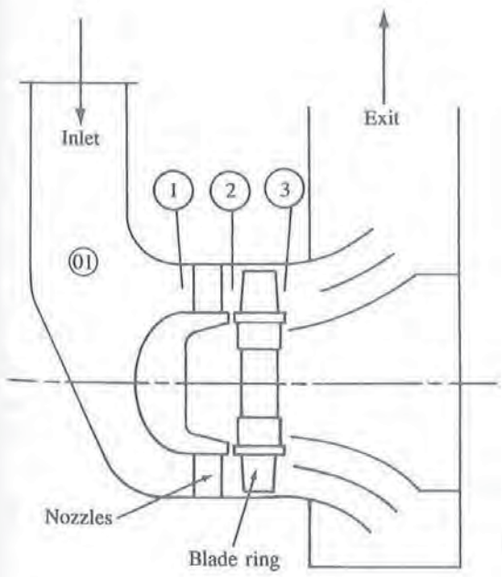


FIGURE 6-53
Schematic of single-stage axial flow turbine.

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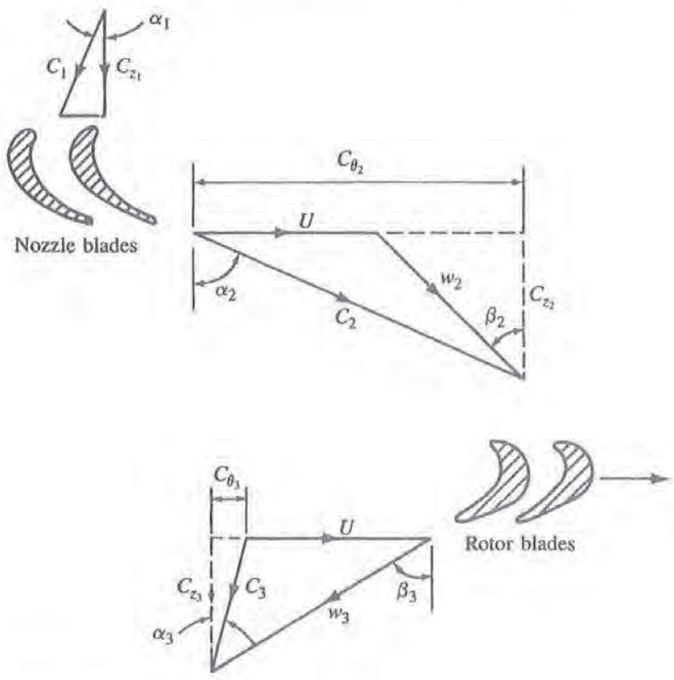


FIGURE 6-54
Velocity diagrams at entry (2) and exit (3) to axial flow turbine blade ring.[40]

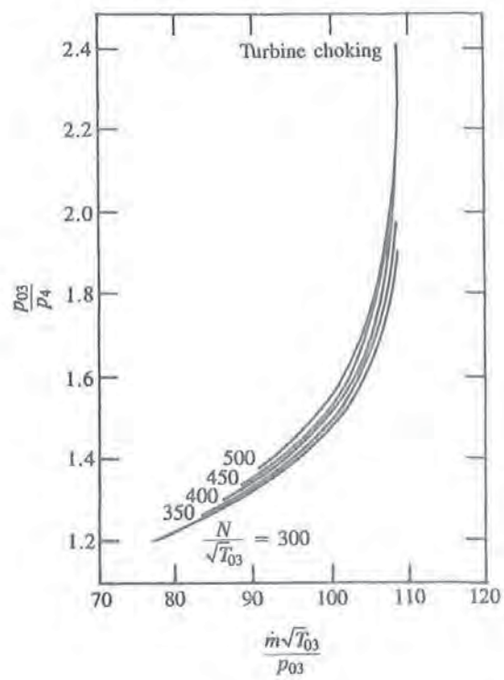


FIGURE 6-55
Axial flow turbine performance map: pressure ratio is plotted against corrected mass flow rate. T_{03} = turbine inlet temperature (K), p_{03} = turbine inlet pressure (bar), p_4 = turbine exit pressure (bar), $\dot{m}$ = mass flow rate (kg/s), N = speed (rev/min).[40]

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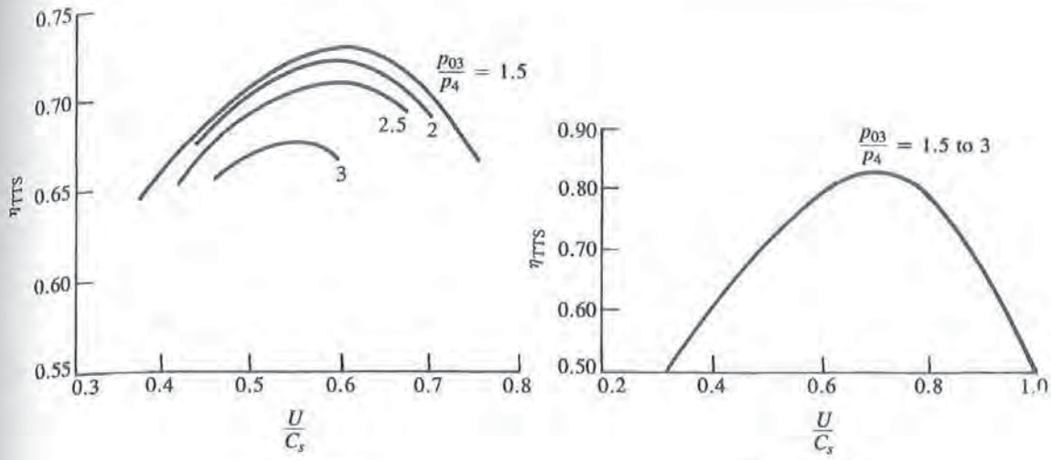


FIGURE 6-56
Plot of turbine total-to-static efficiency versus blade speed ratio U/C_s for (*a*) axial flow and (*b*) radial flow turbines.[40]

This method of displaying performance is useful for matching compressor and turbine wheel size for operation of the turbine at optimum efficiency. Figure 6-56 shows such plots for an axial and radial flow turbine. The peak efficiency can occur for $0.4 < U/C_s < 0.8$, depending on turbine design and application.[40]

For a given turbocharger, the compressor and turbine characteristics are linked. Since the compressor and turbine are on a common shaft with speed N:

$$\frac{N}{\sqrt{T_{01}}} = \frac{N}{\sqrt{T_{03}}} \left(\frac{T_{03}}{T_{01}}\right)^{1/2} \tag{6.65}$$

For $\dot{m}_C = \dot{m}_T = \dot{m}$ (if $\dot{m}_C[1 + (F/A)] = \dot{m}_T$, the equation is easily modified):

$$\frac{\dot{m}\sqrt{T_{01}}}{p_{01}} = \frac{\dot{m}\sqrt{T_{03}}}{p_{03}} \left(\frac{p_{03}}{p_{01}}\right)\left(\frac{T_{01}}{T_{03}}\right)^{1/2} \tag{6.66}$$

Since the compressor and turbine powers are equal in magnitude:

$$h_{02} - h_{01} = \eta_m(h_{03} - h_{04}) \tag{6.67}$$

or, with an ideal gas model,

$$c_{p,C}(T_{02} - T_{01}) = \eta_m c_{p,T}(T_{03} - T_{04}) \tag{6.68}$$

Equation (6.68), with Eqs. (6.40) and (6.46), gives

$$\left(\frac{p_{02}}{p_{01}}\right)^{(\gamma C - 1)/\gamma C} - 1 = \eta_C \eta_T \eta_m \frac{c_{p,T}}{c_{p,C}} \left[1 - \left(\frac{p_4}{p_{03}}\right)^{(\gamma T - 1)/\gamma T}\right] \frac{T_{03}}{T_{01}} \tag{6.69}$$

Assuming that the turbine exit pressure p_4 equals atmospheric pressure p_{01}, the equilibrium or steady-state running lines for constant values of T_{03}/T_{01} can be

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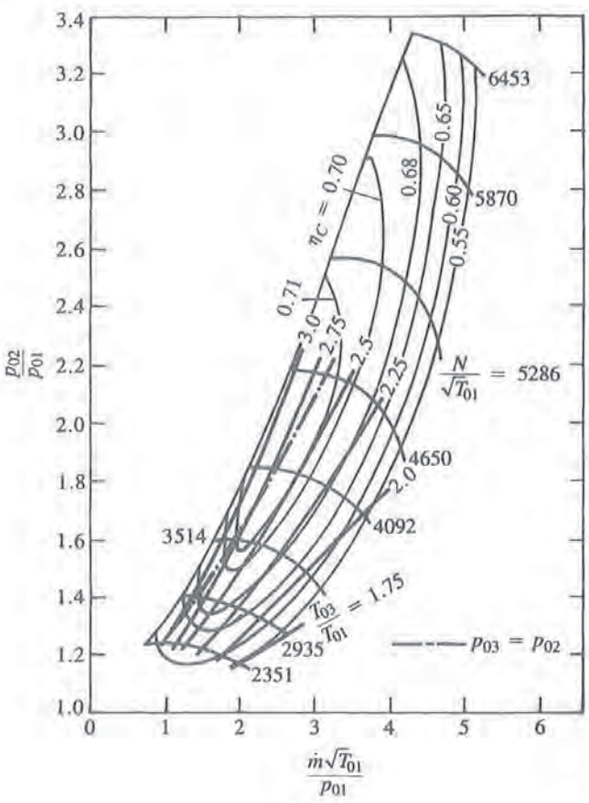


FIGURE 6-57
Steady-state turbocharger operating lines plotted as constant T_{03}/T_{01} lines on compressor map. Turbine characteristics defined by Fig. 6-51. p_{01} = compressor inlet pressure (bar), p_{02} = compressor exit pressure (bar), T_{01} = compressor inlet temperature (K), T_{03} = turbine inlet temperature (K), $\dot{m}$ = mass flow rate (kg/s), N = speed (rev/min).[40]

determined. Figure 6-57 shows an example of such a set of turbocharger characteristics, plotted on a turbocharger compressor map for a radial turbine with characteristics similar to Fig. 6-51. The dash-dot-dash line is for $p_{02} = p_{03}$. To the right of this line, $p_{03} > p_{02}$; to the left of this line $p_{02} > p_{03}$.[40]

The problem of overspeeding the turbocharger and generating very high cylinder pressures often requires that some of the exhaust be bypassed around the turbine. The bypass valve or *wastegate* is usually built into the turbocharger casing. It consists of a spring-loaded valve acting in response to the inlet manifold pressure on a controlling diaphragm. When the wastegate is open, only a portion of the exhaust gases will flow through the turbine and generate power; the remainder passes directly into the exhaust system downstream of the turbine.

6.8.5 Wave-Compression Devices

Pressure wave superchargers make use of the fact that if two fluids having different pressures are brought into direct contact in long narrow channels, equilization of pressure occurs faster than mixing. One such device, the Comprex, has been developed for internal combustion engine supercharging which operates using this principle.[46] It is shown schematically in Fig. 6-58. The working channels of the Comprex are arranged on a rotor or cell wheel (*b*) which is rotated

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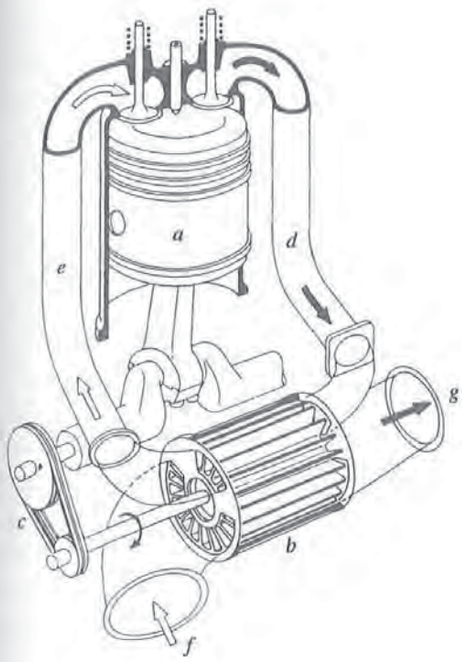


FIGURE 6-58
Schematic of Comprex supercharger.[47] *a* Engine, *b* Cell wheel or rotor, *c* Belt drive, *d* High-pressure exhaust gas (G-HP), *e* High-pressure air (A-HP), *f* Low-pressure air (A-LP), *g* Low-pressure exhaust gas (G-LP)

between two castings by a belt driven from the crankshaft (*c*). There is no contact between the rotor and the casing, but the gaps are kept small to minimize leakage. The belt drive merely overcomes friction and maintains the rotor at a speed proportional to engine speed (usually 4 or 5 times faster): it provides no compression work. One casing (the air casing) contains the passage which brings low-pressure air (*f*) to one set of ports and high-pressure air (*e*) from another set of ports in the rotor-side inner casing. The other casing (the gas casing) connects the high-pressure engine exhaust gas (*d*) to one set of ports at the other end of the rotor, and connects a second set of ports to the exhaust system (*g*). Fluid can flow into and out of the rotor channels through these ports. The exhaust gas inlet port is made small enough to cause a significant pressure rise in the exhaust manifold (e.g., 2 atm) when the engine is operated at its rated power. The pressure wave process does not depend on the pressure and flow fluctuations within the manifold caused by individual cylinder exhaust events: its operation can be explained assuming constant pressure at each set of ports. As the rotor makes one revolution, the ends of each channel are alternatively closed, or are open to a flow passage. By appropriate arrangement of these passages and selection of the geometry and location of the ports, an efficient energy transfer between the engine exhaust gases and the fresh charge can be realized.[46]

The wave-compression process in the Comprex can be explained in more detail with the aid of Fig. 6-59, where the rotational motion of the channels has been unrolled. Consider the channel starting at the top; it is closed at both ends and contains air at atmospheric pressure. As it opens at the upper edge of the high-pressure gas (G-HP) duct, a compression or shock wave (1) propagates from

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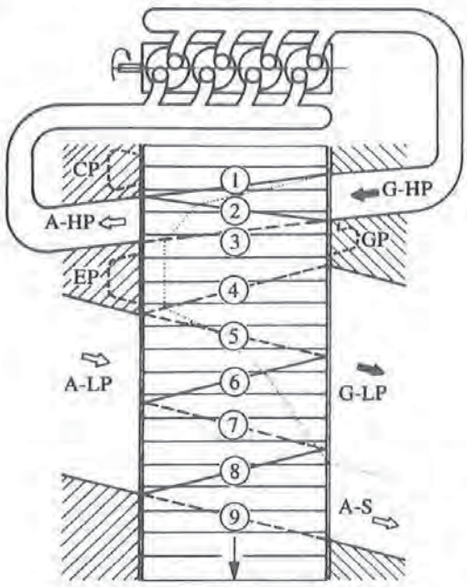


FIGURE 6-59
Unrolled view of the Comprex pressure-wave process.[47] A Air, G Gas, S Scavenging, HP High pressure, LP Low pressure; CP, EP, GP are pockets.

the right end of the channel toward the left, compressing the air through which it passes. The compressed air behind the wave occupies less space so the high-pressure exhaust gas moves into the channel as indicated by the dotted line. This line is the boundary between the two fluids. As this wave (1) reaches the left end, the channel is opened and compressed air flows into the engine inlet duct (A-HP). The inlet duct is shaped to provide the same mass flow at lower velocity: this deceleration of the air produces a second compression wave (2) which propagates back into the channel. As a result the compressed air leaving the cell on the left has a higher pressure than the driving gas on the right. As this wave (2) arrives at the right-hand side, the high-pressure gas (G-HP) channel closes. An expansion wave (3) then propagates back to the left, separating the now motionless and partly expanded fluid on the right from still-moving fluid on the left. When this wave (3) reaches the left-hand end, A-HP is closed and all the gases in the channel are at rest. Note that the first gas particles (dotted line) have not quite reached the air end of the channel: a cushion of air remains to prevent break-through.

The cell's contents are still at a higher pressure than the low pressure in the exhaust gas duct. When the right-hand end of the cell reaches this duct, the cell's contents expand into the exhaust. This motion is transferred through the channel by an expansion wave (4) which propagates to the left at sonic speed. When this wave reaches the left-hand end, the cell opens to the low-pressure air duct (A-LP) and fresh air is drawn into the cell. The flow to the right continues, but with decreasing speed due to wave action (5, 6, 7, 8) and pressure losses at each end of the cell. When the dotted line—the interface between air and the exhaust gas—reaches the right end of the cell, all the driving gas has left. The cell is then

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purged by the scavenging air flow (A-S) and filled with fresh air at atmospheric pressure. At wave (9), the cell is closed at both ends, restoring it to its initial state.[47]

The speed of these pressure waves is the local sound speed and is a function of local gas temperature only. Thus, the above process will only work properly for a given exhaust gas temperature at a particular cell speed. The operating range is extended by the use of "pockets" as shown in Fig. 6-59. The pockets prevent the reflection of sound waves from a closed channel end which would cause a substantial change in flow velocity in the channel. These pockets, marked CP and EP on the air side and GP on the exhaust gas side, allow flow from one channel to adjacent channels via the pocket if the wave action requires it. Thus the device can be tuned for full-load medium-speed operation and still give acceptable performance at other loads and speeds because the pockets allow the particle paths to change without major losses.[46]

Figure 6-60 shows the apparent compressor performance map of a Comprex when connected to a small three-cylinder diesel engine. Note that the map depends on the engine to which the device is coupled because the exhaust gas expansion process and fresh air compression process occur within the same rotor. The volume flow rate is the net air: it is the total air flow into the device less the scavenging air flow. The values of isentropic efficiency [defined by Eq. (6.39)] are comparable to those of mechanical and aerodynamic compressors.

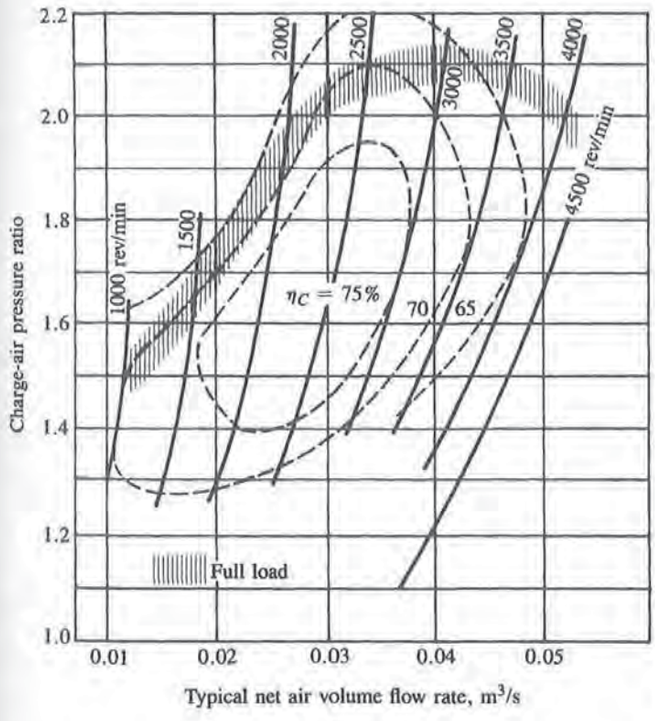


FIGURE 6-60
Apparent compressor map of Comprex connected to a 1.2-dm³ diesel engine: charge-air pressure ratio plotted versus net air volume flow rate (total air flow less scavenging air flow).[46]

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PROBLEMS

6.1. A conventional spark-ignition engine operating with gasoline will not run smoothly (due to incomplete combustion) with an equivalence ratio leaner than about $\phi = 0.8$. It is desirable to extend the smooth operating limit of the engine to leaner equivalence ratios so that at part-throttle operation (with intake pressure less than 1 atmosphere) the pumping work is reduced. Leaner than normal operation can be achieved by adding hydrogen gas (H_2) to the mixture in the intake system. The addition of H_2 makes the fuel-air mixture easier to burn.

(*a*) The fuel composition with "mixed" fuel operation is $H_2 + C_8H_{18}$—one mole of hydrogen to every mole of gasoline, which is assumed the same as isooctane. What is the stoichiometric air/fuel ratio for the "mixed" fuel?

(*b*) The lower heating value of H_2 is 120 MJ/kg and for isooctane is 44.4 MJ/kg. What is the heating value per kilogram of fuel mixture?

(*c*) Engine operation with isooctane and the mixed ($H_2 + C_8H_{18}$) fuel is compared in a particular engine at a part-load condition (brake mean effective pressure of 275 kPa and 1400 rev/min). You are given the following information about the engine operation:

Fuel	C_8H_{18}	$H_2 + C_8H_{18}$
Equivalence ratio	0.8	0.5
Gross indicated fuel conversion efficiency	0.35	0.4
Mechanical rubbing friction mep	138 kPa	138 kPa
Inlet manifold pressure	46 kPa	?
Pumping mep	55 kPa	?

Estimate approximately the inlet manifold pressure and the pumping mean effective pressure with ($H_2 + C_8H_{18}$) fuel. Explain your method and assumptions clearly. Note that mechanical efficiency η_m is defined as

$$\eta_m = \frac{\text{bmep}}{\text{imep}_g} = \frac{\text{bmep}}{\text{bmep} + \text{rfmep} + \text{pmep}}$$

6.2. Hydrogen is a possible future fuel for spark-ignition engines. The lower heating value of hydrogen is 120 MJ/kg and for gasoline (C_8H_{14}) is 44 MJ/kg. The stoichiometric air/fuel ratio for hydrogen is 34.3 and for gasoline is 14.4. A disadvantage of hydrogen fuel in the SI engine is that the partial pressure of hydrogen in the H_2-air mixture reduces the engine's volumetric efficiency, which is proportional to the partial pressure of air. Find the partial pressure of air in the intake manifold downstream of the hydrogen fuel-injection location at wide-open throttle when the total intake manifold pressure is 1 atmosphere; the equivalence ratio is 1.0. Then estimate the ratio of the fuel energy per unit time entering a hydrogen-fueled engine operating with a stoichiometric mixture to the fuel energy per unit time entering an identical gasoline-fueled engine operating at the same speed with a stoichiometric mixture. (Note that the "fuel energy" per unit mass of *fuel* is the fuel's heating value.)

6.3. Sketch (*a*) shows an ideal cycle p-V diagram for a conventional throttled spark-ignition engine, 1-2-3-4-5-6-7-1. The gas properties c_v, c_p, γ, R throughout the cycle are constant. The mass of gas in the cylinder is m. The exhaust pressure is p_e.

Sketch (*b*) shows an ideal cycle p-V diagram 1-2-3-4-5-6-8-1 for a spark-ignition engine with novel inlet valve timing. The inlet manifold is unthrottled; it has essentially the same pressure as the exhaust. To reduce the mass inducted at part

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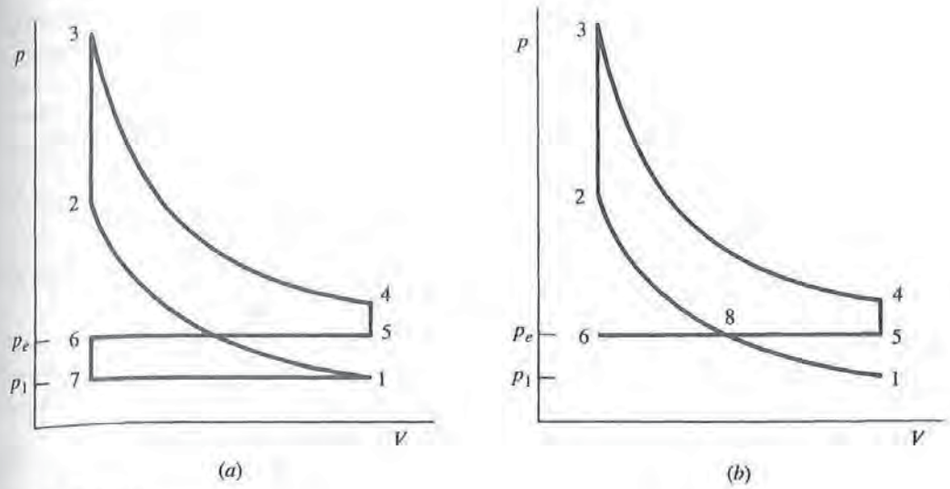


FIGURE P6-3

load, the inlet valve is closed rapidly *partway through the intake stroke* at point 8. The gas in the cylinder at inlet valve closing at 8 is then expanded isentropically to 1 with the inlet valve closed. The pressure p_1 at the start of compression is the same for both cycles.

(*a*) Indicate on p-V diagrams the area that corresponds to the pumping work per cycle for cycles (*a*) and (*b*). Which area is greater?

(*b*) Derive expressions for the pumping work per cycle W_p in terms of m, c_v, γ, T_1, (p_e/p_1), and the compression ratio r_c for cycles (*a*) and (*b*). Be consistent about the signs of the work transfers to and from the gas.

(*c*) For $\gamma = 1.3$, $r_c = 8$, and $(p_e/p_1) = 2$ find the ratio $W_p(b)/W_p(a)$, assuming the values of T_1 and m are the same in both cases.

6.4. For four-stroke cycle engines, the inlet and exhaust valve opening and closing crank angles are typically: IVO 15° BTC; IVC 50° ABC; EVO 55° BBC; EVC 10° ATC. Explain why these valve timings improve engine breathing relative to valve opening and closing at the beginnings and ends of the intake and exhaust strokes. Are there additional design issues that are important?

6.5. Estimate approximately the pressure drop across the inlet valve about halfway through the intake stroke and across the exhaust valve halfway through the exhaust stroke, when the piston speed is at its maximum for a typical four-stroke cycle spark-ignition engine with $B = L = 85$ mm at 2500 and 5000 rev/min at WOT. Assume appropriate values for any valve and port geometric details required, and for the gas composition and state.

6.6. Using the data in Fig. 6-21, estimate the fraction of the original mass left in the cylinder: (*a*) at the end of the blowdown process and (*b*) at the end of the exhaust stroke.

6.7. Compare the engine residual gas fraction data in Fig. 6-19 with ideal cycle estimates of residual gas fraction as follows. Using Eq. (5.47) plot the fuel-air cycle residual mass fraction x_r against p_i/p_e for $r_c = 8.5$ on the same graph as the engine data in Fig. 6-19 at 1400 rev/min and 27° valve overlap. Assume $T_r = 1400$ K and $(\gamma - 1)/\gamma = 0.24$ in Eq. (5.47). Suggest an explanation for any significant difference.

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6.8. One concept that would increase SI engine efficiency is early intake valve closing (EIVC) where the intake valve closes *before* the piston reaches BC on the intake stroke, thus limiting the amount of charge inducted into the cylinder.
(*a*) Explain why EIVC improves engine efficiency at part load. (*Hint:* consider what must happen to the inlet manifold pressure in order to maintain constant mass in the cylinder as the intake valve is closed sooner.)
(*b*) This part load reduction in charge could be achieved by using late intake valve closing where the intake valve is not closed until the compression stroke has pushed some of the cylinder gases back out into the intake manifold. Based on a comparison of p-V diagrams, is this method inferior to EIVC?

6.9. An eight-cylinder turbocharged aftercooled four-stroke cycle diesel engine operates with an inlet pressure of 1.8 atmospheres at its maximum rated power at 2000 rev/min. $B = 128$ mm, $L = 140$ mm, η_v (based on inlet manifold conditions of 1.8 atm and 325 K after the aftercooler) = 0.9. The compressor isentropic efficiency is 0.7.
(*a*) Calculate the power required to drive the turbocharger compressor.
(*b*) If the exhaust gas temperature is 650°C and the turbocharger isentropic efficiency is 0.65, estimate the pressure at turbine inlet. The turbine exhausts to the atmosphere.

6.10. The charging efficiency of two-stroke cycle diesel engines can be estimated from measurement of the concentration of O_2 and CO_2 in the burned gases within the cylinder, or in the exhaust blowdown pulse prior to any mixing with fresh air. The engine bore = 125 mm, stroke = 150 mm, compression ratio = 15. The fuel flow rate at 1800 rev/min is 1.6 g/s per cylinder. The conditions used to evaluate the air density for the reference mass are 300 K and 1 atm. The molar concentrations (dry) of CO_2 and O_2 in the in-cylinder burned gases are 7.2 and 10.4 percent (see Fig. 4-22). The scavenging air flow rate is 80 g/s. Evaluate (*a*) the charging efficiency, (*b*) the delivery ratio, and (*c*) the trapping efficiency (assuming the trapped mass equals the reference mass).

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CHAPTER 7

SI ENGINE FUEL METERING AND MANIFOLD PHENOMENA

7.1 SPARK-IGNITION ENGINE MIXTURE REQUIREMENTS

The task of the engine induction and fuel systems is to prepare from ambient air and fuel in the tank an air-fuel mixture that satisfies the requirements of the engine over its entire operating regime. In principle, the optimum air/fuel ratio for a spark-ignition engine is that which gives the required power output with the lowest fuel consumption, consistent with smooth and reliable operation. In practice, the constraints of emissions control may dictate a different air/fuel ratio, and may also require the recycling of a fraction of the exhaust gases (EGR) into the intake system. The relative proportions of fuel and air that provide the lowest fuel consumption, smooth reliable operation, and satisfy the emissions requirements, at the required power level, depend on engine speed and load. Mixture requirements and preparation are usually discussed in terms of the air/fuel ratio or fuel/air ratio (see Sec. 2.9) and percent EGR [see Eq. (4.2)]. While the fuel metering system is designed to provide the appropriate fuel flow for the *actual* air flow at each speed and load, the relative proportions of fuel and air can be stated more generally in terms of the fuel/air equivalence ratio ϕ, which is the actual fuel/air ratio normalized by dividing by the stoichiometric fuel/air ratio [Eq.

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(3.8)]. The combustion characteristics of fuel-air mixtures and the properties of combustion products, which govern engine performance, efficiency, and emissions, correlate best for a wide range of fuels relative to the stoichiometric mixture proportions. Where appropriate, therefore, the equivalence ratio will be used as the defining parameter. A typical value for the stoichiometric air/fuel ratio of gasoline is 14.6.† Thus, for gasoline,

$$\phi \approx \frac{14.6}{A/F} \tag{7.1}$$

The effects of equivalence ratio variations on engine combustion, emissions, and performance are discussed more fully in Chaps. 9, 11, and 15. A brief summary is sufficient here. Mixture requirements are different for full-load (wide-open throttle) and for part-load operation. At the former operating condition, complete utilization of the *inducted air* to obtain maximum power for a given displaced volume is the critical issue. Where less than the maximum power at a given speed is required, efficient utilization of the *fuel* is the critical issue. At wide-open throttle, maximum power for a given volumetric efficiency is obtained with rich-of-stoichiometric mixtures, $\phi \approx 1.1$ (see the discussion of the fuel-air cycle results in Sec. 5.5.3). Mixtures that are richer still are sometimes used to increase volumetric efficiency by increasing the amount of charge cooling that accompanies fuel vaporization [see Eq. (6.5)], thereby increasing the inducted air density.

At part-load (or part-throttle) operating conditions, it is advantageous to dilute the fuel-air mixture, either with excess air or with recycled exhaust gas. This dilution improves the fuel conversion efficiency for three reasons:[1] (1) the expansion stroke work for a given expansion ratio is increased as a result of the change in thermodynamic properties of the burned gases—see Secs. 5.5.3 and 5.7.4; (2) for a given mean effective pressure, the intake pressure increases with increasing dilution, so pumping work decreases—see Fig. 5-10; (3) the heat losses to the walls are reduced because the burned gas temperatures are lower. In the absence of strict engine NO_x emission requirements, excess air is the obvious diluent, and at part throttle engines have traditionally operated lean. When tight control of NO_x, HC, and CO emissions is required, operation of the engine with a stoichiometric mixture is advantageous so that a three-way catalyst‡ can be used to clean up the exhaust. The appropriate diluent is then recycled exhaust gases which significantly reduces NO_x emissions from the engine itself. The amount of diluent that the engine will tolerate at any given speed and load depends on the details of the engine's combustion process. Increasing excess air

† Typical value only. Most gasolines have $(A/F)_s$ in the range 14.4 to 14.7. $(A/F)_s$ could lie between 14.1 and 15.2.

‡ A three-way catalyst system, when operated with a close-to-stoichiometric mixture, achieves substantial reductions in NO_x, CO, and HC emissions simultaneously; see Sec. 11.6.2.

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or the amount of recycled exhaust slows down the combustion process and increases its variability from cycle to cycle. A certain minimum combustion repeatability or stability level is required to maintain smooth engine operation. Deterioration in combustion stability therefore limits the amount of dilution an engine can tolerate. As load decreases, less dilution of the *fresh* mixture can be tolerated because the internal dilution of the mixture with residual gas increases (see Sec. 6.4). At idle conditions, the fresh mixture will not usually tolerate any EGR and may need to be stoichiometric or fuel-rich to obtain adequate combustion stability.

Mixture composition requirements over the engine load and speed range are illustrated schematically for the two approaches outlined above in Fig. 7-1. If stoichiometric operation and EGR are not required for emissions control, as load increases the mixture is leaned out from a fuel-rich or close-to-stoichiometric composition at very light load. As wide-open throttle operation is approached at each engine speed, the mixture is steadily enriched to rich-of-stoichiometric at the maximum bmep point. With the stoichiometric operating conditions required for three-way-catalyst-equipped engines, when EGR is used, the percentage of recycled exhaust increases from zero at light load to a maximum at mid-load, and then decreases to zero as wide-open throttle conditions are approached so maximum bmep can be obtained. Combinations of these strategies are possible. For example, lean operation at light load can be used for best efficiency, and

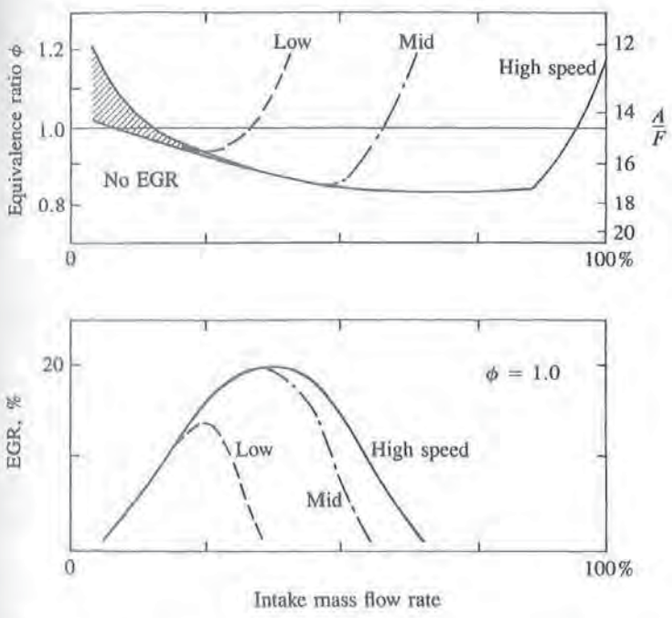


FIGURE 7-1
Typical mixture requirements for two common operating strategies. Top diagram shows equivalence ratio variation with intake mass flow rate (percent of maximum flow at rated speed) at constant low, mid, and high engine speeds. Bottom diagram shows recycled exhaust (EGR) schedule as a function of intake flow rate, for low, mid, and high speeds for stoichiometric operation.

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stoichiometric mixtures (with a three-way catalyst) and/or EGR can be used at mid loads to control NO_x emissions.

In practical spark-ignition engine induction systems, the fuel and air distribution between engine cylinders is not uniform (and also varies in each individual cylinder on a cycle-by-cycle basis). A spread of one or more air/fuel ratios between the leanest and richest cylinders over the engine's load and speed range is not uncommon in engines with conventional carburetors. The average mixture composition must be chosen to avoid excessive combustion variability in the leanest operating cylinder. Thus, as the spread in mixture nonuniformity increases, the mean equivalence ratio must be moved toward stoichiometric and away from the equivalence ratio which gives minimum fuel consumption.

7.2 CARBURETORS

7.2.1 Carburetor Fundamentals

A carburetor has been the most common device used to control the fuel flow into the intake manifold and distribute the fuel across the air stream. In a carburetor, the air flows through a converging-diverging nozzle called a venturi. The pressure difference set up between the carburetor inlet and the throat of the nozzle (which depends on the air flow rate) is used to meter the appropriate fuel flow for that air flow. The fuel enters the air stream through the fuel discharge tube or ports in the carburetor body and is atomized and convected by the air stream past the throttle plate and into the intake manifold. Fuel evaporation starts within the carburetor and continues in the manifold as fuel droplets move with the air flow and as liquid fuel flows over the throttle and along the manifold walls. A modern carburetor which meters the appropriate fuel flow into the air stream over the complete engine operating range is a highly developed and complex device. There are many types of carburetors; they share the same basic concepts which we will now examine.

Figure 7-2 shows the essential components of an elementary carburetor. The air enters the intake section of the carburetor (1) from the air cleaner which removes suspended dust particles. The air then flows into the carburetor venturi (a converging-diverging nozzle) (2) where the air velocity increases and the pressure decreases. The fuel level is maintained at a constant height in the float chamber (3) which is connected via an air duct (4) to the carburetor intake section (1). The fuel flows through the main jet (a calibrated orifice) (5) as a result of the pressure difference between the float chamber and the venturi throat and through the fuel discharge nozzle (6) into the venturi throat where the air stream atomizes the liquid fuel. The fuel-air mixture flows through the diverging section of the venturi where the flow decelerates and some pressure recovery occurs. The flow then passes the throttle valve (7) and enters the intake manifold.

Note that the flow may be unsteady even when engine load and speed are constant, due to the periodic filling of each of the engine cylinders which draws air through the carburetor venturi. The induction time, 1/(2N) (20 ms at 1500

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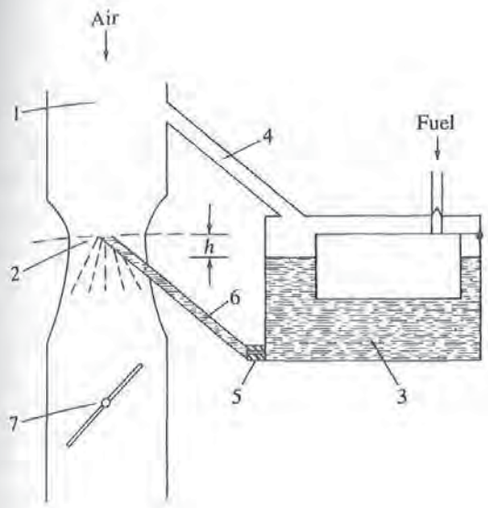


FIGURE 7-2
Schematic of elementary carburetor.
1 Inlet section
2 Venturi throat
3 Float chamber
4 Pressure equalizing passage
5 Calibrated orifice
6 Fuel discharge tube
7 Throttle plate

rev/min), is the characteristic time of this periodic cylinder filling process. Generally, the characteristic times of changes in throttle setting are longer; it takes several engine operating cycles to reestablish steady-state engine operation after a sudden change in throttle position.[2] It is usually assumed that the flow processes in the carburetor can be modeled as quasi steady.

FLOW THROUGH THE VENTURI. Equation (C.8) in App. C relates the mass flow rate of a gas through a flow restriction to the upstream stagnation pressure and temperature, and the pressure at the throat. For the carburetor venturi:

$$\dot{m}_a = \frac{C_{D_T} A_T p_0}{\sqrt{RT_0}} \left(\frac{p_T}{p_0}\right)^{1/\gamma} \left\{ \frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{p_T}{p_0}\right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2} \tag{7.2}$$

where C_{D_T} and A_T are the discharge coefficient and area of the venturi throat, respectively. If we assume the velocity at the carburetor inlet can be neglected, Eq. (7.2) can be rearranged in terms of the pressure drop from upstream conditions to the venturi throat for the air stream, $\Delta p_a = p_0 - p_T$, as

$$\dot{m}_a = C_{D_T} A_T (2\rho_{a_0} \Delta p_a)^{1/2} \Phi \tag{7.3}$$

where

$$\Phi = \left[\left(\frac{\gamma}{\gamma - 1} \right) \frac{(p_T/p_0)^{2/\gamma} - (p_T/p_0)^{(\gamma+1)/\gamma}}{1 - (p_T/p_0)} \right]^{1/2} \tag{7.4}$$

and accounts for the effects of compressibility. Figure C-3 shows the value of Φ as a function of pressure drop. For the normal carburetor operating range, where $\Delta p_a/p_0 \leq 0.1$, the effects of compressibility which reduce Φ below 1.0 are small.

FLOW THROUGH THE FUEL ORIFICE. Since the fuel is a liquid and therefore essentially incompressible, the fuel flow rate through the main fuel jet is given by

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Eq. (C.2) in App. C as

$$\dot{m}_f = C_{D_o} A_o (2\rho_f \Delta p_f)^{1/2} \tag{7.5}$$

where C_{D_o} and A_o are the discharge coefficient and area of the orifice, respectively, Δp_f is the pressure difference across the orifice, and the orifice area is assumed much less than the passage area. Usually, the fuel level in the float chamber is held below the fuel discharge nozzle, as shown in Fig. 7-2, to prevent fuel spillage when the engine is inclined to the horizontal (e.g., in a vehicle on a slope). Thus,

$$\Delta p_f = \Delta p_a - \rho_f g h$$

where h is typically of order 10 mm.

The discharge coefficient C_{D_o} in Eq. (7.5) represents the effect of all deviations from the ideal one-dimensional isentropic flow. It is influenced by many factors of which the most important are the following: (1) fluid mass flow rate; (2) orifice length/diameter ratio; (3) orifice/approach-area ratio; (4) orifice surface area; (5) orifice surface roughness; (6) orifice inlet and exit chamfers; (7) fluid specific gravity; (8) fluid viscosity; and (9) fluid surface tension. The use of the orifice Reynolds number, $\mathrm{Re}_o = \rho V D_o/\mu$, as a correlating parameter for the discharge coefficient accounts for effects of mass flow rate, fluid density and viscosity, and length scale to a good first approximation. The discharge coefficient of a typical carburetor main fuel-metering system orifice increases smoothly with increasing Re_o.[3]

CARBURETOR PERFORMANCE. The air/fuel ratio delivered by this carburetor is given by

$$\left(\frac{A}{F}\right) = \frac{\dot{m}_a}{\dot{m}_f} = \left(\frac{C_{D_T}}{C_{D_o}}\right)\left(\frac{A_T}{A_o}\right)\left(\frac{\rho_{a_0}}{\rho_f}\right)^{1/2}\left(\frac{\Delta p_a}{\Delta p_a - \rho_f g h}\right)^{1/2} \Phi \tag{7.6}$$

and the equivalence ratio ϕ $[=(A/F)_s/(A/F)]$ by

$$\phi = \frac{(A/F)_s}{\Phi}\left(\frac{C_{D_o}}{C_{D_T}}\right)\left(\frac{A_o}{A_T}\right)\left(\frac{\rho_f}{\rho_{a_0}}\right)^{1/2}\left(1 - \frac{\rho_f g h}{\Delta p_a}\right)^{1/2} \tag{7.7}$$

where $(A/F)_s$ is the stoichiometric air/fuel ratio. The terms A_o, A_T, ρ_f, and ρ_{a_0} are all constant for a given carburetor, fuel, and ambient conditions. Also, except for very low flows, $\rho_f g h \ll \Delta p_a$. The discharge coefficients, C_{D_o} and C_{D_T}, and Φ vary with flow rates, however. Hence, the equivalence ratio of the mixture delivered by an elementary carburetor is not constant.

Figure 7-3 illustrates the performance of the elementary carburetor. The top set of curves shows how Φ, C_{D_T}, and C_{D_o} typically vary with the venturi pressure drop.[4] Note that for $\Delta p_a \leq \rho_f g h$ there is no fuel flow. Once fuel starts to flow, as a consequence of these variations the fuel flow rate increases more rapidly than the air flow rate. The carburetor delivers a mixture of increasing fuel/air equivalence ratio as the flow rate increases. Suppose the venturi and orifice are sized to

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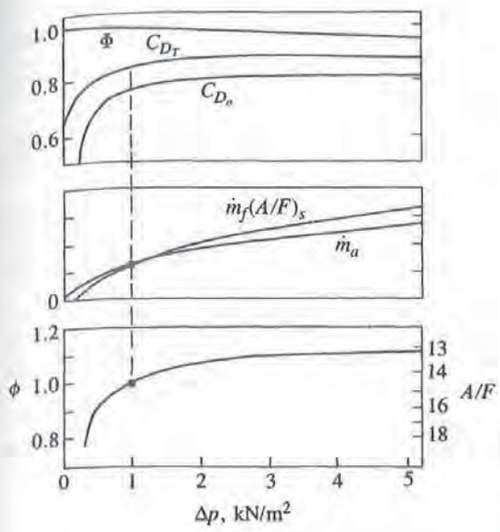


FIGURE 7-3
Performance of elementary carburetor: variation of C_{D_T}, C_{D_o}, Φ, $\dot{m}_f\,(A/F)_s$, $\dot{m}_a$, and equivalence ratio ϕ with venturi pressure drop.

give a stoichiometric mixture at an air flow rate corresponding to 1 kN/m^2 venturi pressure drop (middle graph in Fig. 7-3). At higher air flow rates, the carburetor will deliver a fuel-rich mixture; at very high flow rates it will eventually deliver an essentially constant equivalence ratio. At lower air flow rates, the mixture delivered leans out rapidly. Thus, the elementary carburetor cannot provide the variation in mixture ratio which the engine requires over the complete load range at any given speed (see Fig. 7-1).

The deficiencies of the elementary carburetor can be summarized as follows:

1. At low loads the mixture becomes leaner; the engine requires the mixture to be enriched at low loads.
2. At intermediate loads, the mixture equivalence ratio increases slightly as the air flow increases. The engine requires an almost constant equivalence ratio.
3. As the air flow approaches the maximum wide-open throttle value, the equivalence ratio remains essentially constant. However, the mixture equivalence ratio should increase to 1.1 or greater to provide maximum engine power.
4. The elementary carburetor cannot compensate for transient phenomena in the intake manifold. Nor can it enrich the mixture during engine starting and warm-up.
5. The elementary carburetor cannot adjust to changes in ambient air density (due primarily to changes in altitude).

7.2.2 Modern Carburetor Design

The changes required in the elementary carburetor so that it provides the equivalence ratio versus air flow distribution shown in Fig. 7-1 are:

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1. The *main metering system* must be compensated to provide essentially constant lean or stoichiometric mixtures over the 20 to 80 percent air flow range.
2. An *idle system* must be added to meter the fuel flow at idle and light loads.
3. An *enrichment system* must be added so the engine can provide its maximum power as wide-open throttle is approached.
4. An *accelerator pump* which injects additional fuel when the throttle is opened rapidly is required to maintain constant the equivalence ratio delivered to the engine cylinder.
5. A *choke* must be added to enrich the mixture during engine starting and warm-up to ensure a combustible mixture within each cylinder at the time of ignition.
6. *Altitude compensation* is required to adjust the fuel flow to changes in air density.

In addition, it is necessary to increase the magnitude of the pressure drop available for controlling the fuel flow. Two common methods used to achieve this are the following.

BOOST VENTURIS. The carburetor venturi should give as large a vacuum at the throat as possible at maximum air flow, within the constraints of a low pressure loss across the complete venturi and diffuser. In a single venturi, as the diameter of the throat is decreased at a given air flow to increase the flow velocity and hence the metering signal at the throat, the pressure loss increases. A higher vacuum signal at the venturi throat and higher velocities for improved atomization can be obtained without increasing the overall pressure loss through the use of multiple venturis. Figure 7-4 shows the geometry and the pressure distribution in a typical double-venturi system. A boost venturi is positioned upstream of the throat of the larger main venturi, with its discharge at the location of maximum velocity in the main venturi. Only a fraction of the air flows through the boost venturi. Since the pressure at the boost venturi *exit* equals the pressure

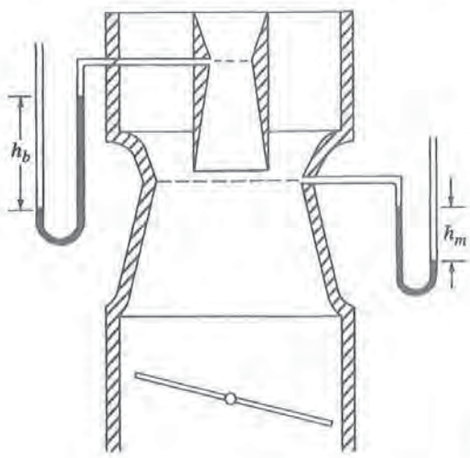


FIGURE 7-4
Schematic of carburetor double-venturi system.

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at the main venturi *throat*, a higher vacuum $\Delta p_b = \rho_m g h_b$ is obtained at the boost venturi throat which can be used to obtain more precise metering of the fuel (ρ_m is the manometer fluid density). Best results are obtained with the boost venturi exit slightly upstream (≈ 5 mm) of the main venturi throat. Because only a fraction of the total air flow goes through the boost venturi, the use of multiple venturis makes it possible to obtain a high velocity air stream (up to 200 m/s) where the fuel is introduced at the boost venturi throat, and adequate vacuum, and to reduce the pressure loss across the total venturi system, without increasing the height of the carburetor. The fuel is better atomized in the smaller boost venturi with its higher air velocity, and since this air and fuel mixture is discharged centrally into the surrounding venturi, a more homogeneous mixture results. The vacuum developed at the venturi throat of a typical double-venturi system is about twice the theoretical vacuum of a single venturi of the same flow area.[5] A triple-venturi system can be used to give further increases in metering signal. The overall discharge coefficient of a multiple-venturi carburetor is lower than a single-venturi carburetor of equal cross-sectional area. The throat area of the main venturi in a multiple-venturi system is usually increased, therefore, above the single-venturi size to compensate for this. Some decrease in air stream velocity is tolerated to maintain a high discharge coefficient (and hence a high volumetric efficiency).[6]

MULTIPLE-BARREL CARBURETORS. Use of carburetors with venturi systems in parallel is a common way of maintaining an adequate part-load metering signal, high volumetric efficiency at wide-open throttle, and minimum carburetor height as engine size and maximum air flow increases. As venturi size in a single-barrel carburetor is increased to provide a higher engine air flow at maximum power, the venturi length increases and the metering signal generated at low flows decreases. Maximum wide-open throttle air flow is some 30 to 70 times the idle air flow (the value depending on engine displacement). Two-barrel carburetors usually consist of two single-barrel carburetors mounted in parallel. Four-barrel carburetors consist of a pair of two-barrel carburetors in parallel, with throttle plates compounded on two shafts. Air flows through the primary barrel(s) at low and intermediate engine loads. At higher loads, the throttle plate(s) on the secondary barrel(s) (usually of larger cross-sectional area) start to open when the air flow exceeds about 50 percent of the maximum engine air flow.

There are many different designs of complete carburetors. The operating principles of the methods most commonly used to achieve the above listed modifications will now be reviewed. Figure 7-5 shows a schematic of a conventional modern carburetor and the names of the various components and fuel passages.

COMPENSATION OF MAIN METERING SYSTEM. Figure 7-6 shows a main fuel-metering system with air-bleed compensation. As the pressure at the venturi throat decreases, air is bled through an orifice (or series of orifices) into the main fuel well. This flow reduces the pressure difference across the main fuel-metering orifice which no longer experiences the full venturi vacuum. The mixing of bleed

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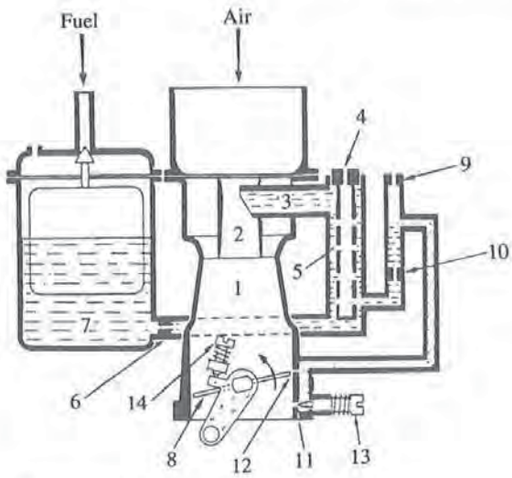


FIGURE 7-5
Schematic of modern carburetor.

1 Main venturi
2 Boost venturi
3 Main metering spray tube or nozzle
4 Air-bleed orifice
5 Emulsion tube or well
6 Main fuel-metering orifice
7 Float chamber
8 Throttle plate
9 Idle air-bleed orifice
10 Idle fuel orifice
11 Idle mixture orifice
12 Transition orifice
13 Idle mixture adjusting screw
14 Idle throttle setting adjusting screw

Fuel enters the air stream from the main metering system through (3). At idle, fuel enters air at (11). During transition, fuel enters at (11), (12), and (3). (*Courtesy S.p.A.E. Weber.*)

air with the fuel forms an emulsion which atomizes more uniformly to smaller drop sizes on injection at the venturi throat. The schematic in Fig. 7-6 illustrates the operating principle. When the engine is not running, the fuel is at the same level in the float bowl and in the main well. With the engine running, as the throttle plate is opened, the air flow and the vacuum in the venturi throat increase. For $\Delta p_v (= p_0 - p_v) < \rho_f g h_1$, there is no fuel flow from the main metering system. For $\rho_f g h_1 < \Delta p_v < \rho_f g h_2$, only fuel flows through the main well and nozzle, and the system operates just like an elementary carburetor. For $\Delta p_v > \rho_f g h_2$, air enters the main well together with fuel. The amount of air entering the well is controlled by the size of the main air-bleed orifice. The amount of air is

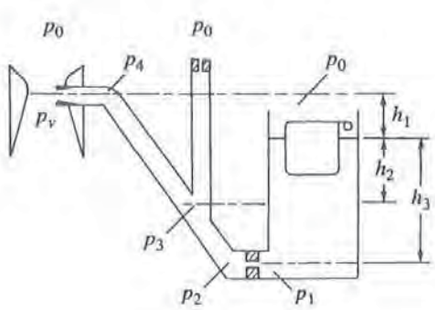


FIGURE 7-6
Schematic of main metering system with air-bleed compensation.

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small and does not significantly affect the composition of the mixture. The air-bleed mass flow rate is given by

$$\dot{m}_{a_b} = C_{D_b} A_b [2(p_0 - p_3)\rho_a]^{1/2} \tag{7.8}$$

where C_{D_b} and A_b are the discharge coefficient and the area of the air-bleed orifice. The fuel mass flow rate through the fuel orifice is given by

$$\dot{m}_f = C_{D_o} A_o [2(p_1 - p_2)\rho_f]^{1/2} \tag{7.9}$$

where

$$p_1 = p_0 + \rho_f g h_3 \qquad \text{and} \qquad p_2 = p_3 + \rho_f g(h_3 - h_2)$$

The density of the emulsion ρ_{em} in the main well and nozzle is usually approximated by

$$\rho_{em} = \frac{\dot{m}_{a_b} + \dot{m}_f}{\dot{m}_{a_b}/\rho_a + \dot{m}_f/\rho_f} \tag{7.10}$$

Since typical values are $\rho_f = 730$ kg/m^3 and $\rho_a = 1.14$ kg/m^3, usually $\rho_f \gg \rho_{em} \gg \rho_a$. Thus, as the air-bleed flow rate increases, the height of the column of emulsion becomes less significant. However, the decrease in emulsion density due to increasing air bleed increases the flow velocity, which results in a significant pressure drop across the main nozzle. This pressure drop depends on nozzle length and diameter, fuel flow rate, bleed air flow rate, relative velocity between fuel and bleed air, and fuel properties. It is determined empirically, and has been found to correlate with ρ_{em} [as defined by Eq. (7.10)].[2, 6] The pressure loss at the main discharge nozzle with two-phase flow can be several times the pressure loss with single-phase flow.

Figure 7-7 illustrates the behavior of the system shown in Fig. 7-6: $\dot{m}_a$, $\dot{m}_f$, and the fuel/air equivalence ratio ϕ are plotted against Δp_v. Once the bleed system is operating ($\Delta p_v > \rho_f g h_2$) the fuel flow rate is reduced below its equivalent elementary carburetor value (the $A_b = 0$ line). As the bleed orifice area is increased, in the limit of large A_b and neglecting the pressure losses in the main nozzle, the fuel flow rate remains constant ($A_b \to \infty$). An appropriate choice of bleed orifice area A_b will provide the desired equivalence ratio versus pressure drop or air flow characteristic.

Additional control flexibility is obtained in practice through use of a second orifice, or of a series of holes in the main well or emulsion tube as shown in Fig. 7-5. Main metering systems with controlled air bleed provide reliable and stable control of mixture composition at part throttle engine operation. They are simple, have considerable design flexibility, and atomize the fuel effectively. In some carburetor designs, an additional compensation system consisting of a tapered rod or needle in the main metering orifice is used. The effective open area of the main metering orifice, and hence the fuel flow rate, can thus be directly related to throttle position (and manifold vacuum).

A wide range of two-phase flow patterns can be generated by bleeding an air flow into a liquid flow. Fundamental studies of the generation and flow of

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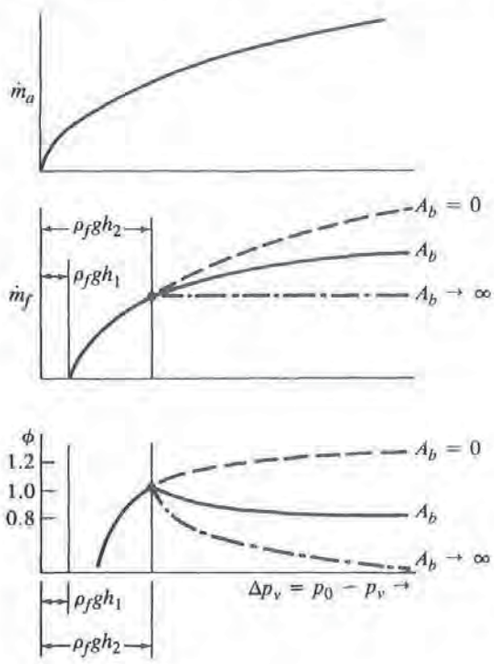


FIGURE 7-7
Metering characteristics of system with air-bleed compensation: mass flow rate of air $\dot{m}_a$, mass flow rate of fuel $\dot{m}_f$, and equivalence ratio ϕ as functions of venturi pressure drop for different air-bleed orifice areas A_b.

two-phase mixtures in small diameter tubes with bleed holes similar to those used in carburetors have been carried out.[7] For a given pipe and bleed hole size, the type of flow pattern set up depends on the flow rates of the two phases.

IDLE SYSTEM. The idle system is required because at low air flows through the carburetor insufficient vacuum is created at the venturi throat to draw fuel into the air stream. However, at idle and light loads, the manifold vacuum is high, with the pressure drop occurring across the almost-closed throttle plate. This low manifold pressure at idle is exploited for the idle fuel system by connecting the main fuel well to an orifice in the throttle body downstream of the throttle plate. Figure 7-5 shows the essential features of an idle system. The main well (5) is connected through one or more orifices (10), past one or more idle air-bleed orifices (or holes) (9), past an idle mixture adjusting screw (13), to the idle discharge port (11) in the throttle body. Emulsifying air is admitted into the idle system [at (9) and (12)] to reduce the pressure drop across the idle port and permit larger-sized ports (which are easier to manufacture) to be used. Satisfactory engine operation at idle is obtained empirically by means of the idle throttle position stop adjustment (14) and the idle mixture adjustment (13). As the throttle is opened from its idle position, the idle metering system performs a transitional function. One or more holes (12) located above the idle discharge port (11) assist as air bleeds when the throttle is at or near its idle position. As the throttle plate opens and the air flow increases, these additional discharge holes are exposed to the manifold vacuum. Additional fuel is forced out of these holes into the air stream to provide the appropriate mixture ratio. As the throttle plate is opened further, the main fuel metering system starts to supply fuel also. Because the two

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systems are coupled, they interact and the main system behavior in this *transition region* is modified by the fuel flow through the idle system. The total combined fuel flow provides a rich (or close-to-stoichiometric) mixture at idle, a progressive leaning of the mixture as air flow increases, and eventually (as the main system takes over full control of the fuel flow rate) an approximately constant mixture composition.

POWER ENRICHMENT SYSTEM. This system delivers additional fuel to enrich the mixture as wide-open throttle is approached so the engine can deliver its maximum power. The additional fuel is normally introduced via a submerged valve which communicates directly with the main discharge nozzle, bypassing the metering orifice. The valve, which is spring loaded, is operated either mechanically through a linkage with the throttle plate (opening as the throttle approaches its wide-open position) or pneumatically (using manifold vacuum).

ACCELERATOR PUMP. When the throttle plate is opened rapidly, the fuel-air mixture flowing into the engine cylinder leans out temporarily. The primary reason for this is the time lag between fuel flow into the air stream at the carburetor and the fuel flow past the inlet valve (see Sec. 7.6.3). While much of the fuel flow into the cylinder is fuel vapor or small fuel droplets carried by the air stream, a fraction of the fuel flows onto the manifold and port walls and forms a liquid film. The fuel which impacts on the walls evaporates more slowly than fuel carried by the air stream and introduces a lag between the air/fuel ratio produced at the carburetor and the air/fuel ratio delivered to the cylinder. An accelerator pump is used as the throttle plate is opened rapidly to supply additional fuel into the air stream at the carburetor to compensate for this leaning effect. Typically, fuel is supplied to the accelerator pump chamber from the float chamber via a small hole in the bottom of the fuel bowl, past a check valve. A pump diaphragm and stem is actuated by a rod attached to the throttle plate lever. When the throttle is opened to increase air flow, the rod-driven diaphragm will increase the fuel pressure which shuts the valve and discharges fuel past a discharge check valve or weight in the discharge passage, through the accelerator pump discharge nozzle(s), and into the air stream. A calibrated orifice controls the fuel flow. A spring connects the rod and diaphragm to extend the fuel discharge over the appropriate time period and to reduce the mechanical strain on the linkage.

CHOKE. When a cold engine is started, especially at low ambient temperatures, the following factors introduce additional special requirements for the complete carburetor:

1. Because the starter-cranked engine turns slowly (70 to 150 rev/min) the intake manifold vacuum developed during engine start-up is low.
2. This low manifold vacuum draws a lower-than-normal fuel flow from the carburetor idle system.

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3. Because of the low manifold temperature and vacuum, fuel evaporation in the carburetor, manifold, and inlet port is much reduced.

Thus, during cranking, the mixture which reaches the engine cylinder would be too lean to ignite. Until normal manifold conditions are established, fuel distribution is also impaired. To overcome these deficiencies and ensure prompt starts and smooth operation during engine warm-up, the carburetor must supply a fuel-rich mixture. This is obtained with a choke. Once normal manifold conditions are established, the choke must be excluded. The primary element of a typical choke system is a plate, upstream of the carburetor, which can close off the intake system. At engine start-up, the choke plate is closed to restrict the air flow into the carburetor barrel. This causes almost full manifold vacuum within the venturi which draws a large fuel flow through the main orifice. When the engine starts, the choke is partly opened to admit the necessary air flow and reduce the vacuum in the venturi to avoid flooding the intake with fuel. As the engine warms up, the choke is opened either manually or automatically with a thermostatic control. For normal engine operation the choke plate is fully open and does not influence carburetor performance. A manifold vacuum control is often used to close the choke plate partially if the engine is accelerated during warm-up. During engine warm-up the idle speed is increased to prevent engine stalling. A fast idle cam is rotated into position by the automatic choke lever.

ALTITUDE COMPENSATION. An inherent characteristic of the conventional float type carburetor is that it meters fuel flow in proportion to the air *volume* flow rate. Air density changes with ambient pressure and temperature, with changes due to changes in pressure with altitude being most significant. For example, at 1500 m above sea level, mean atmospheric pressure is 634 mmHg or 83.4 percent of the mean sea-level value. While ambient temperature variations, winter to summer, can produce changes of comparable magnitude, the temperature of the air entering the carburetor for warmed-up engine operation is controlled to within much closer tolerances by drawing an appropriate fraction of the air from around the exhaust manifold.

Equation (7.6) shows how the air/fuel ratio delivered by the main metering system will vary with inlet air conditions. The primary dependence is through the $\sqrt{\rho_{a_0}}$ term; depending on what is held constant (e.g., throttle setting or air mass flow rate) there may be an additional, much smaller dependence through Φ and Δp_a (see Ref. 5). To a good approximation, the enrichment E with increasing altitude z is given by

$$1 + E = \frac{(F/A)_z}{(F/A)_0} = \left(\frac{\rho_{a_0}}{\rho_{a_z}}\right)^{1/2} \tag{7.11}$$

For $z = 1500$ m, $E = 9.5$ percent; thus, a cruise equivalence ratio of 0.9 or $(A/F) = 16.2$ would be enriched to close to stoichiometric.

The effects of increase in altitude on the carburetor flow curve shown in Fig. 7-1 are: (1) to enrich the entire part-throttle portion of the curve and (2) to

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bring in the power-enrichment system at a lower air flow rate due to decreased manifold vacuum. To reduce the impact to changes in altitude on engine emissions of CO and HC, modern carburetors are altitude compensated. A number of methods can be used to compensate for changes in ambient pressure with altitude:

1. *Venturi bypass method.* To keep the air volume flow rate through the venturi equal to what it was at sea-level atmospheric pressure (calibration condition), a bypass circuit around the venturi for the additional volume flow is provided.
2. *Auxiliary jet method.* An auxiliary fuel metering orifice with a pressure-controlled tapered metering rod connects the fuel bowl to the main well in parallel with the main metering orifice.
3. *Fuel bowl back-suction method.* As altitude increases, an aneroid bellows moves a tapered rod from an orifice near the venturi throat, admitting to the bowl an increasing amount of the vacuum signal developed at the throat.
4. *Compensated air-bleed method.* The orifices in the bleed circuits to each carburetor system are fitted with tapered metering pins actuated by a single aneroid bellows.[8]

TRANSIENT EFFECTS. The pulsating and transient nature of the flow through a carburetor during actual engine operation is illustrated by the data shown in Fig. 7-8.[2] The changes in pressure with time in the intake manifold and at the boost venturi throat of a standard two-barrel carburetor installed on a production V-8 engine are shown as the throttle is opened from light load (22°) to wide-open throttle at 1000 rev/min. Note the rapid increase in boost venturi suction as the throttle is suddenly opened. This results from the sudden large increase in the air flow rate and corresponding increase in air velocity within the boost venturi. Note also that the pressure fluctuations decay rapidly, and within a few engine revolutions have stabilized at the periodic values associated with the new throttle angle. At wide-open throttle, the pulsating nature of the flow as each

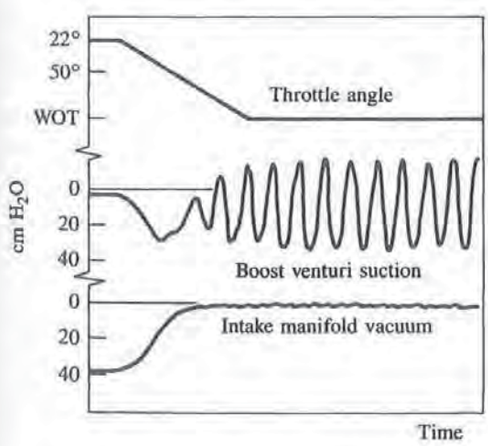


FIGURE 7-8
Throttle angle, boost venturi suction, and intake manifold vacuum variation with time as throttle is opened from light load (22°) to wide-open throttle at 1000 rev/min. Standard two-barrel carburetor and production V-8 engine.[2]

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cylinder draws in its charge is evident. The pressure drop across the main metering jet also fluctuates. The pulsations in the venturi air flow (and hence fuel flow) due to the filling of each cylinder in turn are negligibly small at small throttle angles and increase to a maximum at wide-open throttle. At small throttle openings, the choked flow at the throttle plate prevents the manifold pressure fluctuations from propagating upstream into the venturi. The effective time-averaged boost venturi suction is greater for the pulsating flow case than for the steady flow case. If the ratio of the effective metering signal for a pulse cycle to that for steady air flow at the same average mass flow is denoted as $1 + \Omega$, where Ω is the pulsation factor, then Ω is related to the amplitude and frequency of pressure waves within the intake manifold as well as the damping effect of the throttle plate. An empirical equation for Ω is

$$\Omega = \frac{\text{constant} \times (1 - M)p_m n_R}{N n_{c/b}} \tag{7.12}$$

where M is the throttle plate Mach number, p_m the manifold pressure, n_R the number of revolutions per power stroke, N the crank speed, and $n_{c/b}$ the number of cylinders per barrel. The value of the constant depends on carburetor and engine geometry. For p_m in kilonewtons per square meter and N in revolutions per minute a typical value for the constant is 7.3.[2] Thus, at wide-open throttle at 1500 rev/min, Ω has a value of about 0.2. The transient behavior of the air and fuel flows in the manifold are discussed more fully in Sec. 7.6.

7.3 FUEL-INJECTION SYSTEMS

7.3.1 Multipoint Port Injection

The fuel-injection systems for conventional spark-ignition engines inject the fuel into the engine intake system. This section reviews systems where the fuel is injected into the intake port of each engine cylinder. Thus these systems require one injector per cylinder (plus, in some systems, one or more injectors to supplement the fuel flow during starting and warm-up). There are both mechanical injection systems and electronically controlled injection systems. The advantages of port fuel injection are increased power and torque through improved volumetric efficiency and more uniform fuel distribution, more rapid engine response to changes in throttle position, and more precise control of the equivalence ratio during cold-start and engine warm-up. Fuel injection allows the amount of fuel injected per cycle, for each cylinder, to be varied in response to inputs derived from sensors which define actual engine operating conditions. Two basic approaches have been developed; the major difference between the two is the method used to determine the air flow rate.

Figure 7-9 shows a schematic of a *speed-density* system, where engine speed and manifold pressure and air temperature are used to calculate the engine air flow. The electrically driven fuel pump delivers the fuel through a filter to the fuel line. A pressure regulator maintains the pressure in the line at a fixed value (e.g.,

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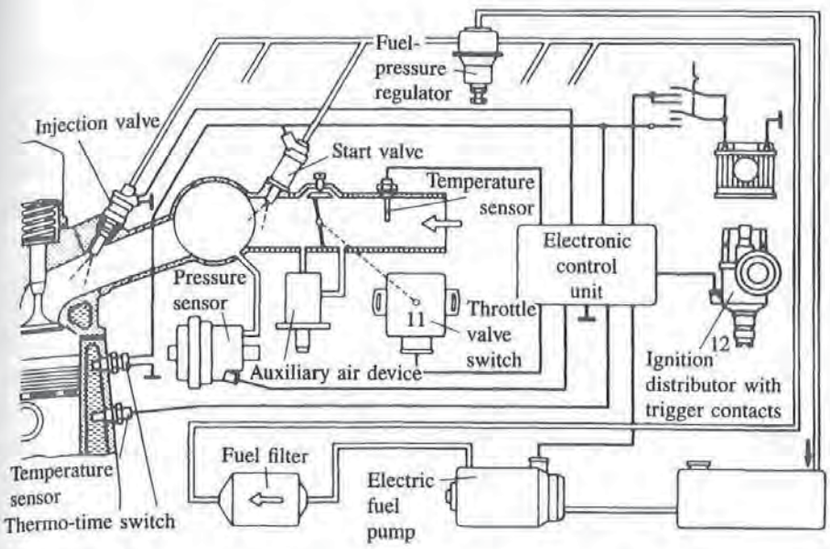


FIGURE 7-9
Speed-density electronic multipoint port fuel-injection system: Bosch D-Jetronic System.[9] (*Courtesy Robert Bosch GmbH and SAE.*)

270 kN/m^2, 39 lb/in^2, usually relative to manifold pressure to maintain a constant fuel pressure drop across the injectors). Branch lines lead to each injector; the excess fuel returns to the tank via a second line. The inducted air flows through the air filter, past the throttle plate to the intake manifold. Separate runners and branches lead to each inlet port and engine cylinder. An electromagnetically actuated fuel-injection valve (see Fig. 7-10) is located either in the intake manifold tube or the intake port of each cylinder. The major components of the injector are the valve housing, the injector spindle, the magnetic plunger to which the spindle is connected, the helical spring, and the solenoid coil. When the solenoid is not excited, the solenoid plunger of the magnetic circuit is forced, with its seal, against the valve seat by the helical spring and closes the fuel passage. When the solenoid coil is excited, the plunger is attracted and lifts the spindle about

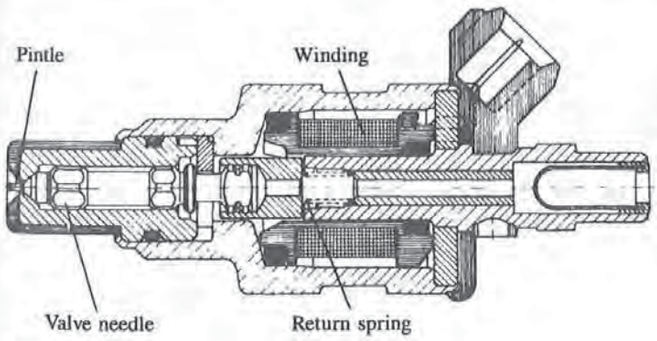


FIGURE 7-10
Cross section of fuel injector.[10]

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0.15 mm so that fuel can flow through the calibrated annular passage around the valve stem. The front end of the injector spindle is shaped as an atomizing pintle with a ground top to atomize the injected fuel. The relatively narrow spray cone of the injector, shown in the photo in Fig. 7-11, minimizes the intake manifold wall wetting with liquid fuel. The mass of fuel injected per injection is controlled by varying the duration of the current pulse that excites the solenoid coil. Typical injection times for automobile applications range from about 1.5 to 10 ms.[11]

The appropriate coil excitation pulse duration or width is set by the electronic control unit (ECU). In the speed-density system, the primary inputs to the ECU are the outputs from the manifold pressure sensor, the engine speed sensor (usually integral with the distributor), and the temperature sensors installed in the intake manifold to monitor air temperature and engine block to monitor the water-jacket temperature—the latter being used to indicate fuel-enrichment requirements during cold-start and warm-up. For warm-engine operation, the mass of air per cylinder per cycle m_a is given by

$$m_a = \eta_v(N)\rho_a(T_i, p_i)V_d = \frac{\eta_v V_d p_i}{R_a T_i} \tag{7.13}$$

where η_v is the volumetric efficiency, N is engine speed, ρ_a is the inlet air density, and V_d is the displaced volume per cylinder. The electronic control unit forms the pulse which excites the injector solenoids. The pulse width depends primarily on the manifold pressure; it also depends on the variation in volumetric efficiency η_v with speed N and variations in air density due to variations in air temperature. The control unit also initiates mixture enrichment during cold-engine operation and during accelerations that are detected by the throttle sensor.

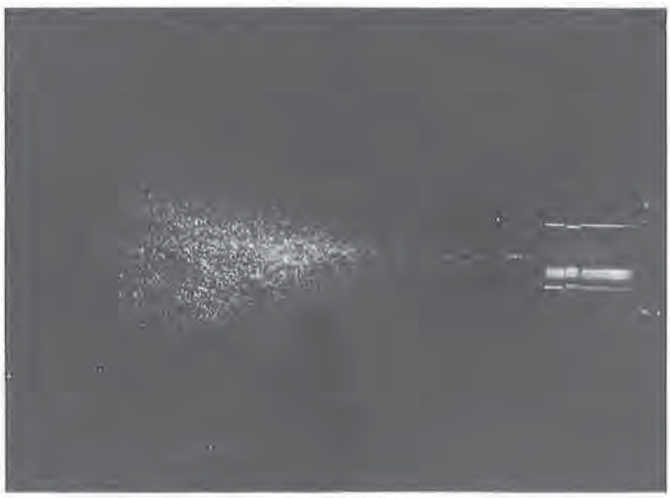


FIGURE 7-11
Short time-exposure photograph of liquid fuel spray from Bosch-type injector. (*Courtesy Robert Bosch GmbH.*)

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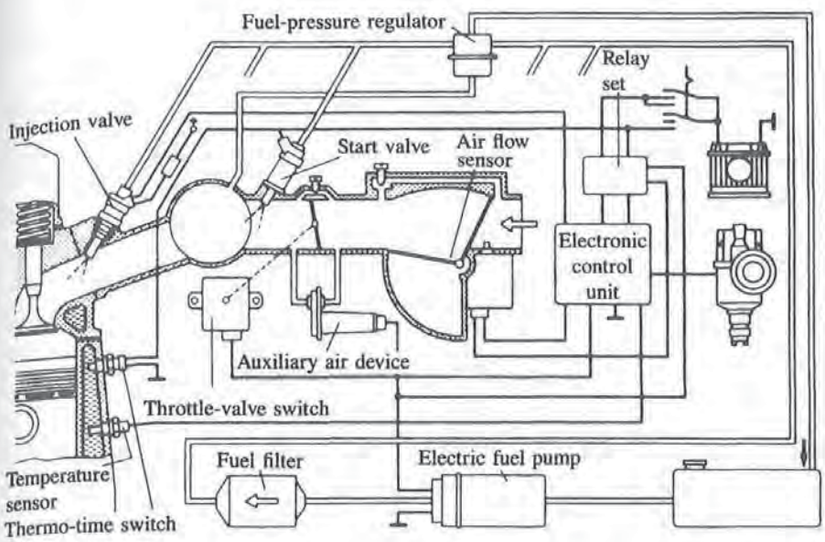


FIGURE 7-12
Electronic multipoint port fuel-injection system with air-flow meter: Bosch L-Jetronic system.[9] (*Courtesy Robert Bosch GmbH and SAE.*)

Figure 7-12 shows an alternative EFI system (the Bosch L-Jetronic) which uses an air-flow meter to measure air flow directly. The air-flow meter is placed upstream of the throttle. The meter shown measures the force exerted on a plate as it is displaced by the flowing air stream; it provides a voltage proportional to the air flow rate. An alternative air-flow measuring approach is to use a hot-wire air mass flow meter.[10] The advantages of direct air-flow measurement are:[12] (1) automatic compensation for tolerances, combustion chamber deposit buildup, wear and changes in valve adjustments; (2) the dependence of volumetric efficiency on speed and exhaust backpressure is automatically accounted for; (3) less acceleration enrichment is required because the air-flow signal precedes the filling of the cylinders; (4) improved idling stability; and (5) lack of sensitivity of the system to EGR since only the fresh air flow is measured.

The mass of air inducted per cycle to each cylinder, m_a, varies as

$$m_a \propto \frac{\dot{m}_a}{N} \tag{7.14}$$

Thus the primary signals for the electronic control unit are air flow and engine speed. The pulse width is inversely proportional to speed and directly proportional to air flow. The engine block temperature sensor, starter switch, and throttle valve switch provide input signals for the necessary adjustments for cold-start, warm-up, idling, and wide-open throttle enrichment.

For cold-start enrichment, one (or more) cold-start injector valve is used to inject additional fuel into the intake manifold (see Figs. 7-9 and 7-12). Since short opening and closing times are not important, this valve can be designed to

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provide extremely fine atomization of the fuel to minimize the enrichment required.

Mechanical, air-flow-based metering, continuous injection systems are also used. Figure 7-13 shows a schematic of the Bosch K-Jetronic system.[9,10] Air is drawn through the air filter, flows past the air-flow sensor, past the throttle valve, into the intake manifold, and into the individual cylinders. The fuel is sucked out of the tank by a roller-cell pump and fed through the fuel accumulator and filter to the fuel distributor. A primary pressure regulator in the fuel distributor maintains the fuel pressure constant. Excess fuel not required by the engine flows back to the tank. The mixture-control unit consists of the air-flow sensor and fuel distributor. It is the most important part of the system, and provides the desired metering of fuel to the individual cylinders by controlling the cross section of the metering slits in the fuel distributor. Downstream of each of these metering slits is a differential pressure valve which for different flow rates maintains the pressure drop at the slits constant.

Fuel-injection systems offer several options regarding the timing and location of each injection relative to the intake event.[10] The K-Jetronic mechanical injection system injects fuel continuously in front of the intake valves with the spray directed toward the valves. Thus about three-quarters of the fuel required for any engine cycle is stored temporarily in front of the intake valve, and one-quarter enters the cylinder directly during the intake process.

With electronically controlled injection systems, the fuel is injected intermittently toward the intake valves. The fuel-injection pulse width to provide the appropriate mass of fuel for each cylinder air charge varies from about 1.5 to 10 ms over the engine load and speed range. In crank angle degrees this varies

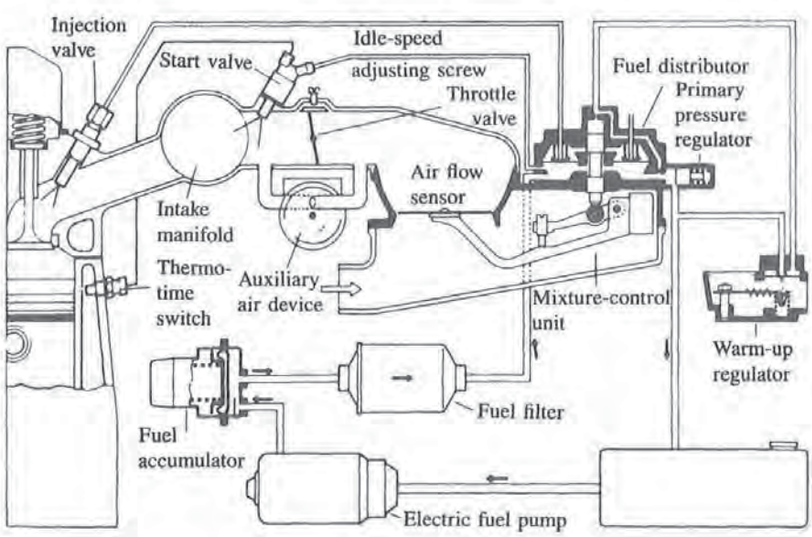


FIGURE 7-13
Mechanical multipoint port fuel-injection system: Bosch K-Jetronic system.[9] (*Courtesy Robert Bosch GmbH and SAE.*)

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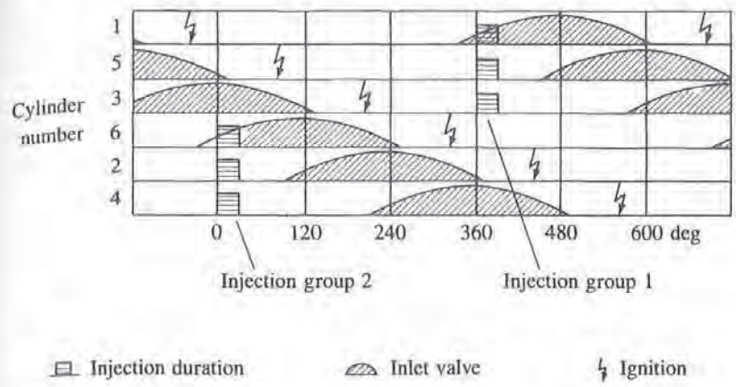


FIGURE 7-14
Injection pulse diagram for D-Jetronic system in six-cylinder engine.[10]

from about 10° at light load and low speed to about 300° at maximum speed and load. Thus the pulse width varies from being much less than to greater than the duration of the intake stroke. To reduce the complexity of the electronic control unit, groups of injectors are often operated simultaneously. In the Bosch L-Jetronic system, all injectors are operated simultaneously. To achieve adequate mixture uniformity, given the short pulse width relative to the intake process over much of the engine load-speed range, fuel is injected twice per cycle; each injection contributes half the fuel quantity required by each cylinder per cycle. (This approach is called simultaneous double-firing.) In the speed-density system, the injectors are usually divided into groups, each group being pulsed simultaneously. For example, for a six-cylinder engine, two groups of three injectors may be used. Injection for each group is timed to occur while the inlet valves are closed or just starting to open, as shown in Fig. 7-14. The other group of injectors inject one crank revolution later. Sequential injection timing, where the phasing of *each* injection pulse relative to its intake valve lift profile is the same, is another option. Engine performance and emissions do change as the timing of the start of injection relative to inlet valve opening is varied. Injection with valve lift at its maximum, or decreasing, is least desirable.[10]

7.3.2 Single-Point Throttle-Body Injection

Single-point fuel-injection systems, where one or two electronically controlled injectors meter the fuel into the air flow directly above the throttle body, are also used. They provide straightforward electronic control of fuel metering at lower cost than multipoint port injection systems. However, as with carburetor systems, the problems associated with slower transport of fuel than the air from upstream of the throttle plate to the cylinder must now be overcome (see Sec. 7.6). Figure 7-15 shows a cutaway of one such system.[13] Two injectors, each in a separate air-flow passage with its own throttle plate, meter the fuel in response to calibrations of air flow based on intake manifold pressure, air temperature, and

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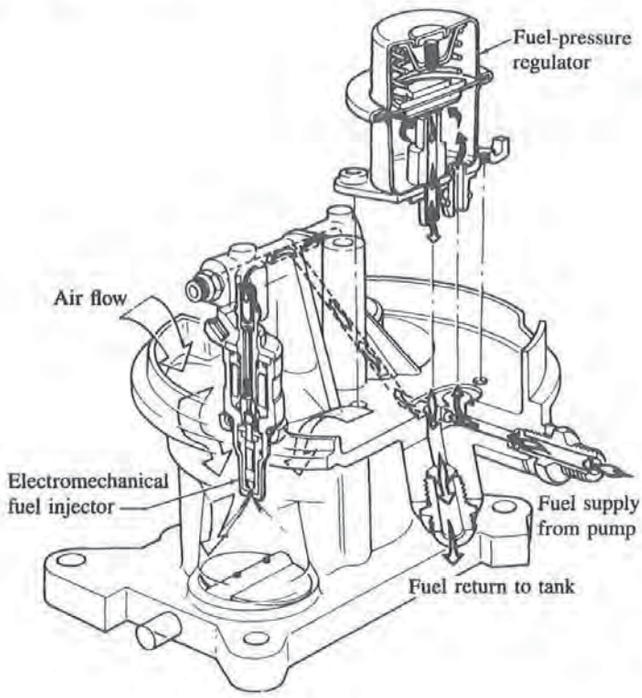


FIGURE 7-15
Cutaway drawing of a two-injector throttle-body electronic fuel-injection system.[13]

engine speed using the speed-density EFI logic described in the previous section. Injectors are fired alternatively or simultaneously, depending on load and speed and control logic used. Under alternative firing, each injection pulse corresponds to one cylinder filling. Smoothing of the fuel-injection pulses over time is achieved by proper placement of the fuel injector assembly above the throttle bore and plate. The walls and plate accumulate liquid fuel which flows in a sheet toward the annular throttle opening (see Sec. 7.6.3). The high air velocity created

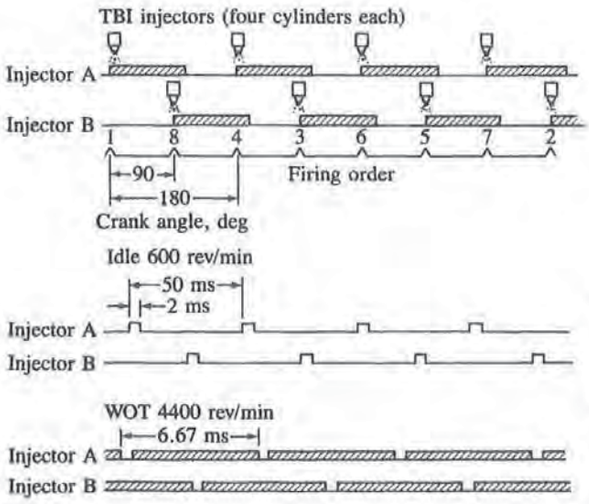


FIGURE 7-16
Injector fuel delivery schedule for two-injector throttle-body injection system for eight-cylinder engine with dual plane intake manifold. Each injection nozzle feeds one plane of the manifold and its four cylinders.[14]

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by the pressure drop across the throttle shears and atomizes the liquid sheet. Vigorous mixing of fuel and air then occurs, especially at part throttle, and provides good mixture uniformity and distribution between cylinders. Injector fuel delivery scheduling is illustrated in Fig. 7-16 for an eight-cylinder engine for a throttle-body fuel-injection system.[14]

7.4 FEEDBACK SYSTEMS

It is possible to reduce engine emissions of the three pollutants—unburned hydrocarbons, carbon monoxide, and oxides of nitrogen—with a single catalyst in the exhaust system if the engine is operated very close to the stoichiometric air/fuel ratio. Such systems (called three-way catalyst systems) are described in more detail in Sec. 11.6.2. The engine operating air/fuel ratio is maintained close to stoichiometric through the use of a sensor in the exhaust system, which provides a voltage signal dependent on the oxygen concentration in the exhaust gas stream. This signal is the input to a feedback system which controls the fuel feed to the intake.

The sensor [called an oxygen sensor or lambda sensor—λ being the symbol used for the relative air/fuel ratio, Eq. (3.9)] is an oxygen concentration cell with a solid electrolyte through which the current is carried by oxygen ions. The electrolyte is yttria (Y_2O_3)-stabilized zirconia (ZrO_2) ceramic which separates two gas chambers (the exhaust manifold and the atmosphere) which have different oxygen partial pressures. The cell can be represented as a series of interfaces as follows:

Exhaust	Metal	Ceramic	Metal	Air
p'_{O_2}	M_e	ZrO_2, Y_2O_3	M_e	p''_{O_2}

p''_{O_2} is the oxygen partial pressure of the air (≈ 20 kN/m^2) and p'_{O_2} is the equilibrium oxygen partial pressure in the exhaust gases. An electrochemical reaction takes place at the metal electrodes:

$$O_2 + 4M_e \rightleftharpoons 2O^{2-}$$

and the oxygen ions transport the current across the cell. The open-circuit output voltage of the cell V_o can be related to the oxygen partial pressures p'_{O_2} and p''_{O_2} through the Nernst equation:

$$V_o = \frac{RT}{4F} \ln \left(\frac{p''_{O_2}}{p'_{O_2}} \right) \tag{7.15}$$

where F is the Faraday constant. Equilibrium is established in the exhaust gases by the catalytic activity of the platinum metal electrodes. The oxygen partial pressure in equilibrated exhaust gases decreases by many orders of magnitude as the equivalence ratio changes from 0.99 to 1.01, as shown in Fig. 7-17*a*. Thus the sensor output voltage increases rapidly in this transition from a lean to a rich

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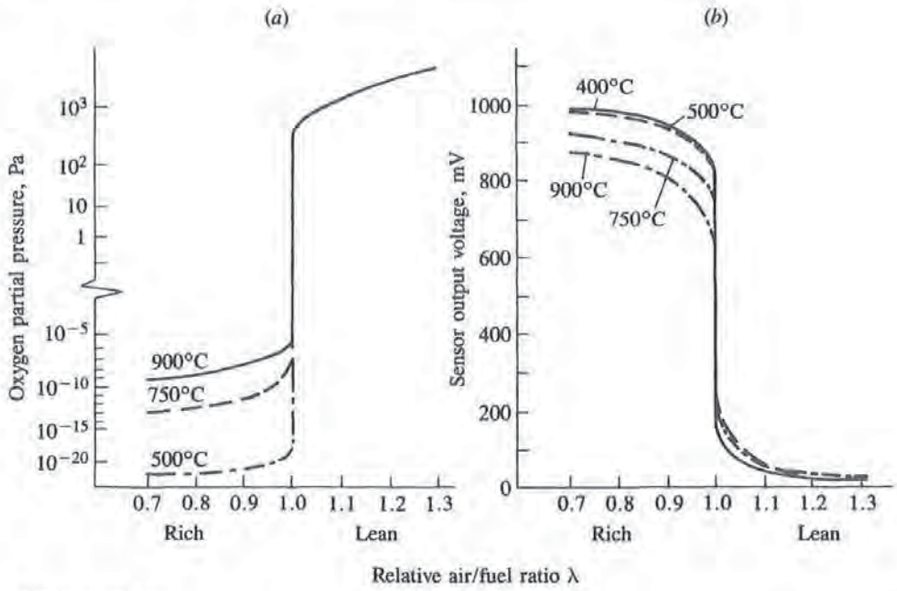


FIGURE 7-17
Oxygen-sensor characteristics. Variation as a function of relative air/fuel ratio and temperature of: (*a*) oxygen partial pressure in equilibrated combustion products; (*b*) sensor output voltage.[15]

mixture at the stoichiometric point, as shown in Fig. 7-17*b*. Since this transition is not temperature dependent, it is well suited as a sensor signal for a feedback system.[15]

Figure 7-18 shows a cross-section drawing of a lambda sensor, screwed into the wall of the exhaust manifold. This location provides rapid warm-up of the sensor following engine start-up. It also gives the shortest flow time from the fuel injector or carburetor location to the sensor—a delay time which is important in the operation of the feedback system. The sensor body is made of ZrO_2 ceramic stabilized with Y_2O_3 to give adequate low-temperature electrical conductivity. The inner and outer electrodes are 10-μm thick porous platinum layers to provide the required catalytic equilibration. The outer electrode which is exposed to the exhaust gases is protected against corrosion and erosion by a 100-μm spinal coat and a slotted shield. Air passes to the inner electrode through holes in the protective sleeve. The shield, protective sleeve, and housing are made from heat- and corrosion-resistant steel alloys. Such sensors were first developed for air/fuel ratio control at close to the stoichiometric value. Use of a similar sensor to control air/fuel ratios at lean-of-stoichiometric values during part-throttle engine operation is also feasible.

For closed-loop feedback control at close-to-stoichiometric, use is made of the sensor's low-voltage output for lean mixtures and a high-voltage output for rich mixtures. A control voltage reference level is chosen at about the mid-point of the steep transition in Fig. 7-17*b*. In the electronic control unit the sensor signal is compared to the reference voltage in the comparator as shown in Fig. 7-19*a*. The comparator output is then integrated in the integral controller whose

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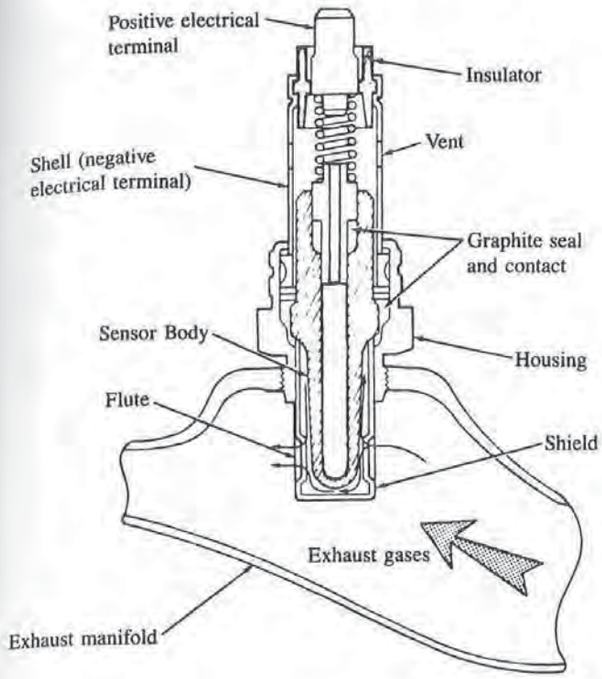


FIGURE 7-18
Cross-section drawing of exhaust oxygen sensor.[16]

output varies the fuel quantity linearly in the opposite direction to the sign of the comparator signal. There is a time lag τ_L in the loop composed of the transport time of fuel-air mixture from the point of fuel admission in the intake system to the sensor location in the exhaust, and the sensor and control system time delay. Because of this time lag, the controller continues to influence the fuel flow rate in the same direction, although the reference point $\phi = 1.0$ has been passed, as shown in Fig. 7-19*b*. Thus, oscillations in the equivalence ratio delivered to the engine exist even under steady-state conditions of closed-loop control. This behavior of the control system is called the *limit cycle*. The frequency f of this limit cycle is given by

$$f_{LC} = \frac{1}{4\tau_L} \tag{7.16}$$

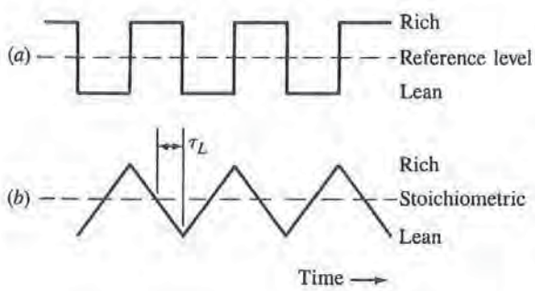


FIGURE 7-19
Operation of electronic control unit for closed-loop feedback: (*a*) sensor signal compared with reference level; (*b*) controller output voltage—the integrated comparator output.[12]

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and the change in equivalence ratio peak-to-peak is

$$\Delta\phi = 2K\tau_L \tag{7.17}$$

where K is the integrator gain (in equivalence ratio units per unit time).

Depending on the details of the three-way catalyst used for cleanup of all three pollutants (CO, HC, and NO_x) in the exhaust, the optimum average equivalence ratio may not be precisely the stoichiometric value. Furthermore, the reference voltage for maximum sensor durability may not correspond exactly to the stoichiometric point or the desired catalyst mean operating point. While a small shift ($\sim \pm 1$ percent) in operating point from the stoichiometric can be obtained by varying the reference voltage level, larger shifts are obtained by modifying the control loop to provide a steady-state bias. One method of providing a bias—asymmetrical gain rate biasing[17]—uses two separate integrator circuits with different gain rates K^+ and K^- to integrate the comparator output, depending on whether the comparator output is positive (rich) or negative (lean). An alternative biasing technique incorporates an additional delay time τ_D so that the controller output continues decreasing (or increasing) even though the sensor signal has switched from the high to the low level (or vice versa). By introducing this additional delay only on the negative slope of the sensor signal, a net lean bias is produced. Introducing the additional delay on the positive slope of the sensor signal produces a net rich bias.[12]

Note that the sensor only operates at elevated temperatures. During engine start-up and warm-up, the feedback system does not operate and conventional controls are required to obtain the appropriate fuel-air mixture for satisfactory engine operation.

7.5 FLOW PAST THROTTLE PLATE

Except at or close to wide-open throttle, the throttle provides the minimum flow area in the entire intake system. Under typical road-load conditions, more than 90 percent of the total pressure loss occurs across the throttle plate. The minimum-to-maximum flow area ratio is large—typically of order 100. Throttle

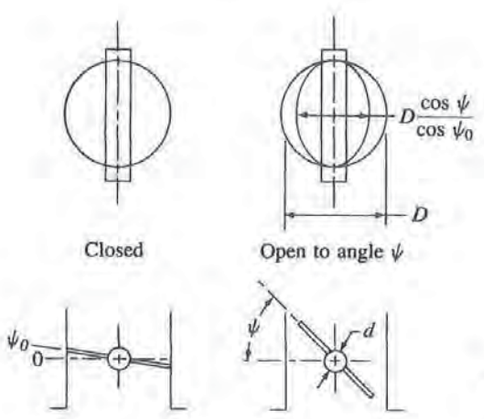
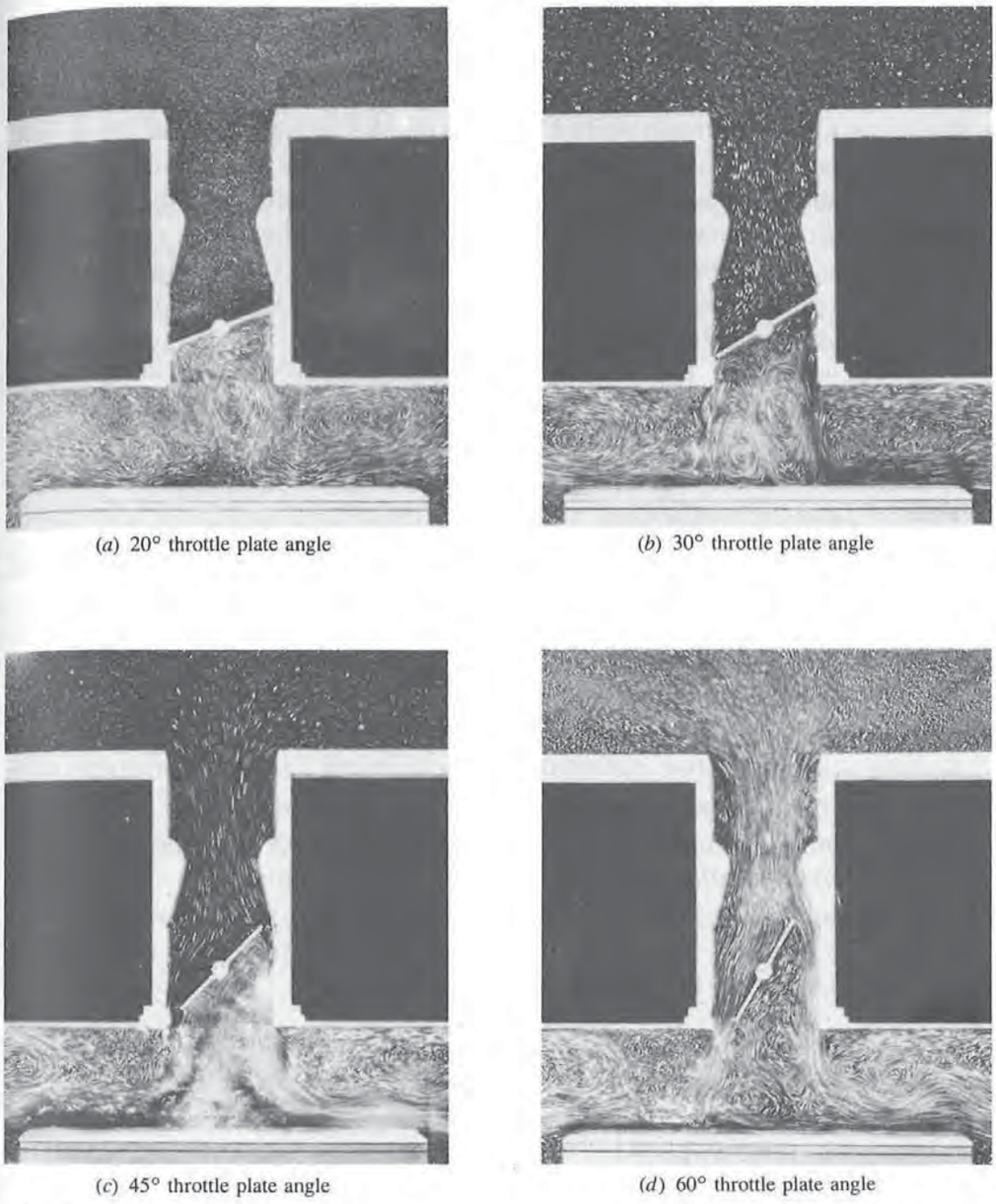


FIGURE 7-20
Throttle plate geometry.[2]

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(*a*) 20° throttle plate angle

(*b*) 30° throttle plate angle

(*c*) 45° throttle plate angle

(*d*) 60° throttle plate angle

FIGURE 7-21
Photographs of flow in two-dimensional hydraulic analog of carburetor venturi, throttle plate, and manifold plenum floor at different throttle plate angles.[18]

plate geometry and parameters are illustrated in Fig. 7-20. A throttle plate of conventional design such as Fig. 7-20 creates a three-dimensional flow field. At part-throttle operating conditions the throttle plate angle is in the 20 to 45° range and jets issue from the "crescent moon"-shaped open areas on either side of the throttle plate. The jets are primarily two dimensional. Figure 7-21 shows photographs taken of a two-dimensional hydraulic analog of a typical carburetor

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venturi and throttle plate in steady flow at different throttle angles. The path lines traced by the particles in the flow indicate the relative magnitude of the flow velocity.[18] The flow accelerates through the carburetor venturi (separation occurs at the corners of the entrance section); it then divides on either side of the throttle plate. There is a stagnation point on the upstream side of the throttle plate. The wake of the throttle plate contains two vortices which rotate in opposite directions. The jets on either side of the wake (at part throttle) are at or near sonic velocity. There is little or no mixing between the two jets. Thus, if maldistribution of the fuel-air mixture occurs above the throttle plate, it is not corrected immediately below the throttle plate.

In analyzing the flow through the throttle plate, the following factors should be considered:[2, 19, 20]

1. The throttle plate shaft is usually of sufficient size to affect the throttle open area.
2. To prevent binding in the throttle bore, the throttle plate is usually completely closed at some nonzero angle (5, 10, or 15°).
3. The discharge coefficient of the throttle plate is less than that of a smooth converging-diverging nozzle, and varies with throttle angle, pressure ratio, and throttle plate Reynolds number.
4. Due to the manufacturing tolerances involved, there is usually some minimum leakage area even when the throttle plate is closed against the throttle bore. This leakage area can be significant at small throttle openings.
5. The measured pressure drop across the throttle depends (± 10 percent) on the circumferential location of the downstream pressure tap.
6. The pressure loss across the throttle plate under the actual flow conditions (which are unsteady even when the engine speed and load are constant, see Fig. 7-8) may be less than under steady flow conditions.

The throttle plate open area A_{th}, as a function of angle ψ for the geometry in Fig. 7-20, is given by[2]

$$\frac{4A_{th}}{\pi D^2} = \left(1 - \frac{\cos\psi}{\cos\psi_0}\right) + \frac{2}{\pi}\left[\frac{a}{\cos\psi}(\cos^2\psi - a^2\cos^2\psi_0)^{1/2} - \frac{\cos\psi}{\cos\psi_0}\sin^{-1}\left(\frac{a\cos\psi_0}{\cos\psi}\right) - a(1-a^2)^{1/2} + \sin^{-1}a\right] \quad (7.18)$$

where $a = d/D$, d is the throttle shaft diameter, D is the throttle bore diameter, and ψ_0 is the throttle plate angle when tightly closed against the throttle bore. When $\psi = \cos^{-1}(a\cos\psi_0)$, the throttle open area reaches its maximum value ($\approx \pi D^2/4 - dD$). The throttle plate discharge coefficient (which varies with A_{th}), and minimum leakage area, must be determined experimentally.

The mass flow rate through the throttle valve can be calculated from standard orifice equations for compressible fluid flow [see App. C, Eqs. (C-8) and

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(C-9)]. For pressure ratios across the throttle less than the critical value ($p_T/p_0 = 0.528$), the mass flow rate is given by

$$\dot{m}_{th} = \frac{C_D A_{th} p_0}{\sqrt{RT_0}} \left(\frac{p_T}{p_0}\right)^{1/\gamma} \left\{\frac{2\gamma}{\gamma - 1}\left[1 - \left(\frac{p_T}{p_0}\right)^{(\gamma-1)/\gamma}\right]\right\}^{1/2} \tag{7.19}$$

where A_{th} is the throttle plate open area [Eq. (7.18)], p_0 and T_0 are the upstream pressure and temperature, p_T is the pressure downstream of the throttle plate (assumed equal to the pressure at the minimum area: i.e., no pressure recovery occurs), and C_D is the discharge coefficient (determined experimentally). For pressure ratios greater than the critical ratio, when the flow at the throttle plate is choked,

$$\dot{m}_{th} = \frac{C_D A_{th} p_0}{\sqrt{RT_0}} \gamma^{1/2} \left(\frac{2}{\gamma + 1}\right)^{(\gamma+1)/2(\gamma-1)} \tag{7.20}$$

The relation between air flow rate, throttle angle, intake manifold pressure, and engine speed for a two-barrel carburetor and a 4.7-dm^3 (288-in^3) displacement eight-cylinder production engine is shown in Fig. 7-22. While the lines are predictions from a quasi-steady computer simulation, the agreement with data is excellent. The figure shows that for an intake manifold pressure below the critical

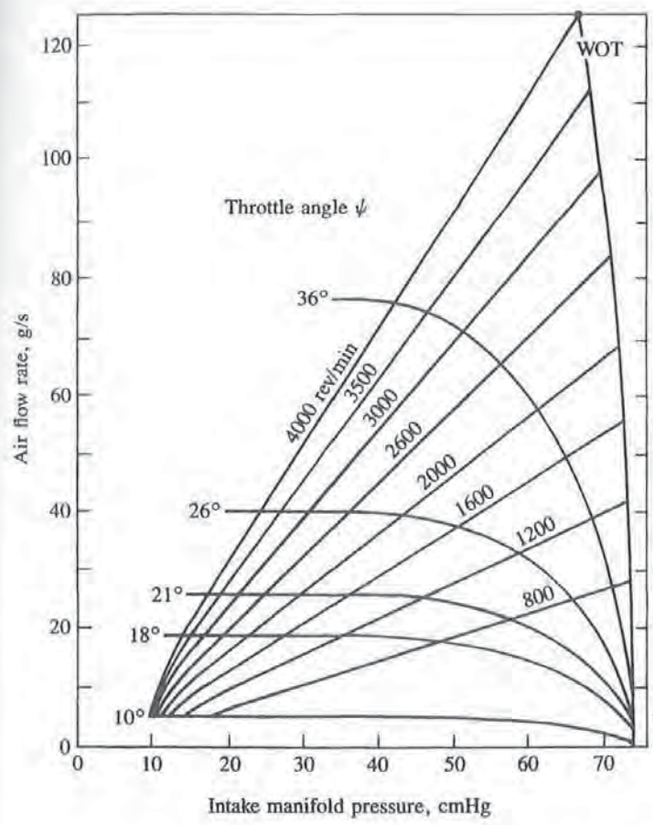


FIGURE 7-22
Variation in air flow rate past a throttle, with inlet manifold pressure, throttle angle, and engine speed. 4.7-dm^3 displacement eight-cylinder engine.[2]

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value ($0.528 \times p_{atm} = 53.5$ kN/m^2 = 40.1 cmHg) the air flow rate at a given throttle position is independent of manifold pressure and engine speed because the flow at the throttle plate is choked.[2]

7.6 FLOW IN INTAKE MANIFOLDS

7.6.1 Design Requirements

The details of the air and fuel flow in intake manifolds are extremely complex. The combination of pulsating flow into each cylinder, different geometry flow paths from the plenum beneath the throttle through each runner and branch of the manifold to each inlet port, liquid fuel atomization, vaporization and transport phenomena, and the mixing of EGR with the fresh mixture under steady-state engine operating conditions are difficult enough areas to untangle. During engine transients, when the throttle position is changed, the fact that the processes which govern the air and the fuel flow to the cylinder are substantially different introduces additional problems. This section reviews our current understanding of these phenomena.

Intake manifolds consist typically of a plenum, to the inlet of which bolts the throttle body, with individual runners feeding branches which lead to each cylinder (or the plenum can feed the branches directly). Important design criteria are: low air flow resistance; good distribution of air and fuel between cylinders; runner and branch lengths that take advantage of ram and tuning effects; sufficient (but not excessive) heating to ensure adequate fuel vaporization with carbureted or throttle-body injected engines. Some compromises are necessary; e.g., runner and branch sizes must be large enough to permit adequate flow without allowing the air velocity to become too low to transport the fuel droplets. Some of these design choices are illustrated in Fig. 7-23 which shows an inlet manifold and carburetor arrangement for a modern four-cylinder 1.8-dm^3 engine. In this design the four branches that link the plenum beneath the carburetor and throttle with the inlet ports are similar in length and geometry, to provide closely comparable flow paths. This manifold is heated by engine coolant as shown and uses an electrically heated grid beneath the carburetor to promote rapid fuel evaporation.[21] Exhaust gas heated stoves at the floor of the plenum are also used in some intake manifolds to achieve adequate fuel vaporization and distribution. Note that with EGR, the intake manifold may contain passages to bring the exhaust gas to the plenum or throttle body.

With port fuel-injection systems, the task of the inlet manifold is to control the air (and EGR) flow. Fuel does not have to be transported from the throttle body through the entire manifold. Larger and longer runners and branches, with larger angle bends, can be used to provide equal runner lengths and take greater advantage of ram and tuning effects. With port fuel injection it is not normally necessary to heat the manifold.

A large number of different manifold arrangements are used in practice. Different cylinder arrangements (e.g., four, V-six, in-line-six, etc.) provide quite different air and fuel distribution problems. Air-flow phenomena in manifolds can

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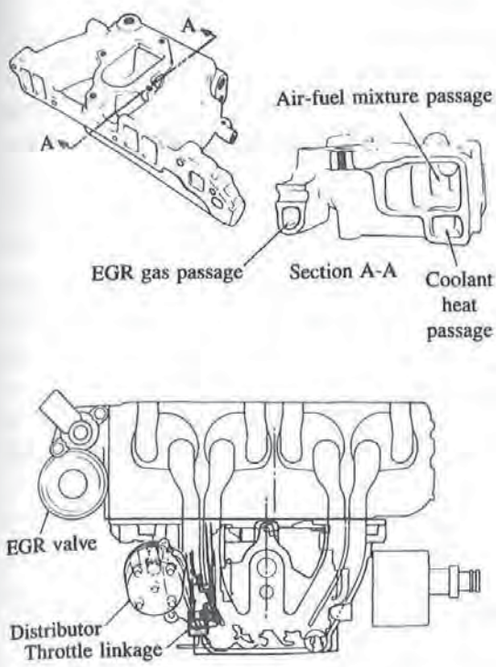


FIGURE 7-23
Inlet manifold for four-cylinder 1.8-dm^3 displacement spark-ignition engine.[21]

be considered as unaffected by the fuel flow. The reverse is definitely not the case: the fuel flow—liquid and vapor—depends strongly on the air flow. These two topics will therefore be reviewed in sequence.

7.6.2 Air-Flow Phenomena

The air flow out of the manifold occurs in a series of pulses, one going to each cylinder. Each pulse is approximately sinusoidal in shape. For four- and eight-cylinder engines, these flow pulses sequence such that the outflow is essentially zero between pulses. For six-cylinder arrangements the pulses will overlap. When the engine is throttled, backflow from the cylinder into the intake manifold occurs during the early part of the intake process until the cylinder pressure falls below the manifold pressure. Backflow can also occur early in the compression stroke before the inlet valve closes, due to rising cylinder pressure. The flow at the throttle will fluctuate as a consequence of the pulsed flow out of the manifold into the cylinders. At high intake vacuum, the flow will be continuously inward at the throttle and flow pulsations will be small. When the outflow to the cylinder which is undergoing its intake stroke is greater than the flow through the throttle, the cylinder will draw mixture from the rest of the intake manifold. During the portion of the intake stroke when the flow into the cylinder is lower than the flow through the throttle, mixture will flow back into the rest of the manifold. At wide-open throttle when the flow restriction at the throttle is a minimum, flow pulsations at the throttle location will be much more pronounced.[19]

The air flows to each cylinder of a multicylinder engine, even under steady operating conditions, are not identical. This is due to differences in runner and

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branch length and other geometric details of the flow path to each cylinder. Also, as each cylinder's intake flow commences, air is drawn from the branch and runner leading to the cylinder, the plenum, and the other runners and branches feeding the plenum, as well as past the throttle. This "drawing down" of other parts of the intake manifold depends on the arrangement of the plenum, runners, and branches, and the firing order of the cylinders. Thus the air flow to each individual cylinder is affected by the details of its own branch, how that branch connects to the rest of the intake manifold, and the cylinder firing order.[22] The differences between the air flows to individual cylinders have been measured. Variations in the average air mass flow rate to each individual cylinder of up to about 5 percent above and below the average are quite common. Larger peak-to-peak variations (± 15 percent) have been measured. The extent of each cylinder's difference from the average flow varied significantly as engine speed and load were varied.[23, 24]

Typical quantities that characterize manifold air flow are given in Table 7.1 for a four- and an eight-cylinder spark-ignition engine. The volume of mixture pulled into each cylinder per cycle is about the same as the volume of one direct flow path between the throttle plate and inlet valve. Thus, one stroke loads the manifold, the next one pulls the charge into the cylinder.

An additional phenomenon becomes important when engine load is changed by opening or closing the throttle: the mass of air in the induction system volume takes a finite time to adjust to the new engine operating conditions. For example, as the throttle is opened the air flow into the manifold increases as the throttle open area increases. However, due to the finite volume of

TABLE 7.1
Parameters that characterize manifold air flow

Engine geometry	I-4†	V-8‡
Typical flow-path distance between throttle bore and intake valve, cm	33	30
Average intake-passage flow area, cm^2	9.4	16
Volume of one direct flow path from throttle bore to intake valve, cm^3	300	500
Range of speeds, etc.	**Maximum**	**Minimum**
Crankshaft, rev/min	5000	650
Peak air velocity in manifold branch, m/s	130†, 100‡	15
Peak Reynolds number in manifold branch	4×10^5	5×10^4
Duration of individual cylinder intake process, ms	6	46

† 1.8-dm^3 four-cylinder in-line SI engine.[21]
‡ 5.6-dm^3 V eight-cylinder SI engine.[25]

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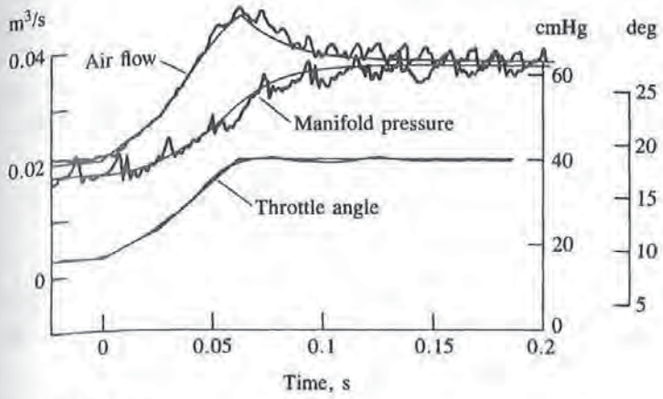


FIGURE 7-24
Throttle angle, intake manifold pressure, and air flow rate past the throttle versus time for 10° part-load throttle opening. 5-dm³ V-8 engine.[25]

the manifold, the pressure level in the manifold increases more slowly than would be the case if steady-state conditions prevailed at each throttle position. Thus, the pressure difference across the throttle is larger than it would be under steady flow conditions and the throttle air flow overshoots its steady-state value. The air flow into each cylinder depends on the pressure in the manifold, so this lags the throttle air flow. This transient air-flow phenomenon affects fuel metering. For throttle-body injection or a carburetor, fuel flow should be related to throttle air flow. For port fuel injection, fuel flow should be related to cylinder air flow. Actual results for the air flow rate and manifold pressure in response to an opening of the throttle (increase in throttle angle) are shown in Fig. 7-24. The overshoot in throttle air flow and lag in manifold pressure as the throttle angle is increased are evident. Opposite effects will occur for a decrease in throttle angle.

AIR-FLOW MODELS. Several models of the flow in an intake manifold have been proposed.[26, 27] One simple manifold model that describes many of the above phenomena is the *plenum* or *filling and emptying* model. It is based on the assumption that at any given time the manifold pressure is uniform. The continuity equation for air flow into and out of the intake manifold is

$$\frac{dm_{a,\,m}}{dt} = \dot{m}_{a,\,\mathrm{th}} - \sum \dot{m}_{a,\,\mathrm{cyl}} \tag{7.21}$$

where $m_{a,\,m}$ is the mass of air in the manifold, and $\dot{m}_{a,\,\mathrm{th}}$ and $\dot{m}_{a,\,\mathrm{cyl}}$ are the air mass flow rates past the throttle and into each cylinder, respectively. The flow rate past the throttle is given by Eq. (7.19) or (7.20). For manifold pressures sufficiently low to choke the flow past the throttle plate, the flow rate is independent of manifold pressure. The mass flow rate to the engine cylinders can be modeled at several levels of accuracy. The air flow through the valve to each cylinder can be computed from the valve area, discharge coefficient, and pressure

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difference across the valve; or a sine wave function can be assumed. In the general case, Eq. (7.21) must be combined with the first law for an open system (see Sec. 14.2.2). For calculating the manifold response to a change in load or throttle setting, simplifying assumptions can be made. A quasi-steady approximation for the cylinder air flow:

$$\sum \dot{m}_{a,\,\mathrm{cyl}} = \frac{\eta_v \rho_{a,\,m} V_d N}{2}$$

is usually adequate, and the air temperature can be assumed constant.[25] Then, using the ideal gas law for the manifold, $p_m V_m = m_{a,\,m} R_a T_m$, Eq. (7.21) can be written as

$$\frac{dp_m}{dt} + \frac{\eta_v V_d N}{2V_m} p_m = \dot{m}_{a,\,\mathrm{th}} \frac{RT_m}{V_m} \tag{7.22}$$

Both η_v and $\dot{m}_{a,\,\mathrm{th}}$ have some dependence on p_m [e.g., see Eq. (6.2)]. In the absence of this weak dependence, Eq. (7.22) would be a first-order equation for p_m with a time constant $\tau = 2V_m/(\eta_v V_d N) \approx V_m/\dot{V}_{\mathrm{cyl}}$, which is 2 to 4 times the intake stroke duration. The smooth curves in Fig. 7-24 are predictions made with Eq. (7.22) and show good agreement with the data. The plenum model is useful for investigating manifold pressure variations that result from load changes. It provides no information concerning pressure variations associated with momentum effects.

Helmholtz resonator models for the intake system have been proposed. This type of model can predict the resonant frequencies of the combined intake and engine cylinder system, and hence the engine speeds at which increases in air flow due to intake tuning occur. It does not predict the magnitude of the increase in volumetric efficiency. The Helmholtz resonator theory analyzes what happens during one inlet stroke, as the air in the manifold pipe is acted on by a forcing function produced by the piston motion. As the piston moves downward during the intake stroke, a reduced pressure occurs at the inlet valve relative to the pressure at the open end of the inlet pipe. A rarefraction wave travels down the intake pipe to the open end and is reflected as a compression wave. A positive tuning effect occurs when the compression wave arrives at the inlet valve as the valve is closing.[27] A single-cylinder engine modeled as a Helmholtz resonator is shown in Fig. 7-25*a*. The effective resonator volume V_{eff} is chosen to be one-half of the displaced volume plus the clearance volume; the piston velocity is then close to its maximum and the pressure in the inlet system close to its minimum. The tuning peak occurs when the natural frequency of the cylinder volume coupled to the pipe is about twice the piston frequency. For a single-cylinder, fed by a single pipe open to the atmosphere, the resonant tuning speed N_t is given by

$$N_t(\mathrm{rev/min}) = \frac{955}{K}\, a\left(\frac{A}{lV_{\mathrm{eff}}}\right)^{1/2} \tag{7.23}$$

where a is the sound speed (m/s), A the effective cross-sectional area of the inlet system (cm^2), l the effective length of the inlet system (cm), K a constant equal to about 2 for most engines, and $V_{\mathrm{eff}} = V_d(r_c + 1)/[2(r_c - 1)]$ (cm^3).[28]

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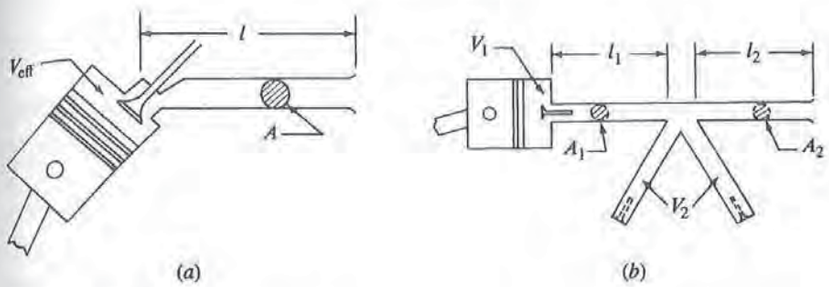


FIGURE 7-25
Helmholtz resonator models for (*a*) single-cylinder engine and (*b*) multicylinder engine.[27]

The Helmholtz theory for multicylinder engines treats the pipes of cylinders not undergoing induction as an additional volume. The two pipes, (l_1, A_1) and (l_2, A_2), and two volumes, V_1 and V_2, in Fig. 7-25*b* form a vibrating system with two degrees of freedom and two resonant frequencies. The following equation, based on an electrical analog (in which capacitors represent volumes and inductors pipes), gives the two frequencies at which the manifold shown in Fig. 7-25*b* would be tuned:[28]

$$f_{\mp} = \frac{1}{2\pi}\left\{\frac{(\alpha\beta + \alpha + 1) \mp [(\alpha\beta + \alpha + 1)^2 - 4\alpha\beta]^{1/2}}{2\alpha\beta L_1 C_1}\right\}^{1/2} \tag{7.24}$$

where $\alpha = L_2/L_1$, $\beta = C_2/C_1$, $C_1 = V_1$, $C_2 = V_2$, $L_1 = (l/A)_1$, $L_2 = (l/A)_2$, and $V_{eff} = V_1$. The Helmholtz theory predicts the engine speeds at which positive tuning resonances occur with reasonable accuracy.[27]

The dynamics of the flow in multicylinder intake (and exhaust) systems can be modeled most completely using one-dimensional unsteady compressible flow equations. The standard method of solution of the governing equations has been the method of characteristics (see Benson[29]). Recently, finite difference techniques which are more efficient have been used.[30] The assumptions usually made in this type of analysis are:

1. The intake (or exhaust) system can be modeled as a combination of pipes, junctions, and plenums.
2. Flow in the pipes is one dimensional and no axial heat conduction occurs.
3. States in the engine cylinders and plenums are uniform in space.
4. Boundary conditions are considered quasi steady.
5. Coefficients of discharge, heat transfer, pipe friction, and bend losses for steady flow are valid for unsteady flow.
6. The gases can be modeled as ideal gases.

This approach to intake and exhaust flow analysis is discussed more fully in Sec. 14.3.4.

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7.6.3 Fuel-Flow Phenomena

TRANSPORT PROCESSES. With conventional spark-ignition engine liquid-fuel metering systems, the fuel enters the air stream as a liquid jet. The liquid jet atomizes into droplets. These mix with the air and also deposit on the walls of the intake system components. The droplets vaporize; vaporization of the liquid fuel on the walls occurs. The flow of liquid fuel along the walls can be significant. The transport of fuel as vapor, droplets, and liquid streams or films can all be important. The fuel transport processes in the intake system are obviously extremely complex.

The details of the fuel transport process are different for multipoint fuel-injection systems than for carburetor and throttle-body injection systems. For the latter systems, fuel must be transported past the throttle plate and through the complete intake manifold. For the former systems, the liquid fuel is injected in the inlet port, toward the back of the intake valve. For all these practical fuel metering systems, the quality of the mixture entering the engine is imperfect. The fuel, air, recycled exhaust, mixture is not homogeneous; the fuel may not be fully vaporized as it enters the engine. The charge going to each cylinder is not usually uniform in fuel/air ratio throughout its volume, and the distribution of fuel between the different engine cylinders is not exactly equal. During engine transients, when engine fuel and air requirements and manifold conditions change, it is obvious that the above fuel transport processes will not all vary with time in the same way. Thus, in addition to the transient non-quasi-steady air-flow phenomena described above, there are transient fuel-flow phenomena. These have to be compensated for in the fuel metering strategy.

Since gasoline, the standard spark-ignition engine fuel, is a mixture of a large number of individual hydrocarbons it has a boiling temperature range rather than a single boiling point. Typically, this range is 30 to 200°C. Individual hydrocarbons have the saturation pressure-temperature relationships of a pure substance. The lower the molecular weight, the higher will be the saturated vapor pressure at a given temperature. The boiling point of hydrocarbons depends primarily on their molecular weight: the vapor pressure also depends on molecular structure. The equilibrium state of a hydrocarbon-air mixture depends therefore on the vapor pressure of the hydrocarbon at the given temperature, the relative amounts of the hydrocarbon and air, and the total pressure of the mixture. The equilibrium fraction of fuel evaporated at a given temperature and pressure can be calculated from Bridgeman charts[31] and the distillation characteristics of the fuel (defined by the ASTM distillation curve[32]). Figure 7-26*a* shows the effect of mixture temperature on percent of indolene fuel (a specific gasoline) evaporated at equilibrium at atmospheric pressure. Figure 7-26*b* shows the effect of reduced manifold pressure on the amount evaporated.[18] While insufficient time is usually available in the manifold to establish equilibrium, the trends shown are indicative of what happens in practice: lower pressures increase the relative amount of fuel vaporized and charge heating is usually required to vaporize a substantial fraction of the fuel.

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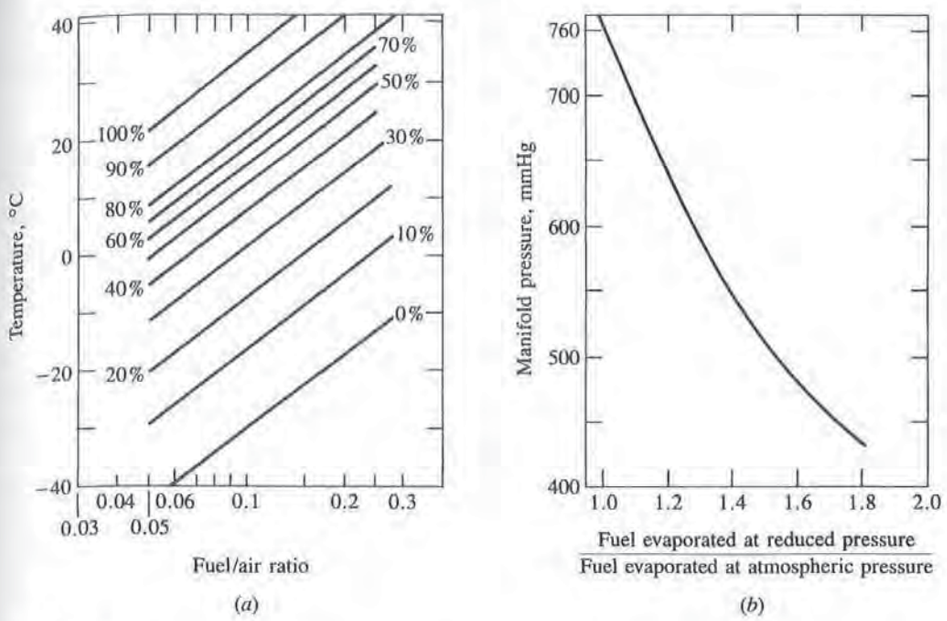


FIGURE 7-26
(*a*) Percentage of indolene fuel evaporated at equilibrium at 1 atmosphere pressure. (*b*) Effect of pressure on amount of indolene fuel evaporated.[18]

For carbureted and throttle-body injection systems, the fuel path is the following. Until the throttle plate is close to fully opened, most of the fuel metered into the air stream impacts on the throttle plate and throttle-body walls. Only a modest fraction of the fuel vaporizes upstream of the throttle. The liquid is reentrained as the air flows at high velocity past the throttle plate. The fuel does not usually divide equally on either side of the throttle plate axis. The air undergoes a 90° bend in the plenum beneath the throttle; much of the fuel which has not evaporated is impacted on the manifold floor. Observations of fuel behavior in intake manifolds with viewing ports or transparent sections show that there is substantial liquid fuel on the walls with carburetor fuel metering systems. Figure 7-27 shows the engine conditions under which liquid fuel was observed on the floor of the manifold plenum beneath the throttle plate and in the manifold runners, in a standard four-cylinder production engine.[23] This manifold was heated by engine coolant at 90°C. The greatest amount of liquid was present at high engine loads and low speeds. Heating the manifold to a higher temperature with steam at 115°C resulted in a substantial reduction in the amount of liquid: there was no extensive puddling on the plenum floor, liquid films or rivulets were observed in a zone bounded by 120 mmHg vacuum and 2500 rev/min, and there were no films or rivulets in the manifold runner. Depending on engine operating conditions, transport of fuel as a liquid film or rivulet in the manifold and vaporization from these liquid fuel films and rivulets and subsequent transport as vapor may occur.

Vaporized fuel and liquid droplets which remain suspended in the air

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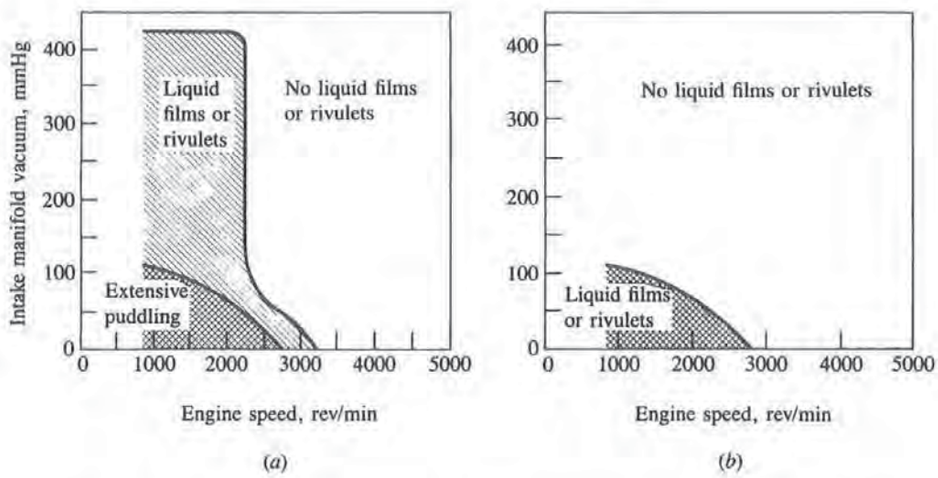


FIGURE 7-27
Regions of engine load and speed range where extensive pools or puddles, liquid films, or rivulets were observed: (*a*) on manifold plenum floor and (*b*) in manifold runner. Four-cylinder automobile engine. Manifold heated by coolant at 90°C.[23]

stream will be transported with the air stream. However, droplet deposition on the manifold walls may occur due to gravitational settling and to inertial effects as the flow goes round bends in the manifold.

The fuel transport processes for port fuel-injection systems are different and will depend significantly on the timing and duration of the injection pulse. Fuel is injected onto the back of the inlet valve (and surrounding port wall), usually while the valve is closed or only partly open. Vaporization of liquid fuel off the valve and walls occurs, enhanced by the backflow of hot residual gases from the cylinder (especially at part load). There is evidence that, even under fully warmed-up engine conditions, some fuel is carried as liquid drops into the cylinder.[33]

FUEL DROPLET BEHAVIOR. With carburetor and throttle-body injection systems, the liquid fuel atomizes as it enters the air stream. In the carburetor venturi this occurs as the fuel-air emulsion from the fuel jet(s) enters the high-velocity (>100 m/s) air stream. With an injector, the velocity of the liquid jet as it exits the nozzle is high enough to shatter the flowing liquid, and its interaction with the coaxial air flow further atomizes the fuel. Typical droplet-size distributions are not well defined; size would vary over the load and speed range. Droplet diameters in the 25 to 100 μm range are usually assumed to be representative: larger drops are also produced. The liquid fuel drops are accelerated by the surrounding air stream and start to vaporize. Vaporization rates have been calculated using established formulas for heat and mass transfer between a droplet and a surrounding flowing gas stream (see Ref. 34 for a review of methods of calculating droplet vaporization rates). Calculations of fuel vaporization in a carburetor venturi and upstream of the throttle plate show that the temperature

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of the liquid fuel droplets decreases rapidly (by up to about 30°C[35]), and the fraction of the fuel vaporized is small (in the 2 to 15 percent range[35, 36]).

Liquid fuel drops, due to their density being many times that of the air, will not exactly follow the air flow. Droplet impaction on the walls may occur as the flow changes direction, and the greater inertia of the droplets causes them to move across the streamlines to the outer wall. Deposition on the manifold floor due to gravity may also occur. The equation of motion for an individual droplet in a flowing gas stream is

$$(\tfrac{1}{6}\pi D_d^3 \rho_f)\mathbf{a} = m_d \mathbf{g} - \tfrac{1}{2}(\mathbf{v}_d - \mathbf{v}_g)|\mathbf{v}_d - \mathbf{v}_g|\rho_g C_D \frac{\pi D_d^2}{4} \tag{7.25}$$

where D_d is the droplet diameter, ρ_f and ρ_g are liquid and gas densities, $\mathbf{v}_d$ and $\mathbf{v}_g$ are the droplet and gas velocities, $\mathbf{a}$ is the droplet acceleration, $\mathbf{g}$ acceleration due to gravity, and C_D is the drag coefficient. For $6 < \mathrm{Re} < 500$ the drag coefficient of an evaporating droplet is a strong function of the Reynolds number, Re: e.g.,

$$C_D = 27\ \mathrm{Re}^{-0.84} \tag{7.26}$$

where $\mathrm{Re} = (\rho_g D_d|\mathbf{v}_d - \mathbf{v}_g|/\mu_g)$.

Studies of droplet impaction and evaporation using the above equations and typical manifold conditions and geometries indicate the following.[26, 35, 37] For 90° bends, drops of less than 10 μm diameter are essentially carried by the gas stream (<10 percent impaction); almost all droplets larger than 25 μm impact on the walls. Droplet sizes produced first in the carburetor venturi or fuel injector spray and then by secondary atomization as liquid fuel is entrained from the throttle plate and throttle-body walls depend on the local gas velocity: higher local relative velocities between the gas and liquid produce smaller drop sizes. Approximate estimates which combine the two phenomena outlined above show that at low engine air flow rates, almost all of the fuel will impact first on the throttle plate and then on the manifold floor as the flow turns 90° into the manifold runners. At high air flows, because the drops are smaller, a substantial fraction of the drops may stay entrained in the air flow. Secondary atomization at the throttle at part-load operating conditions is important to the fuel transport process: the very high air velocities at the edge of the throttle plate produce droplets of order or less than 10 μm diameter. However, coalescence and deposition on the walls and subsequent reentrainment probably increase the mean droplet size. In the manifold, gravitational settling of large (>100 μm) droplets would occur at low air flow rates,[38] but these drops are also likely to impact the walls due to their inertia as the flow is turned.

Estimates of droplet evaporation rates in the manifold indicate the following. With a representative residence time in the manifold of about one crank revolution (10 ms at 6000 rev/min, 100 ms at 600 rev/min), only drops of size less than about 10 μm will evaporate at the maximum speed; 100 μm droplets will not vaporize fully at any speed. Most of these large droplets impact on the walls, anyway. Drops small enough to be carried by the air stream are likely to vaporize in the manifold.[26]

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FUEL-FILM BEHAVIOR. The fuel which impacts on the wall will also vaporize and, depending on where in the manifold deposition occurs and the local manifold geometry, may be transported along the manifold as a liquid film or rivulet. If the vaporization rate off the wall is sufficiently high, then a liquid film will not build up. Any liquid film or pool on the manifold floor or walls is important because it introduces additional fuel transport processes—deposition, liquid transport, and evaporation—which together have a much longer time constant than the air transport process. Thus changes in the air and the fuel flow into each engine cylinder, during a change in engine load, will not occur in phase with each other unless compensation is made for the slower fuel transport.

Several models of the behavior of liquid-fuel wall-films have been developed. One approach analyzes a liquid puddle on the floor of the manifold plenum.[38] Metered fuel enters the puddle; fuel leaves primarily through vaporization. The equation for rate of change of mass of fuel in the puddle is

$$\dot{m}_{f,p} = \dot{m}_{f,\text{in}} - \dot{m}_{f,\text{out}} = x\dot{m}_{f,m} - \frac{m_{f,p}}{\tau} \tag{7.27}$$

where $m_{f,p}$ is the mass of fuel in the puddle, $\dot{m}_{f,m}$ is the metered fuel flow rate, and x is the fraction of the metered flow that enters the puddle. It is assumed that the reentrainment/evaporation rate is proportional to the mass of fuel in the puddle divided by the characteristic time τ of the reentrainment/evaporation process. The puddle behavior predicted by this model in response to a step increase in engine load is shown in Fig. 7-28*a*. Because only part $(1 - x)$ of the fuel flows directly with the air, as the throttle is opened rapidly a lean air/fuel ratio excursion is predicted. Figure 7-28*b* shows that this behavior (without any metering compensation) is observed in practice. Estimates of the volume of fuel in the puddle (for a 5-liter V-8 engine) are of order 1000 mm^3, and increase with

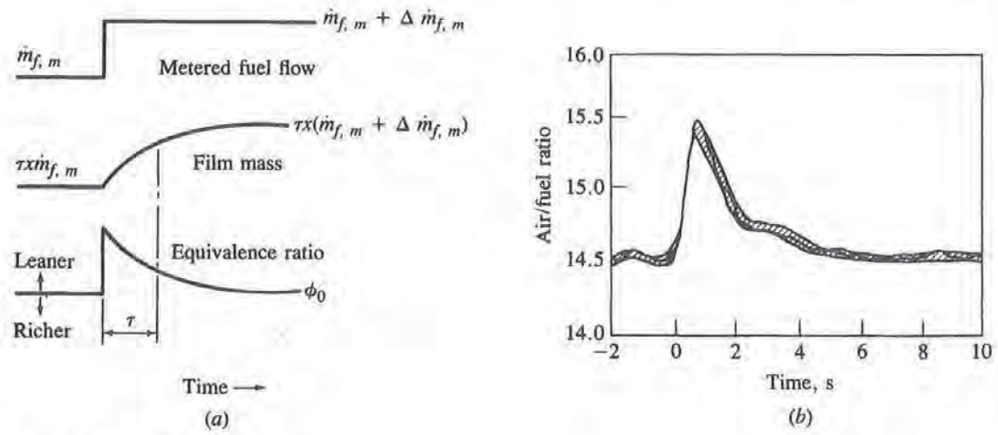


FIGURE 7-28
(*a*) Predicted behavior of the fuel film for an uncompensated step change in engine operating conditions. (*b*) Observed variation in air/fuel ratio for uncompensated throttle opening at 1600 rev/min which increased manifold pressure from 48 to 61 cmHg.[38]

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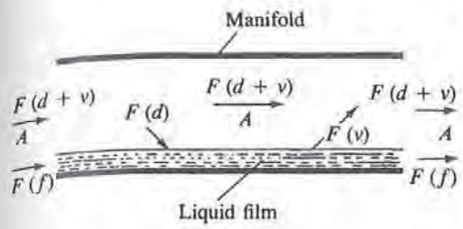


FIGURE 7-29
Schematic of fuel flow paths in the manifold when liquid film flows along the manifold runner floor.
A Air
F Fuel
f Liquid fuel film
d Liquid fuel droplets
v Fuel vapor

increasing load and speed. The time constant is of order 2 seconds for a fully warmed-up engine; it varies with engine operating conditions and is especially sensitive to intake manifold temperature. Such models have been used primarily to develop fuel metering strategies which compensate for the fuel transport lag.[38]

An alternative model, for liquid film flow in the manifold runner and branch, has been developed.[37] Fuel is deposited on the manifold walls and forms a film which flows toward the cylinder due to the shear force at the gas/liquid interface as shown in Fig. 7-29. Vaporization from the film also occurs. An analysis of the dynamics of the fuel film leads to expressions for steady-state film velocity and thickness. As air and metered fuel flows change due to a throttle position change, the characteristic time for reestablishing steady state is $l/(2u_f)$, where l is the manifold length and u_f the average film velocity. This characteristic response time is of order 1 second for typical manifold conditions, in approximate agreement with values obtained from transient engine experiments.

A more extensive analysis of both fuel droplet and film evaporation in a complete carburetor, throttle, manifold system,[35] with a multicomponent model for gasoline based on its distillation curve, indicates the following phenomena are important. Secondary atomization of the liquid fuel at the throttle, which produces the smallest droplet sizes when the throttle open angle is small, significantly increases the fraction of fuel evaporation in the manifold. Increasing inlet air temperature increases the fraction of fuel vaporized; this effect is larger at lower loads since secondary atomization under these conditions increases the liquid fuel surface area significantly. Heating the wall, which heats the liquid film on the wall directly, provides a greater increase in fraction evaporated than does equivalent heating of the air flow upstream of the carburetor. Due to the multicomponent nature of the fuel, the residual liquid fuel composition changes significantly as fuel is transported from the carburetor to the manifold exit. Of the full boiling range liquid composition at entry, all the light ends, most of the mid-range components, but only a modest amount of the high boiling point fraction have evaporated at the manifold exit. The predicted fuel fraction evaporated ranged from 40 to 60 percent for the conditions examined. One set of measurements of the fraction of fuel vaporized in the manifold of a warmed-up four-cylinder engine showed that 70 to 80 percent of the fuel had vaporized, confirming that under these operating conditions "most" but not necessarily "all" the fuel enters the cylinder in vapor form.[39]

The engine operating range where fuel puddling, fuel films, and rivulets are observed (see Fig. 7-27) can now be explained. At light load, secondary atom-

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ization at the throttle and the lower manifold pressure would reduce the amount of liquid fuel impinging on the manifold plenum floor. Also, typical manifold heating at light load substantially exceeds the heat required to vaporize the fuel completely,[40] and manifold floor temperatures are of order 15°C higher than at full load. All the above is consistent with less liquid on the floor and none in the runners at light load, compared to what occurs at full load. At high speed, drop sizes produced in the carburetor are much smaller, so impingement on the walls is much reduced.

The fuel flow to each cylinder per cycle is not exactly the same. There is a *geometric variation* where fuel is not divided equally among individual cylinders. There is also a *time variation* under steady-state engine conditions where the air/fuel ratio in a given cylinder varies cycle-by-cycle.[41] Data on time-averaged air/fuel ratios in each cylinder of multicylinder engines show that the extent of the maldistribution varies from engine to engine, and for a particular engine varies over the load and speed range. Spreads in the equivalence ratio (maximum to minimum) of about 5 percent of the mean value are typical at light load for carbureted engines. Largest variations between cylinders usually occur at wide-open throttle. WOT spreads in the equivalence ratio of about 15 percent of the mean appear to be typical, again for carbureted engines, while spreads as high as 20 to 30 percent are not uncommon at particular speeds for some engines.[23, 40] Time variations are less well defined; the limited data available suggest they could be of comparable magnitude.[41]

With multipoint port fuel-injection systems, the fuel transport processes are substantially different and are not well understood. Air-flow phenomena are comparable to those with carbureted or throttle-body injection systems. However, manifold design can be optimized for air flow alone since fuel transport from the throttle through the manifold is no longer a design constraint. Because the manufacture and operation of individual fuel injectors are not identical, there is still some variation in fuel mass injected cylinder-to-cylinder and cycle-to-cycle. Since individual cylinder air flows depend on the design of the manifold, whereas the amount of fuel injected does not, uniform air distribution is especially important with port injection systems. The fuel vaporization and transport processes will depend on the duration of injection and the timing of injection pulse(s) relative to the intake valve-lift profile. Some of the injected fuel will impinge on the port walls, valve stem, and backside of the valve, especially when injection toward a closed valve occurs. Backflow of hot residual gases at part-load operation will have a substantial effect on fuel vaporization. Compensation for fuel lag during transient engine operation is still required; sudden throttle openings are accompanied by a "lean spike" in the mixture delivered to the engine, comparable to though smaller than that shown in Fig. 7-28 for a throttle-body fuel-injection system. Thus wall wetting, evaporation off the wall, and liquid flow along the wall are all likely to be important with port fuel-injection systems also.

With port fuel-injection systems, liquid fuel enters the cylinder and droplets are present during intake and compression. Limited measurements have been made of the distribution, size, and number density of these fuel droplets. During

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intake, the droplet number density in the clearance volume increased to a maximum at the end of injection (the injection lasted from 45 to 153° ATC) and then decreased due to evaporation during compression to a very small value at the time of ignition. Average droplet size during intake was 10 to 20 μm in diameter; it increased during compression as the smaller drops in the distribution evaporated. At the conditions tested, some 10 to 20 percent of the fuel was in droplet form at the end of injection. At ignition, the surviving droplets contained a negligible fraction of the fuel. During injection, the distribution of droplets across the clearance volume was nonuniform. It became much more uniform with time, after injection ended.[33]

PROBLEMS

7.1. The equivalence ratio in a conventional spark-ignition engine varies from no load (idle) to full load, at a fixed engine speed, as shown at the top of Fig. P7-1. (By load is meant the percentage of the maximum brake torque at that speed.) Also shown is the variation in total friction (pumping plus mechanical rubbing plus accessory friction). Using formats similar to those shown, draw *carefully proportioned qualitative* graphs of the following parameters versus load (0 to 100 percent):

Combustion efficiency, η_c
Gross indicated fuel conversion efficiency, $\eta_{f,ig}$
Gross indicated mean effective pressure, imep_g
Brake mean effective pressure, bmep
Mechanical efficiency, η_m

Indicate clearly where the maximum occurs if there is one, and where the value is zero or unity or some other obvious value, if appropriate. Provide a *brief justification* for the shape of the curves you draw.

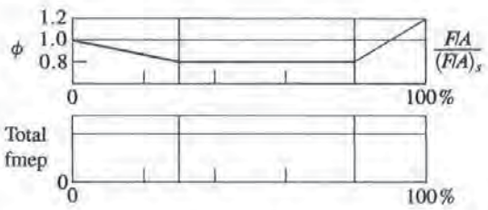


FIGURE P7-1

7.2. The four-cylinder spark-ignition engine shown in the figure uses an oxygen sensor in the exhaust system to determine whether the exhaust gas composition is lean or rich of the stoichiometric point, and a throttle-body injection system with feedback to maintain engine operation close to stoichiometric. However, since there is a time delay between a change in the fuel/air ratio at the injector location and the detection of that change by the sensor (corresponding to the flow time between the injector and the sensor), the control system shown results in oscillations in fuel/air ratio about the stoichiometric point.

(*a*) Estimate the average flow time between the injector and the sensor at an engine speed of 2000 rev/min.

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(*b*) The sensor and control unit provide a voltage V of $+V_s$ volts when the fuel/air equivalence ratio ϕ is less than one and a voltage of $-V_s$ volts when ϕ is greater than one. The feedback injection system provides a fuel/air ratio (F/A) given by

$$\left(\frac{F}{A}\right) = \left(\frac{F}{A}\right)_{t=0} (1 + CVt)$$

where t is the time (in seconds) after the voltage signal last changed sign, $(F/A)_{t=0}$ is the fuel/air ratio at the injector at $t = 0$, and C is a constant. Develop carefully proportioned quantitative sketches of the variation in the fuel/air ratio at the injector and at the exhaust sensor, with time, showing the phase relation between the two curves. Explain briefly how you developed these graphs.

(*c*) Find the value of the constant C, in volts^{-1}-seconds^{-1} (the feedback system gain), such that (F/A) variations about the stoichiometric value do not exceed ± 10 percent for $V_s = 1$ V.

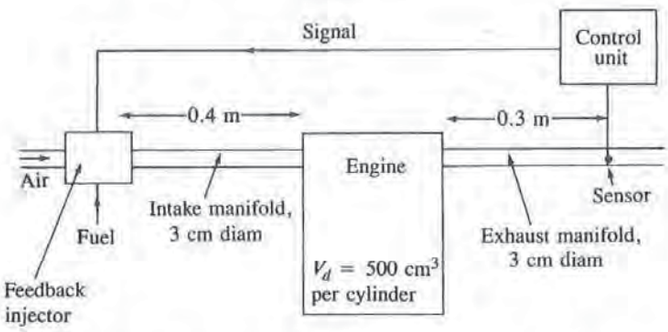


FIGURE P7-2

7.3. In many spark-ignition engines, liquid fuel is added to the inlet air upstream of the inlet manifold above the throttle. The inlet manifold is heated to ensure that under steady-state conditions the fuel is vaporized before the mixture enters the cylinder.

(*a*) At normal wide-open throttle operating conditions, in a four-stroke cycle 1.6-dm^3 displacement four-cylinder engine, at 2500 rev/min, the temperature of the air entering the carburetor is 40°C. The heat of vaporization of the fuel is 350 kJ/kg and the rate of heat transfer to the intake mixture is 1.4 kW. Estimate the temperature of the inlet mixture as it passes through the inlet valve, assuming that the fuel is fully vaporized. The volumetric efficiency is 0.85. The air density is 1.06 kg/m^3 and c_p for air is 1 kJ/kg · K. You may neglect the effects of the heat capacity of the liquid and vapor fuel.

(*b*) With port electronic fuel-injection systems, the fuel is injected directly into the intake port. The intake manifold is no longer heated. However, the fuel is only partly vaporized prior to entering the cylinder. Estimate the mixture temperature as it passes through the inlet valve with the EFI system, assuming that the air temperature entering the intake manifold is still 40°C and 50 percent of the fuel is vaporized.

(*c*) Estimate the ratio of the maximum indicated power obtained at these conditions with this engine with a carburetor, to the maximum power obtained with port fuel injection. Assume that the inlet valve is the dominant restriction in the flow into the engine and that the pressure ratio across the inlet valve is the same for both carbureted and port-injection fueled engines. The intake mixture pressure and equivalence ratio remain the same in both these cases.

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7.4. Port fuel-injection systems are replacing carburetors in automobile spark-ignition engines. List the major advantages and any disadvantages of fuel metering with port fuel injection relative to carburetion.

7.5. With multipoint port fuel injection and single-point injection systems, the fuel flow rate is controlled by the injection pulse duration. If each injector operates continuously at the maximum rated power point (wide-open throttle, $A/F = 12$, 5500 rev/min) of an automobile spark-ignition engine, estimate approximately the injection pulse duration (in crank angle degrees) for the same engine at idle. Idle conditions are: 700 rev/min, 0.3 atm inlet manifold pressure, stoichiometric mixture.

7.6. The fuel-air cycle results indicate that the maximum imep is obtained with gasoline-air mixtures at equivalence ratios of about 1.0. In practice, the maximum wide-open throttle power of a spark-ignition engine at a given speed is obtained with an air/fuel ratio of about 12. The vaporization of the additional gasoline lowers the temperature of the inlet air and the richer mixture has a lower ratio of specific heats γ_u during compression. Estimate approximately the change in mixture temperature due to vaporization of the additional fuel used to decrease A/F from 14.6 (an equivalence ratio of 1.0) to 12.2 in the intake system, and the combined effect of vaporization and lower γ_u on the unburned mixture temperature at WOT when the cylinder pressure is at its peak of 40 atm. (The principal effect of the richer mixture is its impact on knock.)

7.7. (*a*) Plot dimensionless throttle plate open area $4A_{th}/(\pi D^2)$ as a function of throttle plate angle ψ. Assume $\psi_0 = 10°$, D (throttle bore diameter) = 57 mm, d (throttle shaft diameter) = 10.4 mm. What is the *throttle plate* area?

(*b*) Estimate the average velocity of the air flowing through the throttle plate open area for $\psi = 26°$ at 3000 rev/min and $\psi = 36°$ at 2000 rev/min. Use the relationship between ψ, engine speed, and inlet manifold pressure given in Fig. 7-22. Assume a discharge coefficient $C_D = 0.8$.

(*c*) For the throttle of part (*a*), estimate and plot the total force on the throttle plate and shaft, and the force parallel and perpendicular to the throttle bore axis (i.e., in the mean flow direction and normal to that direction) as a function of throttle angle at 2000 rev/min. Again use Fig. 7-22 for the relationship between ψ and inlet manifold pressure.

7.8. For the engine and intake manifold shown in Fig. 7-23, estimate the ratio of the intake manifold runner cross-sectional area to $(\pi B^2/4)$, the ratio of the length of the flow path from the intake manifold entrance to the inlet valve seat to the bore, the ratio of the volume of each inlet port to each cylinder's displaced volume, and the ratio of the volume of each intake manifold runner to each cylinder's displaced volume. The cylinder bore is 89 mm.

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CHAPTER 8

CHARGE MOTION WITHIN THE CYLINDER

Gas motion within the engine cylinder is one of the major factors that controls the combustion process in spark-ignition engines and the fuel-air mixing and combustion processes in diesel engines. It also has a significant impact on heat transfer. Both the bulk gas motion and the turbulence characteristics of the flow are important. The initial in-cylinder flow pattern is set up by the intake process. It may then be substantially modified during compression. This chapter reviews the important features of gas motion within the cylinder set up by flows into and out of the cylinder through valves or ports, and by the motion of the piston.

8.1 INTAKE JET FLOW

The engine intake process governs many important aspects of the flow within the cylinder. In four-stroke cycle engines, the inlet valve is the minimum area for the flow (see Sec. 6.3) so gas velocities at the valve are the highest velocities set up during the intake process. The gas issues from the valve opening into the cylinder as a conical jet and the radial and axial velocities in the jet are about 10 times the mean piston speed. Figure 8-1 shows the radial and axial velocity components close to the valve exit, measured during the intake process, in a motored model engine with transparent walls and single valve located on the cylinder axis, using laser doppler anemometry (see next section).[1] The jet separates from the valve

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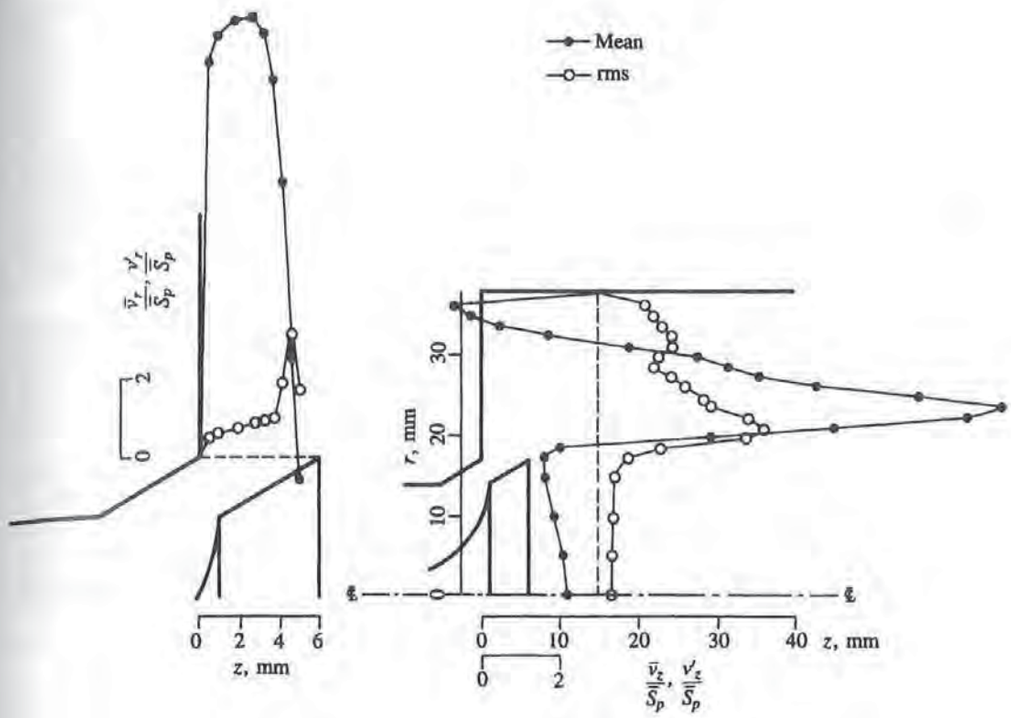


FIGURE 8-1
Radial mean velocity $\bar{v}_r$ and root mean square (rms) velocity fluctuations v'_r at the valve exit plane, and axial mean velocity $\bar{v}_z$ and rms velocity fluctuation v'_z 15 mm below the cylinder head, at 36° ATC in model engine operated at 200 rev/min. Valve lift = 6 mm. Velocities normalized by mean piston speed.[1]

seat and lip, producing shear layers with large velocity gradients which generate turbulence. This separation of the jet sets up recirculation regions beneath the valve head and in the corner between the cylinder wall and cylinder head.

The motion of the intake jet within the cylinder is shown in the schlieren photographs in Fig. 8-2 taken in a transparent engine. This engine had a square cross-section cylinder made up of two quartz walls and two steel walls, to permit easy optical access. The schlieren technique makes regions with density gradients in the flow show up as lighter or darker regions on the film.[2] The engine was throttled to one-half an atmosphere intake pressure, so the jet starts after the intake stroke has commenced, at 35° ATC, following backflow of residual into the intake manifold. The front of the intake jet can be seen propagating from the valve to the cylinder wall at several times the mean piston speed. Once the jet reaches the wall ($\theta > 41°$ ATC), the wall deflects the major portion of the jet downward toward the piston; however, a substantial fraction flows upward toward the cylinder head. The highly turbulent nature of the jet is evident.

The interaction of the intake jet with the wall produces large-scale rotating flow patterns within the cylinder volume. These are easiest to visualize where the engine geometry has been simplified so the flow is axisymmetric. The photograph

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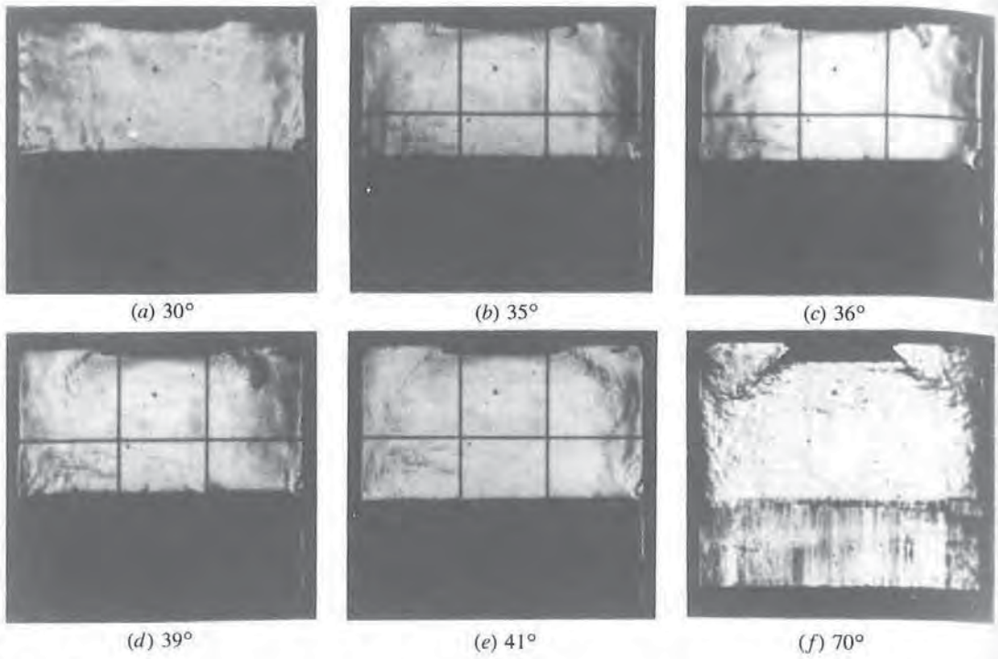


FIGURE 8-2
Sequence of schlieren photographs of intake jet as it develops during intake stroke. Numbers are crank angle degrees after TC.[2]

in Fig. 8-3 of a water analog of an engine intake flow was taken in a transparent model of an engine cylinder and piston. The valve is located in the center of the cylinder head, and the flow into the valve is along the cylinder axis. The experimental parameters have been scaled so that the appropriate dimensionless numbers which govern the flow, the Reynolds and Strouhal numbers, were maintained equal to typical engine values. The photograph shows the major features

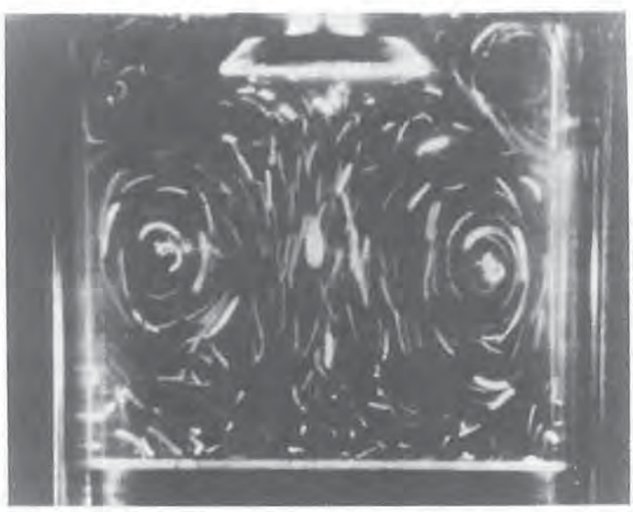
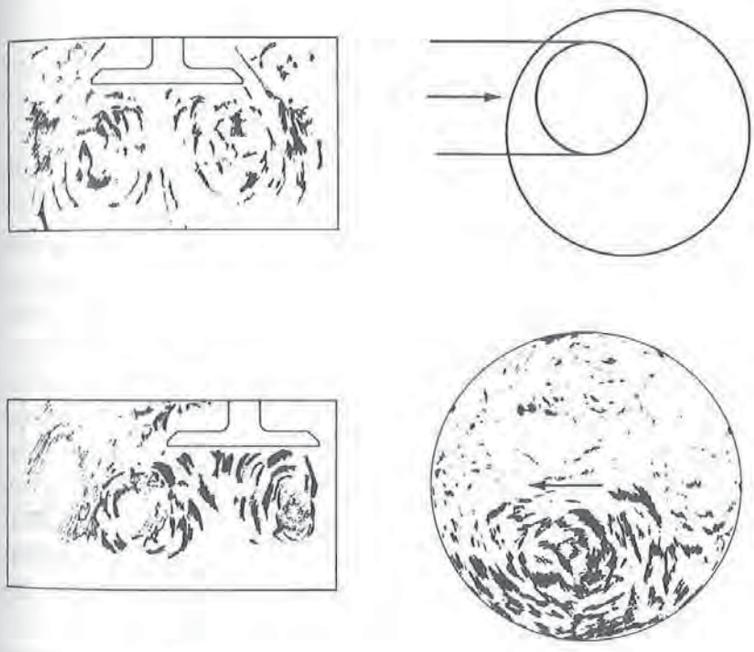


FIGURE 8-3
Large-scale rotating flow pattern set up within the cylinder by the intake jet. Photograph of streak lines in water flow into model engine with axisymmetric valve.[3]

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of the intake generated flow in a thin illuminated plane through the cylinder axis. The streaks are records of the paths of tracer particles in the flow during the period the camera shutter is open. The bulk of the cylinder as the piston moves down is filled with a large ring vortex, whose center moves downward and remains about halfway between the piston and the head. The upper corner of the cylinder contains a smaller vortex, rotating in the opposite direction. These vortices persist until about the end of the intake stroke, when they became unstable and break up.[3]

With inlet valve location and inlet port geometry more typical of normal engine practice, the intake generated flow is more complex. However, the presence of large-scale rotating flow patterns can still be discerned. Figure 8-4*a* shows the effect of off-axis valve location (with the flow into the valve still parallel to the cylinder and valve axis). During the first half of the inlet stroke, at least, a flow pattern similar in character to that in Fig. 8-3 is evident. The vortices are now displaced to one side, however, and the planes of their axes of rotation are no longer perpendicular to the cylinder axis but are tipped at an angle to it. The vortices become unstable and break up earlier in the intake stroke than was the case with the axisymmetric flow.[3] With an offset valve and a normal inlet port configuration which turns the flow through 50 to 70° (see Fig. 6-13), photographs



(*a*) (*b*)

FIGURE 8-4
Sketches from: (*a*) streak photographs of in-cylinder intake generated flow in water analog of intake process in model engine with offset inlet valve, at 90° ATC;[3] (*b*) streak photographs of flow in diametral plane; 30 mm below cylinder head, with intake port and valve geometry shown, with steady water flow into cylinder. Valve lift = 4 mm.[4]

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of the flow pattern in a diametral plane show an additional large-scale rotation. Figure 8-4*b* shows the flow pattern observed in a water-flow model of the cylinder in a plane 30 mm (one-third of the bore) from the cylinder head, with a standard inlet port design. The direction of flow with this vortex pair is toward the left across the center of the cylinder. This flow pattern occurs because the cylinder wall closest to the valve impedes the flow out of the valve and forces the flow on either side of the plane passing through the valve and cylinder axes to circulate around the cylinder in opposite directions. The upper vortex follows the flow direction of the port and becomes larger still as the valve lift increases. The details of this aspect of the intake flow depend on the port design, valve stem orientation, and the valve lift.[4] With suitable port and/or cylinder head design, it is possible to develop a single vortex flow within the bulk of the cylinder. The production and characteristics of such "swirling" flows are reviewed in Sec. 8.3.

In summary, the jet-like character of the intake flow, interacting with the cylinder walls and moving piston, creates large-scale rotating flow patterns within the cylinder. The details of these flows are strongly dependent on the inlet port, valve, and cylinder head geometry. These flows appear to become unstable, either during the intake or the compression process, and break down into three-dimensional turbulent motions. Recirculating flows of this type are usually sensitive to small variations in the flow: hence there are probably substantial cycle-by-cycle flow variations.[5]

8.2 MEAN VELOCITY AND TURBULENCE CHARACTERISTICS

8.2.1 Definitions

The flow processes in the engine cylinder are turbulent. In turbulent flows, the rates of transfer and mixing are several times greater than the rates due to molecular diffusion. This turbulent "diffusion" results from the local fluctuations in the flow field. It leads to increased rates of momentum and heat and mass transfer, and is essential to the satisfactory operation of spark-ignition and diesel engines. Turbulent flows are always dissipative. Viscous shear stresses perform deformation work on the fluid which increases its internal energy at the expense of its turbulence kinetic energy. So energy is required to generate turbulence: if no energy is supplied, turbulence decays. A common source of energy for turbulent velocity fluctuations is shear in the mean flow. Turbulence is rotational and is characterized by high fluctuating vorticity: these vorticity fluctuations can only persist if the velocity fluctuations are three dimensional.[6]

The character of a turbulent flow depends on its environment. In the engine cylinder, the flow involves a complicated combination of turbulent shear layers, recirculating regions, and boundary layers. The flow is unsteady and may exhibit substantial cycle-to-cycle fluctuations. Both large-scale and small-scale turbulent motions are important factors governing the overall behavior of the flow.[5]

An important characteristic of a turbulent flow is its irregularity or ran-

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domness. Statistical methods are therefore used to define such a flow field. The quantities normally used are: the mean velocity, the fluctuating velocity about the mean, and several length and time scales. In a steady turbulent flow situation, the instantaneous local fluid velocity U (in a specific direction) is written:

$$U(t) = \bar{U} + u(t) \tag{8.1}$$

For steady flow, the mean velocity $\bar{U}$ is the time average of $U(t)$:

$$\bar{U} = \lim_{\tau \to \infty} \frac{1}{\tau} \int_{t_0}^{t_0+\tau} U(t)\, dt \tag{8.2}$$

The fluctuating velocity component u is defined by its root mean square value, the turbulence intensity, u':

$$u' = \lim_{\tau \to \infty} \left(\frac{1}{\tau} \int_{t_0}^{t_0+\tau} u^2\, dt \right)^{1/2} \tag{8.3a}$$

Alternatively,

$$u' = \lim_{\tau \to \infty} \left[\frac{1}{\tau} \int_{t_0}^{t_0+\tau} (U^2 - \bar{U}^2)\, dt \right]^{1/2} \tag{8.3b}$$

since the time average of $(u\bar{U})$ is zero.

In engines, the application of these turbulence concepts is complicated by the fact that the flow pattern changes during the engine cycle. Also, while the overall features of the flow repeat each cycle, the details do not because the mean flow can vary significantly from one engine cycle to the next. There are both cycle-to-cycle variations in the mean or bulk flow at any point in the cycle, as well as turbulent fluctuations about that specific cycle's mean flow.

One approach used in quasi-periodic flows such as that which occurs in the engine cylinder is *ensemble-averaging* or *phase-averaging*. Usually, velocity measurements are made over many engine cycles, and over a range of crank angles. The instantaneous velocity at a specific crank angle position θ in a particular cycle i can be written as

$$U(\theta, i) = \bar{U}(\theta, i) + u(\theta, i) \tag{8.4}$$

The ensemble- or phase-averaged velocity, $\bar{U}(\theta)$, is defined as the average of values at a specific phase or crank angle in the basic cycle. Figure 8-5 shows this approach applied schematically to the velocity variation during a two-stroke engine cycle, with small and large cycle-to-cycle variations. The ensemble-averaged velocity is the average over a large number of measurements taken at the same crank angle (two such points are indicated by dots):

$$\bar{U}_{\mathrm{EA}}(\theta) = \frac{1}{N_c} \sum_{i=1}^{N_c} U(\theta, i) \tag{8.5}$$

where N_c is the number of cycles for which data are available. By repeating this

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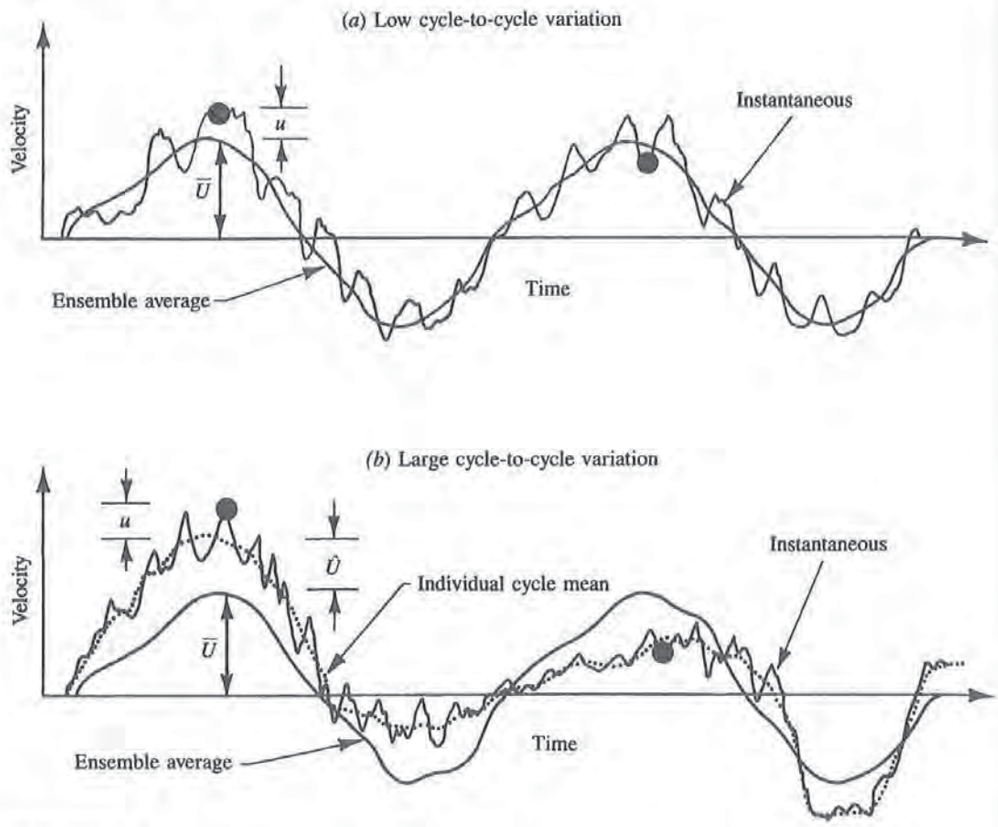


FIGURE 8-5
Schematic of velocity variation with crank angle at a fixed location in the cylinder during two consecutive cycles of an engine. Dots indicate measurements of instantaneous velocity at the same crank angle. Ensemble- or phase-averaged velocity obtained by averaging over a large number of such measurements shown as solid smooth line. Top graph: low cycle-to-cycle flow variations. Here the individual-cycle mean velocity and ensemble-averaged velocity are closely comparable. Bottom graph: large cycle-to-cycle variations. Here the individual-cycle mean velocity (dotted line) is different from the ensemble-averaged mean by $\hat{U}$. The turbulent fluctuation u is then defined in relation to the individual-cycle mean.[5]

process at many crank angle locations, the ensemble-averaged velocity profile over the complete cycle is obtained.

The ensemble-averaged mean velocity is only a function of crank angle since the cyclic variation has been averaged out. The difference between the mean velocity in a particular cycle and the ensemble-averaged mean velocity over many cycles is defined as the cycle-by-cycle variation in mean velocity:

$$\hat{U}(\theta, i) = \bar{U}(\theta, i) - \bar{U}_{EA}(\theta) \tag{8.6}$$

Thus the instantaneous velocity, given by Eq. (8.4), can be split into three components:[7]

$$U(\theta, i) = \bar{U}_{EA}(\theta) + \hat{U}(\theta, i) + u(\theta, i) \tag{8.7}$$

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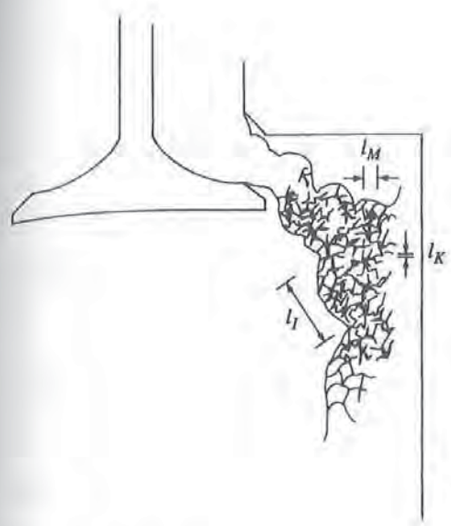


FIGURE 8-6
Schematic of jet created by flow through the intake valve indicating its turbulent structure.[5, 6]

Figure 8-5 illustrates this breakdown of the instantaneous velocity into an ensemble-averaged component, an individual-cycle mean velocity, and a component which randomly fluctuates in time at a particular point in space in a single cycle. This last component is the conventional definition of the turbulent velocity fluctuation. Whether this differs significantly from the fluctuations about the ensemble-averaged velocity depends on whether the cycle-to-cycle fluctuations are small or large. The figure indicates these two extremes.†

In turbulent flows, a number of length scales exist that characterize different aspects of the flow behavior. The largest eddies in the flow are limited in size by the system boundaries. The smallest scales of the turbulent motion are limited by molecular diffusion. The important length scales are illustrated by the schematic of the jet issuing into the cylinder from the intake valve in Fig. 8-6. The eddies responsible for most of the turbulence production during intake are the large eddies in the conical inlet jet flow. These are roughly equal in size to the local jet thickness. This scale is called the *integral scale*, l_I: it is a measure of the largest scale structure of the flow field. Velocity measurements made at two points separated by a distance x significantly less than l_I will correlate with each other; with $x \gg l_I$, no correlation will exist. The integral length scale is, therefore, defined as the integral of the autocorrelation coefficient of the fluctuating velocity at two adjacent points in the flow with respect to the variable distance between

† There is considerable debate as to whether the fluctuating components of the velocity $U(\theta, i)$ defined by Eq. (8.7) (cycle fluctuations in the mean velocity and fluctuations in time about the individual cycle mean) are physically distinct phenomena. The high-frequency fluctuations in velocity are often defined as "turbulence." The low-frequency fluctuations are generally attributed to the variations in the mean flow between individual cycles, a phenomenon that is well established. Whether this distinction is valid has yet to be resolved.

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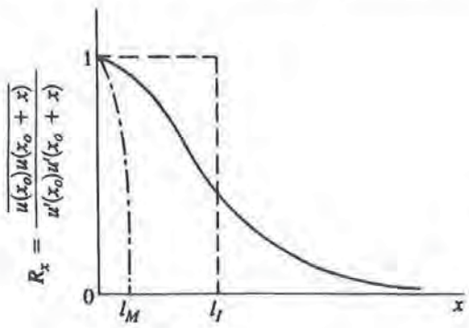


FIGURE 8-7
Spatial velocity autocorrelation R_x as a function of x, defining the integral length scale l_I and the micro length scale l_M.

the points, as shown in Fig. 8-7: i.e.,

$$l_I = \int_0^\infty R_x \, dx \tag{8.8a}$$

where

$$R_x = \frac{1}{N_m - 1} \sum_{i=1}^{N_m} \frac{u(x_0)u(x_0 + x)}{u'(x_0)u'(x_0 + x)} \tag{8.8b}$$

This technique for determining the integral scale requires simultaneous measurements at two points. Due to the difficulty of applying such a technique in engines, most efforts to determine length scales have first employed correlations to determine the *integral time scale*, τ_I. The integral time scale of turbulence is defined as a correlation between two velocities at a fixed point in space, but separated in time:

$$\tau_I = \int_0^\infty R_t \, dt \tag{8.9a}$$

where

$$R_t = \frac{1}{N_m - 1} \sum_{i=1}^{N_m} \frac{u(t_0)u(t_0 + t)}{u'(t_0)u'(t_0 + t)} \tag{8.9b}$$

and N_m is the number of measurements. Under conditions where the turbulence pattern is convected past the observation point without significant distortion and the turbulence is relatively weak, the integral length and time scales are related by

$$l_I = \bar{U}\tau_I \tag{8.10}$$

In flows where the large-scale structures are convected, τ_I is a measure of the time it takes a large eddy to pass a point. In flows without mean motion, the integral time scale is an indication of the lifetime of an eddy.[5, 8]

Superposed on this large-scale flow is a range of eddies of smaller and smaller size, fed by the continual breakdown of larger eddies. Since the smaller eddies respond more rapidly to changes in local flow pattern, they are more likely

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to be isotropic (have no preferred direction) than are the large eddies, and have a structure like that of other turbulent flows. The dissipation of turbulence energy takes place in the smallest structures. At this smallest scale of the turbulent motion, called the *Kolmogorov scale*, molecular viscosity acts to dissipate small-scale kinetic energy into heat. If ε is the energy dissipation rate per unit mass and ν the kinematic viscosity, Kolmogorov length and time scales are defined by

$$l_K = \left(\frac{\nu^3}{\varepsilon}\right)^{1/4} \qquad \tau_K = \left(\frac{\nu}{\varepsilon}\right)^{1/2} \tag{8.11}$$

The Kolmogorov length scale indicates the size of the smallest eddies. The Kolmogorov time scale characterizes the momentum-diffusion of these smallest structures.

A third scale is useful in characterizing a turbulent flow. It is called the *microscale* (or Taylor microscale). The micro length scale l_M is defined by relating the fluctuating strain rate of the turbulent flow field to the turbulence intensity:

$$\frac{\partial u}{\partial x} \approx \frac{u'}{l_M} \tag{8.12}$$

It can be determined from the curvature of the spatial correlation curve at the origin, as shown in Fig. 8-7.[5, 6] More commonly, the micro time scale τ_M is determined from the temporal autocorrelation function of Eq. (8.9):

$$\tau_M^2 = -\frac{2}{(\partial^2 R_t/\partial t^2)_{t_0}}$$

For turbulence which is homogeneous (has no spatial gradients) and is isotropic (has no preferred direction), the microscales l_M and τ_M are related by

$$l_M = \bar{U}\tau_M \tag{8.13}$$

These different scales are related as follows. The turbulent kinetic energy per unit mass in the large-scale eddies is proportional to u'^2. Large eddies lose a substantial fraction of this energy in one "turnover" time l_I/u'. In an equilibrium situation the rate of energy supply equals the rate of dissipation:

$$\varepsilon \approx \frac{u'^3}{l_I}$$

Thus,

$$\frac{l_K}{l_I} \approx \left(\frac{u' l_I}{\nu}\right)^{-3/4} = \mathrm{Re}_T^{-3/4} \tag{8.14}$$

where Re_T is the turbulent Reynolds number, $u' l_I/\nu$.

Within the restrictions of homogeneous and isotropic turbulence, an energy budget can be used to relate l_I and l_M:[6]

$$\varepsilon = \frac{Au'^3}{l_I} = \frac{15\nu u'^2}{l_M^2}$$

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where A is a constant of order 1. Thus,

$$\frac{l_M}{l_I} = \left(\frac{15}{A}\right)^{1/2} \mathrm{Re}_T^{-1/2} \tag{8.15}$$

These restrictions are not usually satisfied within the engine cylinder during intake. They are approximately satisfied at the end of compression.

8.2.2 Application to Engine Velocity Data

As has been explained above, it is necessary to analyze velocity data on an individual cycle basis as well as using ensemble-averaging techniques. The basic definitions for obtaining velocities which characterize the flow will now be developed. The ensemble-averaged velocity $\bar{U}_{\mathrm{EA}}$ has already been defined by Eq. (8.5). The ensemble-averaged fluctuation intensity $u'_{F,\,\mathrm{EA}}$ is given by

$$u'_{F,\,\mathrm{EA}}(\theta) = \left\{\frac{1}{N_c}\sum_{i=1}^{N_c}[u(\theta, i)]^2\right\}^{1/2} = \left\{\frac{1}{N_c}\sum_{i=1}^{N_c}[U(\theta, i)^2 - \bar{U}_{\mathrm{EA}}(\theta)^2]\right\}^{1/2} \tag{8.16}$$

It includes all fluctuations about the ensemble-averaged mean velocity.

Use of Eqs. (8.5) and (8.16) requires values for U and u at each specific crank angle under consideration. While some measurement techniques (e.g., hot-wire anemometry) provide this, the preferred velocity measurement method (laser doppler anemometry) provides an intermittent signal. With laser doppler anemometry (LDA), interference fringes are produced within the small volume created by the intersection of two laser beams within the flow field. When a small particle passes through this volume, it scatters light at a frequency proportional to the particle velocity. By seeding the flow with particles small enough to be carried without slip by the flow and collecting this scattered light, the flow velocity is determined.[9] A signal is only produced when a particle moves through the measurement volume; thus one collects data as velocity crank angle pairs. It is necessary, therefore, to perform the ensemble-averaging over a finite crank angle window $\Delta\theta$ around the specific crank angle of interest, $\bar{\theta}$. The ensemble-averaged velocity equation becomes

$$\bar{U}_{\mathrm{EA}}(\bar{\theta}) = \frac{1}{N_t}\sum_{i=1}^{N_c}\sum_{j=1}^{N_l} U_{i,j}\left(\bar{\theta} \pm \frac{\Delta\theta}{2}\right) \tag{8.17}$$

where N_i is the number of velocity measurements recorded in the window during the ith cycle, N_c is the number of cycles, and N_t is the total number of measurements.† The corresponding equation for the ensemble-averaged root mean square

† This need to ensemble-average over a finite crank angle window introduces an error called crank angle broadening, due to change in the mean velocity across the window. This error depends on the velocity gradient, and can be made negligibly small by suitable choice of window size.[9–11]

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velocity fluctuation is

$$u'_{F,\,\mathrm{EA}}(\bar{\theta}) = \left\{ \frac{1}{N_t} \sum_{i=1}^{N_c} \sum_{j=1}^{N_i} \left[u_{i,j}\left(\bar{\theta} \pm \frac{\Delta\theta}{2} \right) \right]^2 \right\}^{1/2} \tag{8.18}$$

where

$$u_{i,j} = U_{i,j} - \bar{U}_{\mathrm{EA}} \tag{8.19}$$

As has already been explained, this definition of fluctuation intensity [the ensemble-averaged rms velocity fluctuation, Eq. (8.18)] includes cyclic variations in the mean flow as well as the turbulent fluctuations about each cycle's mean flow.[7] It is necessary to determine the mean and fluctuating velocities on an individual-cycle basis to characterize the flow field more completely. The critical part of this process is defining the mean velocity at a specific crank angle (or within a small window centered about that crank angle) in each cycle. Several methods have been used to determine this individual-cycle mean velocity (e.g., moving window, low-pass filtering, data smoothing, conditional sampling; see Ref. 7 for a summary). A high data rate is required.

In this individual-cycle velocity analysis the individual-cycle time-averaged or mean velocity $\bar{U}(\bar{\theta}, i)$ is first determined.[7, 12] The ensemble average of this mean velocity

$$\bar{U}_{\mathrm{EA}}(\bar{\theta}) = \frac{1}{N_c} \sum_{i=1}^{N_c} \bar{U}\left(\bar{\theta} \pm \frac{\Delta\theta}{2}, i \right) \tag{8.20}$$

is identical to the ensemble-averaged value given by Eq. (8.17). The root mean square fluctuation in individual-cycle mean velocity can then be determined from

$$U_{\mathrm{RMS}}(\bar{\theta}) = \left\{ \frac{1}{N_c} \sum_{i=1}^{N_c} \left[\bar{U}\left(\bar{\theta} \pm \frac{\Delta\theta}{2}, i \right) - \bar{U}_{\mathrm{EA}}(\bar{\theta}) \right]^2 \right\}^{1/2} \tag{8.21}$$

This indicates the magnitude of the cyclic fluctuations in the mean motion. The instantaneous velocity fluctuation from the mean velocity, within a specified window $\Delta\theta$ at a particular crank angle $\bar{\theta}$, is obtained from Eq. (8.4). This instantaneous velocity fluctuation is ensemble-averaged, because it varies substantially cycle-by-cycle and because the amount of data is usually insufficient to give reliable individual-cycle results:

$$u'_{T,\,\mathrm{EA}}(\bar{\theta}) = \left\{ \frac{1}{N_c} \sum_{i=1}^{N_c} \left[U\left(\bar{\theta} \pm \frac{\Delta\theta}{2}, i \right) - \bar{U}\left(\bar{\theta} \pm \frac{\Delta\theta}{2}, i \right) \right]^2 \right\}^{1/2} \tag{8.22}$$

This quantity is the ensemble-averaged turbulence intensity.

Several different techniques have been used to measure gas velocities within the engine cylinder (see Refs. 13 and 14 for brief reviews and references). The technique which provides most complete and accurate data is laser doppler anemometry.[9] Sample results obtained with this technique will now be reviewed to

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illustrate the major features of the in-cylinder gas motion. The available results must be interpreted with caution since they have been obtained in special engines where the geometry and flow have been modified to make the experiments and their interpretation easier. Also, the flow within the cylinder is three dimensional in nature. It takes measurements at many points within the flow field and the use of a flow visualization technique to characterize the flow adequately.

Figure 8-8 shows ensemble-averaged velocities throughout the engine cycle at two measurement locations in a special L-head engine designed to generate a swirling flow within the cylinder. The engine was motored at 300 rev/min, giving a mean piston speed of 0.76 m/s. Figure 8-8*b* shows the mean velocity in the path of the swirling intake flow within the clearance volume, in the swirl direction. High velocities are generated during the intake process, rising to a maximum and then decreasing in response to the piston motion (see Fig. 2-2). During the compression stroke, the velocity continues to decrease but at a much slower rate. This is a motored engine cycle. A comparison of intake and compression velocities with an equivalent firing cycle showed close agreement.[15] The expansion and

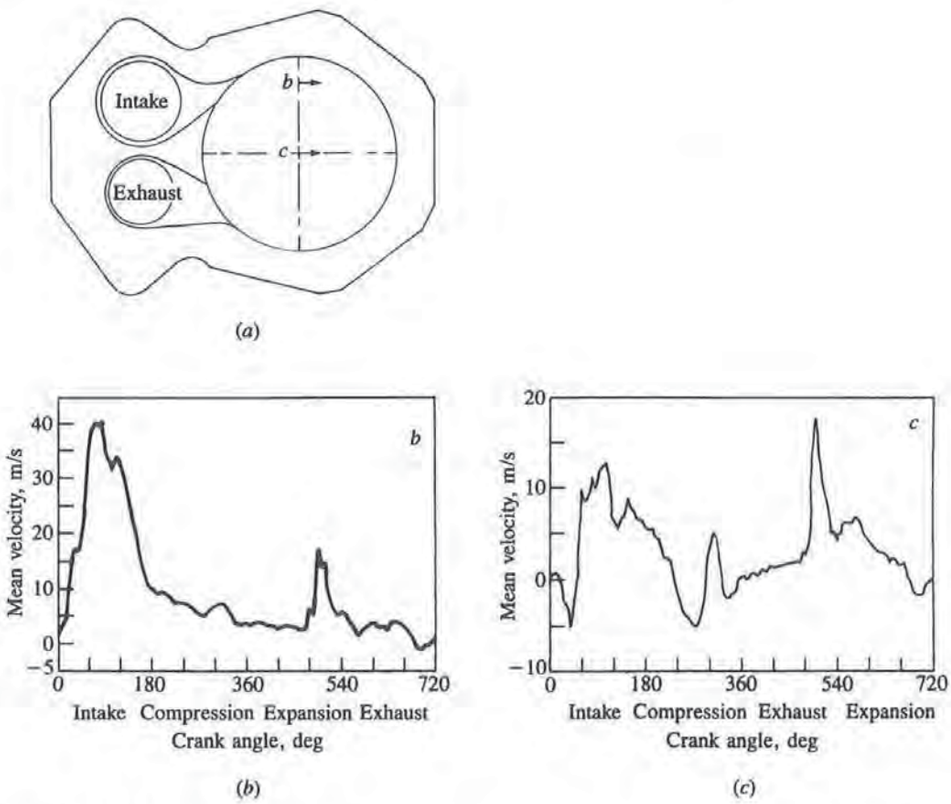


FIGURE 8-8
Ensemble-averaged velocities throughout the engine cycle in motored four-stroke L-head engine: 300 rev/min, mean piston speed 0.76 m/s. (*a*) Engine schematic showing measurement locations and velocity directions; (*b*) velocity at *b* in intake flow path; (*c*) velocity at *c* on cylinder axis.[11]

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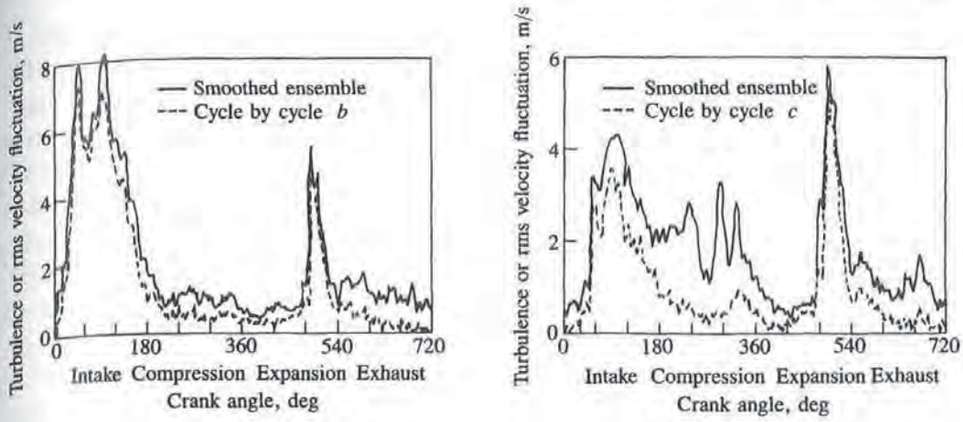


FIGURE 8-9
Ensemble-averaged rms velocity fluctuation and ensemble-averaged individual-cycle turbulence intensity as a function of crank angle: (*a*) at location *b* in Fig. 8-8*a*; (*b*) at location *c* in Fig. 8-8*a*.[11]

exhaust stroke velocities are not typical of firing engine behavior, however.† Figure 8-8*c* shows the mean velocity in the clearance volume in the same direction but on the cylinder axis. At this location, positive and negative flow velocities were measured. Since this location is out of the path of the intake generated flow, velocities during the intake stroke are much lower. The nonhomogeneous character of this particular ensemble-mean flow is evident.

Figure 8-9 shows the ensemble-averaged rms velocity fluctuation (which includes contributions from cycle-by-cycle variations in the mean flow and turbulence) and the ensemble-averaged individual-cycle turbulence intensity at these same two locations and directions. The difference between the two curves in each graph is an indication of the cycle-by-cycle variation in the mean flow [see Eq. (8.7)]. During the intake process, within the directed intake flow pattern, the cycle-by-cycle variation in the mean flow is small in comparison to the high turbulence levels created by the intake flow. Outside this directed flow region, again during intake, this cycle-by-cycle contribution is more significant relative to the turbulence. During compression, the cycle-by-cycle mean flow variation is comparable in magnitude to the ensemble-averaged turbulence intensity. It is therefore highly significant.

Two important questions regarding the turbulence in the engine cylinder are whether it is homogeneous (i.e., uniform) and whether it is isotropic (i.e., independent of direction). The data already presented in Figs. 8-8 and 8-9 show that during intake the flow is far from homogeneous. Nor is it isotropic.[11]

† The increase in velocity when the exhaust valve opens is due to the flow of gas *into* the cylinder because, due primarily to heat losses, the cylinder pressure is then below 1 atm.

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However, it is the character of the turbulence at the end of the compression process that is most important: that is what controls the fuel-air mixing and burning rates. Only limited data are available which relate to these questions. With open disc-shaped combustion chambers, measurements at different locations in the clearance volume around TC at the end of compression show the turbulence intensity to be relatively homogeneous. In the absence of an intake generated swirling flow, the turbulence intensity was also essentially isotropic near TC.[16] These specific results support the more general conclusion that the inlet boundary conditions play the dominant role in the generation of the mean flow and turbulence fields during the intake stroke. However, in the absence of swirl, this intake generated flow structure has almost disappeared by the time the compression process commences. The turbulence levels follow this trend in the mean flow, with the rapid decay process lasting until intake valve closing. Later in the compression process the turbulence becomes essentially homogeneous.[17]

When a swirling flow is generated during intake, an almost solid-body rotating flow develops which remains stable for much longer than the inlet jet generated rotating flows illustrated in Fig. 8-3. With simple disc-shaped chambers, the turbulence still appears to become almost homogeneous at the end of compression. With swirl and bowl-in-piston geometry chambers, however, the flow is more complex (see Sec. 8.3).

The flow through the intake valve or port is responsible for many features of the in-cylinder motion. The flow velocity through the valve is proportional to the piston speed [see Eq. (6.10) for pseudo valve flow velocity, and Eq. 2.10)]. It would be expected therefore that in-cylinder flow velocities at different engine speeds would scale with mean piston speed [Eq. (2.11)]. Figure 8-10 shows ensemble-averaged mean and rms velocity fluctuations, normalized by the mean piston speed through the cycle at three different engine speeds. The measurement location is in the path of the intake generated swirling flow (point *b* in Fig. 8-8*a*). All the curves have approximately the same shape and magnitude, indicating the

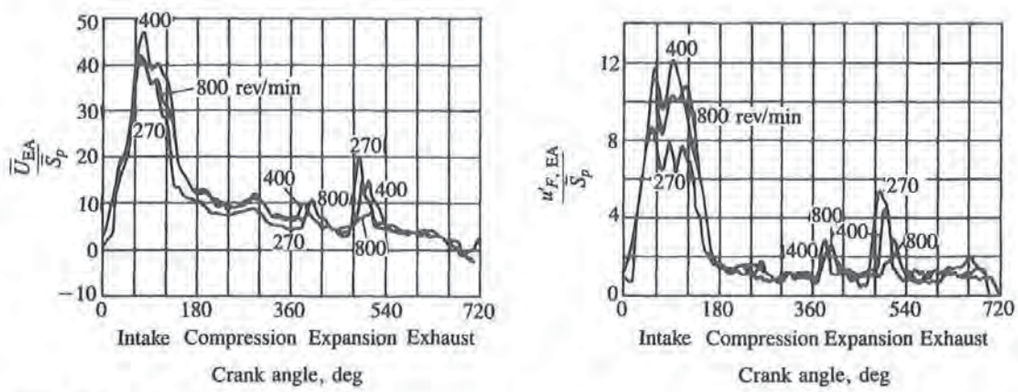


FIGURE 8-10
Ensemble-averaged mean and rms velocity fluctuations, normalized by mean piston speed, throughout the engine cycle for three engine speeds. Location *b* in Fig. 8-8*a*.[11]

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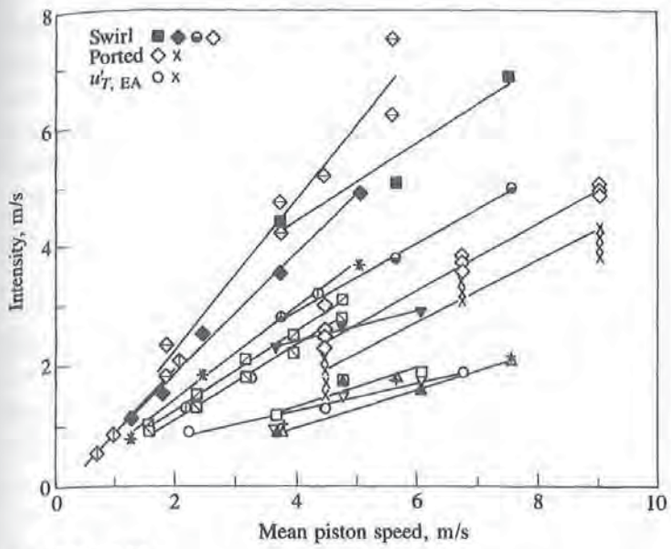


FIGURE 8-11
Individual-cycle turbulence intensity $u'_{T,\,EA}$ (OX) and ensemble-averaged rms fluctuation velocity (remaining symbols) at TC at the end of compression, for a number of different flow configurations and chamber geometries as a function of mean piston speed.[16] Two data sets for two-stroke ported engines. Four data sets with intake generated swirl.

appropriateness of this velocity scaling.† Other results support this conclusion, though in the absence of an ordered mean motion such as swirl when the ensemble-averaged mean velocities at the end of compression are low, this scaling for the mean velocity does not always hold.[16] Figure 8-11 shows a compilation of ensemble-averaged rms fluctuation velocity or ensemble-averaged individual-cycle turbulence intensity results at TC at the end of compression, from 13 different flow configurations and combustion chamber geometries. Two of these sets of results are from two-stroke cycle ported configurations. The measured fluctuating velocities or turbulence intensities are plotted against mean piston speed. The linear relationship holds well. There is a substantial variation in the proportionality constant, in part because in most of these studies (identified in the figure) the ensemble-averaged rms fluctuation velocity was the quantity measured. Since this includes the cycle-by-cycle fluctuation in the mean velocity, it is larger (by up to a factor of 2) than the average turbulence intensity $u'_{T,\,EA}$.

A consensus conclusion is emerging from these studies that the turbulence intensity at top-center, with open combustion chambers in the absence of swirl, has a maximum value equal to about half the mean piston speed:[16, 18]

$$u'_{T,\,EA}(TC) \approx 0.5\bar{S}_p \tag{8.23}$$

† Note that because of the valve and combustion chamber of this particular engine, the ratio of $\bar{U}$ to $\bar{S}_p$ is higher than is typical of normal engine geometries.

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The available data show that the turbulence intensity at TC with swirl is usually higher than without swirl[16] (see the four data sets with swirl in Fig. 8-11). Some data, however, indicate that the rms fluctuation intensity with swirl may be lower.[18] The ensemble-averaged cyclic variation in individual-cycle mean velocity at the end of compression also scales with mean piston speed. This quantity can be comparable in magnitude to the turbulence intensity. It usually decreases when a swirling flow is generated within the cylinder during the intake process.[11, 16]

During the compression stroke, and also during combustion while the cylinder pressure continues to rise, the unburned mixture is compressed. Turbulent flow properties are changed significantly by the large and rapidly imposed distortions that result from this compression. Such distortions, in the absence of dissipation, would conserve the angular momentum of the flow: rapid compression would lead to an increase in vorticity and turbulence intensity. There is evidence that, with certain types of in-cylinder flow pattern, an increase in turbulence intensity resulting from piston motion and combustion does occur toward the end of the compression process. The compression of large-scale rotating flows can cause this increase due to the increasing angular velocity required to conserve angular momentum resulting in a growth in turbulence generation by shear.[19]

Limited results are available which characterize the turbulence time and length scales in automobile engine flows. During the intake process, the integral length scale is of the order of the intake jet diameter, which is of the order of the valve lift ($\lesssim$10 mm in automobile-size engines). During compression the flow relaxes to the shape of the combusion chamber. The integral time scale at the end of compression decreases with increasing engine speed. It is of order 1 ms at engine speeds of about 1000 rev/min. The integral length scale at the end of compression is believed to scale with the clearance height and varies little with engine speed. It decreases as the piston approaches TC to about 2 mm (0.2 $\times$ clearance height). The micro time scale at the end of compression is of order 0.1 ms at 1000 rev/min, and decreases as engine speed increases (again in automobile-size engine cylinders). Micro length scales are of order 1 mm at the end of compression and vary little with engine speed. Kolmogorov length scales at the end of compression are of order 10^{-2} mm.[8, 20, 21]

8.3 SWIRL

Swirl is usually defined as organized rotation of the charge about the cylinder axis. Swirl is created by bringing the intake flow into the cylinder with an initial angular momentum. While some decay in swirl due to friction occurs during the engine cycle, intake generated swirl usually persists through the compression, combustion, and expansion processes. In engine designs with bowl-in-piston combustion chambers, the rotational motion set up during intake is substantially modified during compression. Swirl is used in diesels and some stratified-charge engine concepts to promote more rapid mixing between the inducted air charge

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and the injected fuel. Swirl is also used to speed up the combustion process in spark-ignition engines. In two-stroke engines it is used to improve scavenging. In some designs of prechamber engines, organized rotation about the prechamber axis is also called swirl. In prechamber engines where swirl within the precombustion chamber is important, the flow into the prechamber during the compression process creates the rotating flow. Prechamber flows are discussed in Sec. 8.5.

8.3.1 Swirl Measurement

The nature of the swirling flow in an actual operating engine is extremely difficult to determine. Accordingly, steady flow tests are often used to characterize the swirl. Air is blown steadily through the inlet port and valve assembly in the cylinder head into an appropriately located equivalent of the cylinder. A common technique for characterizing the swirl within the cylinder has been to use a light paddle wheel, pivoted on the cylinder centerline (with low friction bearings), mounted between 1 and 1.5 bore diameters down the cylinder. The paddle wheel diameter is close to the cylinder bore. The rotation rate of the paddle wheel is used as a measure of the air swirl. Since this rotation rate depends on the location of the wheel and its design, and the details of the swirling flow, this technique is being superseded by the impulse swirl meter shown in Fig. 8-12. A honeycomb flow straightener replaces the paddle wheel: it measures the total torque exerted by the swirling flow. This torque equals the flux of angular

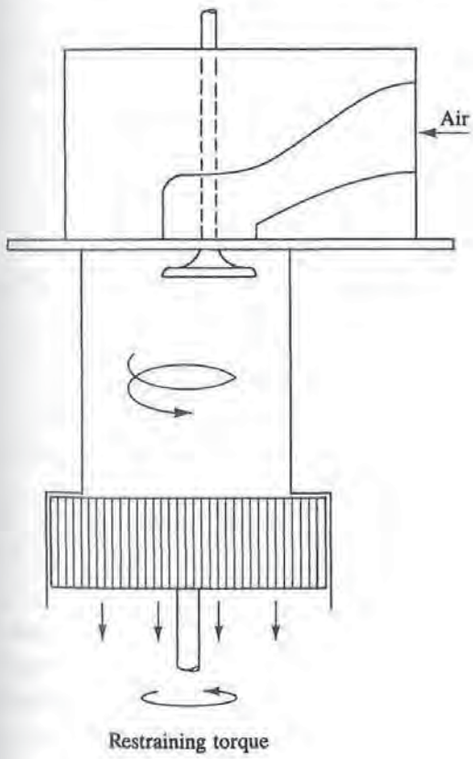


FIGURE 8-12
Schematic of steady-flow impulse torque swirl meter.[22]

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momentum through the plane coinciding with the flow-straightener upstream face.

For each of these approaches, a *swirl coefficient* is defined which essentially compares the flow's angular momentum with its axial momentum. For the paddle wheel, the swirl coefficient C_s is defined by

$$C_s = \frac{\omega_p B}{v_0} \tag{8.24}$$

where ω_p is the paddle wheel angular velocity ($=2\pi N_p$, where N_p is the rotational speed) and the bore B has been used as the characteristic dimension. The characteristic velocity, v_0, is derived from the pressure drop across the valve using an incompressible flow equation:

$$v_0 = \left[\frac{2(p_0 - p_c)}{\rho}\right]^{1/2} \tag{8.25}$$

or a compressible flow equation:

$$v_0 = \left\{\frac{2\gamma}{(\gamma - 1)}\frac{p_0}{\rho_0}\left[1 - \left(\frac{p_c}{p_0}\right)^{(\gamma-1)/\gamma}\right]\right\}^{1/2} \tag{8.26}$$

where the subscripts 0 and c refer to upstream stagnation and cylinder values, respectively. The difference between Eqs. (8.25) and (8.26) is usually small. With the impulse torque meter, characteristic velocity and length scales must also be introduced. Several swirl parameters have been defined.[22, 23] The simplest is

$$C_s = \frac{8T}{\dot{m} v_0 B} \tag{8.27}$$

where T is the torque and $\dot{m}$ the air mass flow rate. The velocity v_0, defined by Eq. (8.25) or Eq. (8.26), and the bore have again been used to obtain a dimensionless coefficient. Note that for solid-body rotation of the fluid within the cylinder at the paddle wheel speed ω_p, Eqs. (8.24) and (8.27) give identical swirl coefficients. In practice, because the swirling flow is not solid-body rotation and because the paddle wheel usually lags the flow due to slip, the impulse torque meter gives higher swirl coefficients.[23] When swirl measurements are made in an operating engine, a *swirl ratio* is normally used to define the swirl. It is defined as the angular velocity of a solid-body rotating flow ω_s, which has equal angular momentum to the actual flow, divided by the crankshaft angular rotational speed:

$$R_s = \frac{\omega_s}{2\pi N} \tag{8.28}$$

During the induction stroke in an engine the flow and the valve open area, and consequently the angular momentum flux into the cylinder, vary with crank angle. Whereas in rig tests the flow and valve open area are fixed and the angular momentum passes down the cylinder continuously, in the engine intake process

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the momentum produced under corresponding conditions of flow and valve lift remains in the cylinder. Steady-state impulse torque-meter flow rig data can be used to estimate engine swirl in the following manner.[23] Assuming that the port and valve retain the same characteristics under the transient conditions of the engine as on the steady-flow rig, the equivalent solid-body angular velocity ω_s at the end of the intake process is given by

$$\omega_s = \frac{8}{B^2}\left(\int_{\theta_1}^{\theta_2} T\, d\theta\right)\Big/\left(\int_{\theta_1}^{\theta_2} \dot{m}\, d\theta\right)$$

where θ_1 and θ_2 are crank angles at the start and end of the intake process and the torque T and mass flow rate $\dot{m}$ are evaluated at the valve lift corresponding to the local crank angle. Using Eq. (8.27) for T, Eq. (6.11) for $\dot{m}$, assuming v_0 and ρ are constant throughout the intake process, and introducing volumetric efficiency η_v based on intake manifold conditions via Eq. (2.27), it can be shown that

$$R_s = \frac{\omega_s}{2\pi N} = \pi \eta_v BL\left[\int_{\theta_1}^{\theta_2} (A_v C_D) C_s\, d\theta\right]\Big/\left[\int_{\theta_1}^{\theta_2} (A_v C_D)\, d\theta\right]^2 \qquad (8.29)$$

where $A_v C_D$ is the effective valve open area at each crank angle. Note that the crank angle in Eq. (8.29) should be in radians. Except for its (weak) dependence on η_v, Eq. (8.29) gives R_s independent of operating conditions directly from rig test results and engine geometry.

The relationship between steady-flow rig tests (which are extensively used because of their simplicity) and actual engine swirl patterns is not fully understood. Steady-flow tests adequately describe the swirl generating characteristics of the intake port and valve (at fixed valve lift) and are used extensively for this purpose. However, the swirling flow set up in the cylinder during intake can change significantly during compression.

8.3.2 Swirl Generation during Induction

Two general approaches are used to create swirl during the induction process. In one, the flow is discharged into the cylinder tangentially toward the cylinder wall, where it is deflected sideways and downward in a swirling motion. In the other, the swirl is largely generated within the inlet port: the flow is forced to rotate about the valve axis *before* it enters the cylinder. The former type of motion is achieved by forcing the flow distribution around the circumference of the inlet valve to be nonuniform, so that the inlet flow has a substantial net angular momentum about the cylinder axis. The directed port and deflector wall port in Fig. 8-13 are two common ways of achieving this result. The directed port brings the flow toward the valve opening in the desired tangential direction. Its passage is straight, which due to other cylinder head requirements restricts the flow area and results in a relatively low discharge coefficient. The deflector wall port uses the port inner side wall to force the flow preferentially through the outer periphery of the valve opening, in a tangential direction. Since only one wall is used to obtain a directional effect, the port areas are less restrictive.

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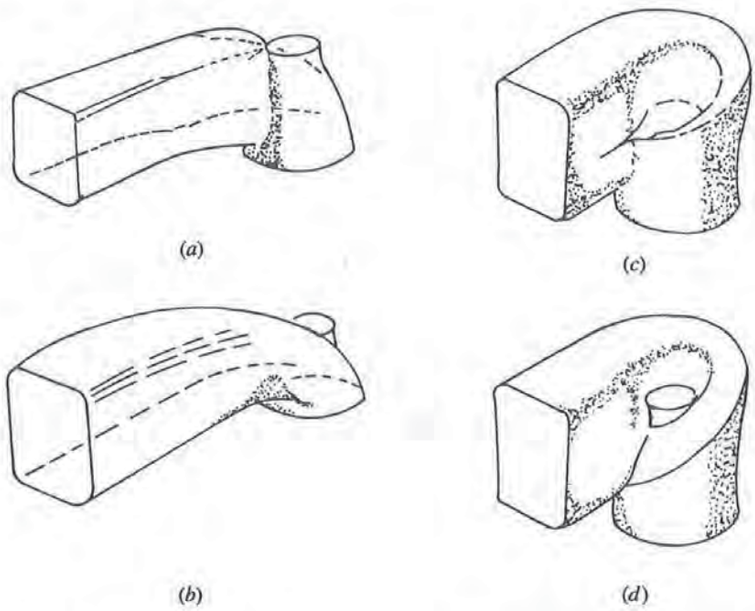


FIGURE 8-13
Different types of swirl-generating inlet ports: (*a*) deflector wall; (*b*) directed; (*c*) shallow ramp helical; (*d*) steep ramp helical.[24]

Flow rotation about the cylinder axis can also be generated by masking off or shrouding part of the peripheral inlet valve open area, as shown in Fig. 8-14. Use is often made of a mask or shroud on the valve in research engines because changes can readily be made. In production engines, the added cost and weight, problems of distortion, the need to prevent valve rotation, and reduced volumetric efficiency make masking the valve an unattractive approach. The more practical alternative of building a mask on the cylinder head around part of the inlet valve periphery is used in production spark-ignition engines to generate swirl. It can easily be incorporated in the cylinder head casting process.

The second broad approach is to generate swirl within the port, about the valve axis, prior to the flow entering the cylinder. Two examples of such *helical ports* are shown in Fig. 8-13. Usually, with helical ports, a higher flow discharge coefficient at equivalent levels of swirl is obtained, since the whole periphery of the valve open area can be fully utilized. A higher volumetric efficiency results. Also, helical ports are less sensitive to position displacements, such as can occur in casting, since the swirl generated depends mainly on the port geometry above the valve and not the position of the port relative to the cylinder axis.

Figure 8-15 compares steady-state swirl-rig measurements of examples of the ports in Fig. 8-13. The rig swirl number increases with increasing valve lift, reflecting the increasing impact of the port shape and decreasing impact of the flow restriction between the valve head and seat. Helical ports normally impart more angular momentum at medium lifts than do directed ports.[23, 25] The swirl

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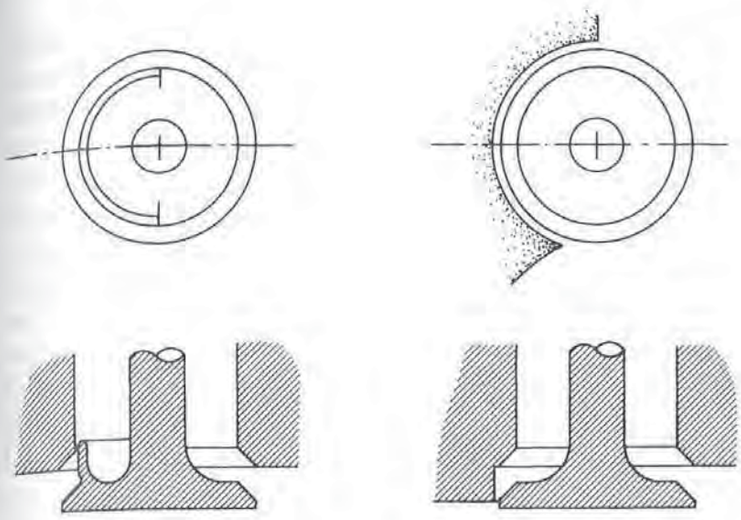


FIGURE 8-14
Shrouded inlet valve and masked cylinder head approaches for producing net in-cylinder angular momentum.

ratios for these ports calculated from this rig data using Eqs. (8.27) and (8.29) are: 2.5 for the directed port, 2.9 for the shallow ramp helical, and 2.6 for the steep ramp helical. Vane swirl-meter swirl ratios were about 30 percent less. These impulse-swirl-meter derived engine swirl ratios are within about 20 percent of the solid-body rotation rate which has equal angular momentum to that of the cylinder charge determined from tangential velocity measurements made within the cylinder of an operating engine with the same port, at the end of the induction process.[23]

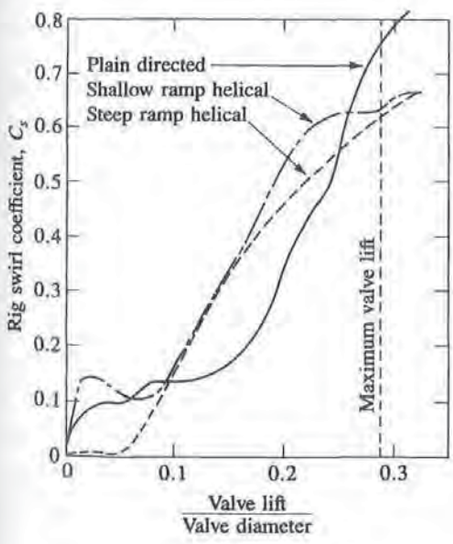


FIGURE 8-15
Steady-state torque meter swirl measurements of directed, shallow ramp, and steep ramp helical ports as a function of inlet valve lift/diameter ratio.[23]

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Directed and deflector wall ports, and masked valve or head designs produce a tangential flow into the cylinder by increasing the flow resistance through that part of the valve open area where flow is not desired. A highly nonuniform flow through the valve periphery results and the flow into the cylinder has a substantial v_θ velocity component in the same direction about the cylinder axis. In contrast, helical ports produce the swirl in the port upstream of the valve, and the velocity components v_r, and v_z through the valve opening, and v_θ *about the valve axis* are approximately uniform around the valve open area. Figure 8-16 shows velocity data measured at the valve exit plane in steady-flow rig tests with examples of these two types of port. The valve and cylinder wall locations are shown. In Fig. 8-16*a*, the deflector wall of the tangentially oriented port effectively prevents any significant flow around half the valve periphery. In contrast, in Fig. 8-16*b* with the helical port, the air flows into the cylinder around

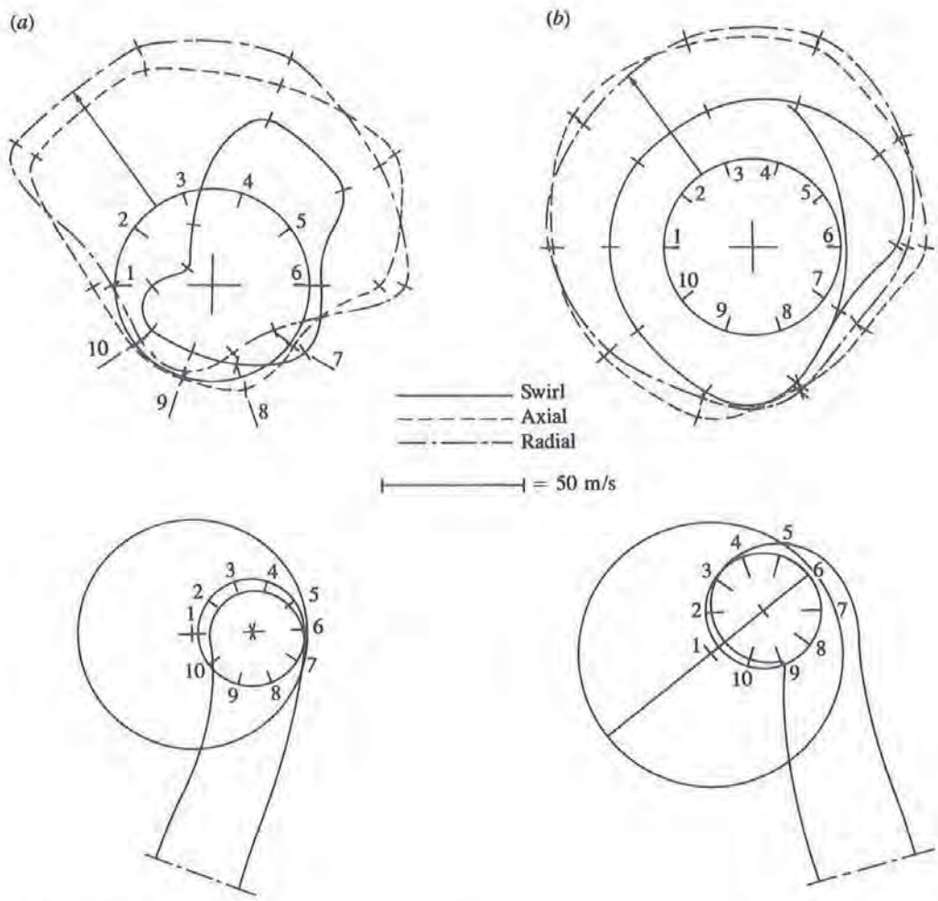


FIGURE 8-16
Swirl, axial, and radial velocities measured 2 mm from cylinder head around the valve circumference for (*a*) tangential deflector-wall port and (*b*) helical port; magnitude of velocity is given by the distance along a radial line (from valve axis), from valve outline to the respective curve scaled by the reference length (examples of radial velocity indicated by two arrows); valve lift = 12.8 mm.[26, 27]

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the full valve open area. The radial and axial velocities are essentially uniform around the valve periphery. The swirl velocity about the valve axis (anticlockwise when viewed from above) for this helical port is relatively uniform and is about half the magnitude of the radial and axial velocities.

The swirling air flow within the cylinder of an operating engine is not uniform. The velocities generated at the valve at each point in the induction process depend on the valve open area and piston velocity. The velocities are highest during the first half of the intake process as indicated in Fig. 6-15. Thus, the swirl velocities generated during this portion of the induction stroke are higher than the swirl generated during the latter half of the stroke: there is swirl stratification. Also, the flow pattern close to the cylinder head during induction is comparatively disorganized, and not usually close to a solid-body rotation. It consists of a system of vortices, created by the high-velocity tangential or spiraling intake jet. Further down the cylinder, the flow pattern is closer to solid-body rotation with the swirl velocity increasing with increasing radius.[23, 24] This more ordered flow directly above the piston produces higher swirl velocities in that region of the cylinder. As the piston velocity decreases during intake, the swirl pattern redistributes, with swirl speeds close to the piston decreasing and swirl speeds in the center of the cylinder increasing.[27] Note that the axis of rotation of the in-cylinder gases may not exactly coincide with the cylinder axis.

8.3.3 Swirl Modification within the Cylinder

The angular momentum of the air which enters the cylinder at each crank angle during induction decays throughout the rest of the intake process and during the compression process due to friction at the walls and turbulent dissipation within the fluid. Typically one-quarter to one-third of the initial moment of momentum about the cylinder axis will be lost by top-center at the end of compression. However, swirl velocities in the charge can be substantially increased during compression by suitable design of the combustion chamber. In many designs of direct-injection diesel, air swirl is used to obtain much more rapid mixing between the fuel injected into the cylinder and the air than would occur in the absence of swirl. The tangential velocity of the swirling air flow set up inside the cylinder during induction is substantially increased by forcing most of the air into a compact bowl-in-piston combustion chamber, usually centered on the cylinder axis, as the piston approaches its top-center position. Neglecting the effects of friction, angular momentum is conserved, and as the moment of inertia of the air is decreased its angular velocity must increase.

However, the total angular momentum of the charge within the cylinder does decay due to friction at the chamber walls. The angular momentum of the cylinder charge Γ_c changes with time according to the moment of momentum conservation equation:

$$\frac{d\Gamma_c}{dt} = J_i - T_f \tag{8.30}$$

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where J_i is the flux of angular momentum into the cylinder and T_f is the torque due to wall friction. At each point in the intake process J_i is given by

$$J_i = \int_{A_v} \rho r v_\theta \mathbf{v} \cdot d\mathbf{A}_v \tag{8.31}$$

where $d\mathbf{A}_v$ is an element of the valve open area, as defined in Fig. 8-17. While the angular momentum entering the cylinder during the intake process is

$$\Gamma_{c,i} = \int_{t_{ivo}}^{t_{ivc}} \int_{A_v} \rho r v_\theta \cdot d\mathbf{A}_v \, dt$$

the actual angular momentum within the cylinder at the end of induction will be less, due to wall friction during the intake process. Friction continues through the compression process so the total charge angular momentum at the end of compression is further reduced.

There is friction on the cylinder wall, cylinder head, and piston crown (including any combustion chamber within the crown). This friction can be estimated with sufficient accuracy using friction formulas developed for flow over a flat plate, with suitable definition of characteristic length and velocity scales. Friction on the cylinder wall can be estimated from the wall shear stress:

$$\tau = \tfrac{1}{2}\rho\left(\frac{\omega_s B}{2}\right)^2 C_F \tag{8.32}$$

where ω_s is the equivalent solid-body swirl. The friction factor C_F is given by the flat plate formula:

$$C_F = 0.037\lambda(\mathrm{Re}_B)^{-0.2} \tag{8.33}$$

where λ is an empirical constant introduced to allow for differences between the flat plate and cylinder wall ($\lambda \approx 1.5$)[28] and Re_B is the equivalent of the flat plate Reynolds number [$\mathrm{Re}_B = \rho(B\omega_s/2)(\pi B)/\mu$]. Friction on the cylindrical walls of a piston cup or bowl can be obtained from the above expressions with D_B, the bowl diameter, replacing the bore.

Friction on the cylinder head, piston crown, and piston bowl floor can be estimated from expressions similar to Eqs. (8.32) and (8.33). However, since the

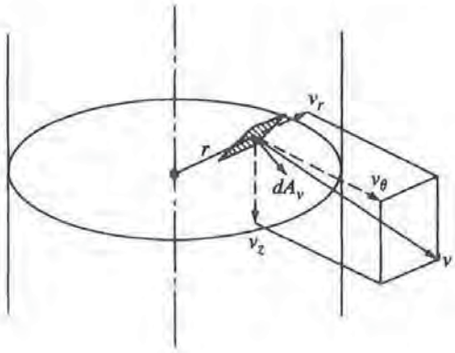


FIGURE 8-17
Definition of symbols in equation for angular momentum flux into the cylinder [Eq. (8.31)].

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tangential velocity v_θ at the wall varies with radius, the shear stress should be evaluated at each radius and integrated over the surface: e.g.,[29]

$$\tau(r) = C_1 \tfrac{1}{2}\rho[v_\theta(r)]^2 \, \mathrm{Re}^{-0.2} \tag{8.34}$$

with

$$\mathrm{Re} = \frac{\rho v_\theta(r) r}{\mu}$$

where C_1 is an empirical constant (≈ 0.055). An alternative approximate approach is to evaluate these components of the wall shear stress at the mean radius.[28]

Next, consider the effects on swirl of radially inward displacement of the air charge during compression. The most common example of this phenomenon occurs with the bowl-in-piston combustion chamber design of medium- and high-speed direct-injection diesels (see Sec. 10.2.1). However, in spark-ignition engines where swirl is used to increase the burning rate, the shape of the combustion chamber close to top-center can also force radially inward motion of the charge. For a given swirling in-cylinder flow at the end of induction and neglecting the effects of friction, as the moment of inertia of the air about the cylinder axis is decreased the air's angular velocity must increase to conserve angular momentum. For example, for solid-body rotation of the cylinder air charge of mass m_c, the initial angular momentum $\Gamma_{c,i}$ and solid-body rotation $\omega_{s,i}$ are related at bottom-center by

$$\Gamma_{c,i} = I_c \, \omega_{s,i}$$

where I_c is the moment of inertia of the charge about the cylinder axis. For a disc-shaped combustion chamber, $I_c = m_c B^2/8$ and is constant. For a bowl-in-piston combustion chamber,

$$I_c = \frac{m_c B^2}{8} \frac{[(z/h_B) + (D_B/B)^4]}{[(z/h_B) + (D_B/B)^2]} \tag{8.35}$$

where D_B and h_B are the diameter and depth of the bowl, respectively, and z is the distance of the piston crown from the cylinder head. At TC crank position, $z \approx 0$ and $I_c \approx m_c D_B^2/8$. At the end of induction, $I_c \approx m_c B^2/8$. Thus, in the absence of friction ω_s would increase by $(B/D_B)^2$, usually a factor of about 4.

In an operating engine with this bowl-in-piston chamber design, the observed increase in swirl in the bowl is less; it is usually about a factor of 2 to 3.[23, 25] This is because of wall friction, dissipation in the fluid due to turbulence and velocity gradients, and the fact that a fraction of the fluid remains in the clearance height above the piston crown. The loss in angular momentum due to these effects will vary with geometric details, initial swirl flow pattern, and engine speed.

Swirl velocity distributions in the cylinder at the end of induction show the tangential velocity increasing with radius, except close to the cylinder wall where friction causes the velocity to decrease. While the velocity distribution is not that of a solid-body rotation, depending on port design and operating conditions it is

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often close to solid-body rotation.[23, 25] Departures from the solid-body velocity distribution are greater at higher engine speeds, suggesting that the flow pattern in the cylinder at this point in the cycle is still developing with time.[23, 30] In the absence of radially inward gas displacement during compression, the flow pattern continues to develop toward a solid-body distribution throughout the compression stroke.[25] Swirl ratios of 3 to 5 at top-center can be achieved with the ports shown in Fig. 8-13, with flat-topped pistons (i.e., in the absence of any swirl amplification during compression).[23, 25]

With combustion chambers where the chamber radius is less than the cylinder bore, such as the bowl in piston, the tangential velocity distribution with radius will change during compression. Even if the solid-body rotation assumption is reasonable at the end of induction, the profile will distort as gas moves into the piston bowl. Neglecting the effects of friction, the angular momentum of each fluid element will remain constant as it moves radially inward. Thus the increase in tangential velocity of each fluid element as it moves radially inward is proportional to the change in the reciprocal of its radius. Measurements of the swirl velocity distribution within the cylinder of bowl-in-piston engine designs support this description. The rate of displacement of gas into the bowl depends on the bowl volume V_B, cylinder volume V, and piston speed S_p, at that particular piston position:

$$\frac{dm_B}{dt} = \frac{m_c}{L}\left(\frac{V_B}{V}\right)\left(\frac{V_d}{V}\right)S_p$$

The gas velocity into the bowl will therefore increase rapidly toward the end of the compression stroke and reach a maximum just before TC (see Sec. 8.4 where this radial "squish" motion is discussed more fully). Thus, there is a rapid increase in v_θ in the bowl as the crank angle approaches TC. The lower layers of the bowl rotate slower than the upper layers because that gas entered the bowl earlier in the compression process.[23, 25]

Velocity measurements illustrating the development of this radial distribution in tangential velocity are shown in Fig. 8-18. These measurements were made by analysing the motion of burning carbon particles in the cylinder of an operating diesel engine from movies of the combustion process. The figure shows the engine geometry and the data compared with a model based on gas displacement and conservation of angular momentum in each element of the charge as it is displaced inward. Different swirl velocity profiles exist within and outside the bowl as the piston approaches TC. Swirl velocities within the bowl increase as TC is approached, roughly as predicted by the ideal model. Outside the bowl, the swirl velocity decreases with increasing radius due to the combined effects of friction and inward gas displacement as the clearance height decreases.

Swirl ratios in bowl-in-piston engine designs of up to about 15 can be achieved with $D_B \approx 0.5B$, at top-center. Amplification factors relative to flat-topped piston swirl are typically about 2.5 to 3, some 30 percent lower than the ideal factor of $(B/D_B)^2$ given by Eq. (8.35) as $z \to 0$. This difference is due to the mass remaining within the clearance height which does not enter the bowl, and

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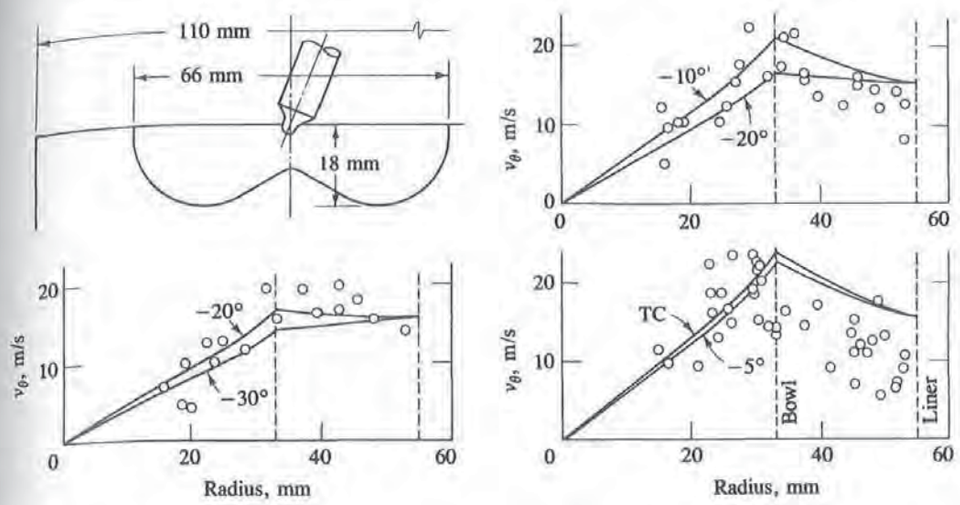


FIGURE 8-18
Velocity measurements as a function of radius across the combustion chamber of a firing, bowl-in-piston, direct-injection diesel engine. Schematic shows the chamber geometry. Solid lines are calculations based on the assumption of constant angular momentum for fluid elements as they move radially inward.[31]

the effects of wall friction (enhanced by the higher gas velocities in the bowl). Sometimes the bowl axis is offset from the cylinder axis and some additional loss in swirl amplification results.[25]

The effect of swirl generation during induction on velocity fluctuations in the combustion chamber at the end of compression has been examined.[32] The turbulence intensity with swirl was higher than without swirl (with the same chamber geometry). Integral scales of the turbulence were smaller with swirl than without. Cyclic fluctuations in the mean velocity are, apparently, reduced by swirl. Also, some studies show that the ensemble-averaged fluctuation intensity goes down when swirl is introduced.[18] There is evidence that swirl makes the turbulence intensity more homogeneous.[30]

8.4 SQUISH

Squish is the name given to the radially inward or transverse gas motion that occurs toward the end of the compression stroke when a portion of the piston face and cylinder head approach each other closely. Figure 8-19 shows how gas is thereby displaced into the combustion chamber. Figure 8-19*a* shows a typical wedge-shaped SI engine combustion chamber and Fig. 8-19*b* shows a bowl-in-piston diesel combustion chamber. The amount of squish is often defined by the *percentage squish area:* i.e., the percentage of the piston area, $\pi B^2/4$, which closely approaches the cylinder head (the shaded areas in Fig. 8-19). Squish-generated gas motion results from using a compact combustion chamber geometry.

A theoretical squish velocity can be calculated from the instantaneous dis-

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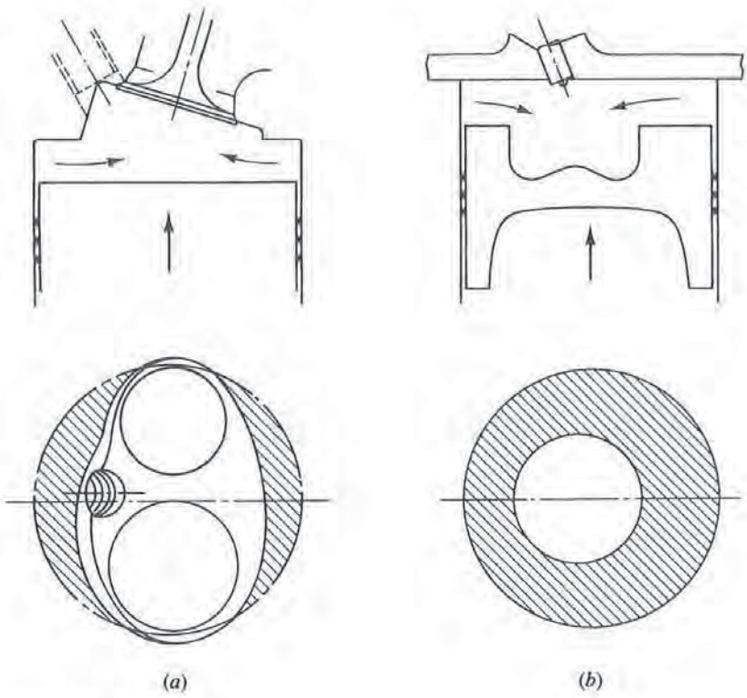


FIGURE 8-19
Schematics of how piston motion generates squish: (*a*) wedge-shaped SI engine combustion chamber; (*b*) bowl-in-piston direct-injection diesel combustion chamber.

placement of gas across the inner edge of the squish region (across the dashed lines in the drawings in Fig. 8-20*a* and *b*), required to satisfy mass conservation. Ignoring the effects of gas dynamics (nonuniform pressure), friction, leakage past the piston rings, and heat transfer, expressions for the squish velocity are:

1. *Bowl-in-piston chamber* (Fig. 8-20*a*):[33]

$$\frac{v_{sq}}{S_p} = \frac{D_B}{4z}\left[\left(\frac{B}{D_B}\right)^2 - 1\right]\frac{V_B}{A_c z + V_B} \tag{8.36}$$

   where V_B is the volume of the piston bowl, A_c is the cross-sectional area of the cylinder ($\pi B^2/4$), S_p is the instantaneous piston speed [Eq. (2.11)], and z is the distance between the piston crown top and the cylinder head ($z = c + Z$, where $Z = l + a - s$; see Fig. 2-1).

2. *Simple wedge chamber* (Fig. 8-20*b*):[34]

$$\frac{v_{sq}}{S_p} = \frac{A_s}{b(Z + c)}\left(1 - \frac{Z + c}{C + Z}\right) \tag{8.37}$$

   where A_s is the squish area, b is the width of the squish region, and C is $Z/(r_c - 1)$ evaluated at the end of induction.

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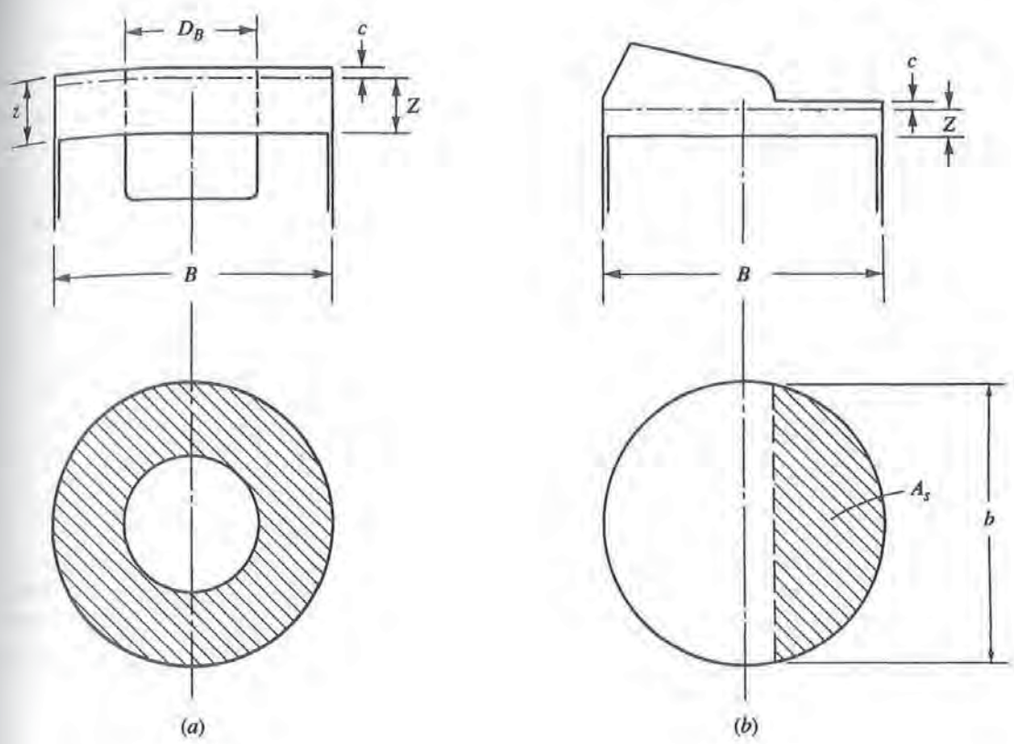


FIGURE 8-20
(*a*) Schematic of axisymmetric bowl-in-piston chamber for Eq. (8.36). (*b*) Schematic of wedge chamber with transverse squish for Eq. (8.37).

The theoretical squish velocity for a bowl-in-piston engine normalized by the mean piston speed $\bar{S}_p$ is shown in Fig. 8-21 for different ratios of D_B/B and clearance heights c. The maximum squish velocity occurs at about 10° before TC. After TC, v_{sq} is negative; a reverse squish motion occurs as gas flows out of the bowl into the clearance height region. Under motored conditions this is equal to the forward motion.

These models omit the effects of gas inertia, friction, gas leakage past the piston rings, and heat transfer. Gas inertia and friction effects have been shown to be small. The effects of gas leakage past the piston rings and of heat transfer are more significant. The squish velocity decrement Δv_L due to leakage is proportional to the mean piston speed and a dimensionless leakage number:

$$N_L = A_{E,L} \frac{\sqrt{\gamma R T_{IVC}}}{N V_d} \tag{8.38}$$

where $A_{E,L}$ is the effective leakage area and T_{IVC} is the temperature of the cylinder gases at inlet valve closing. Leakage was modeled as a choked flow through the effective leakage area. Values of $\Delta v_L/v_{sq}$ are shown in Fig. 8-22. The effect of leakage on v_{sq} is small for normal gas leakage rates. A decrement on squish

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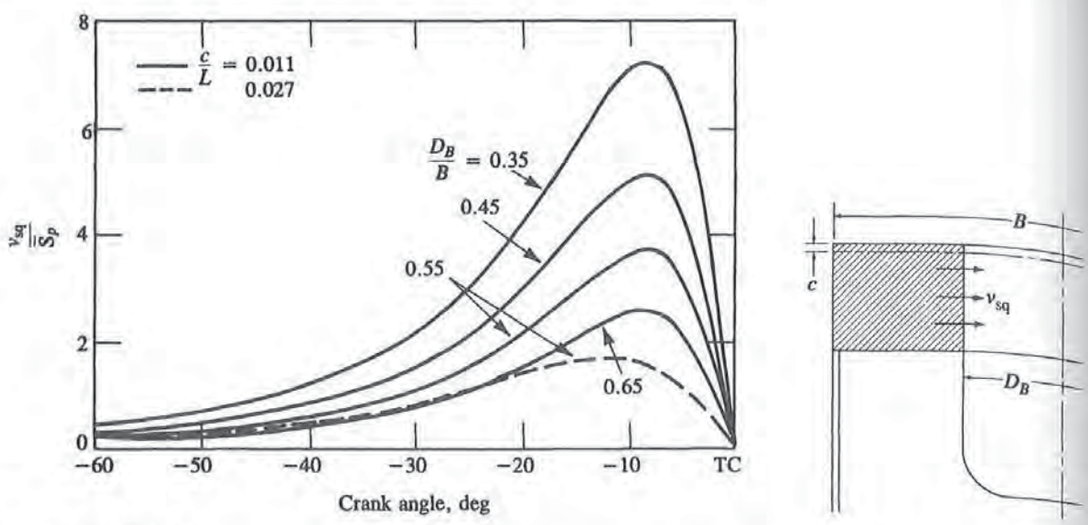


FIGURE 8-21
Theoretical squish velocity divided by mean piston speed for bowl-in-piston chambers, for different D_B/B and c/L (clearance height/stroke). $B/L = 0.914$, $V_B/V_d = 0.056$, connecting rod length/crank radius = 3.76.[35]

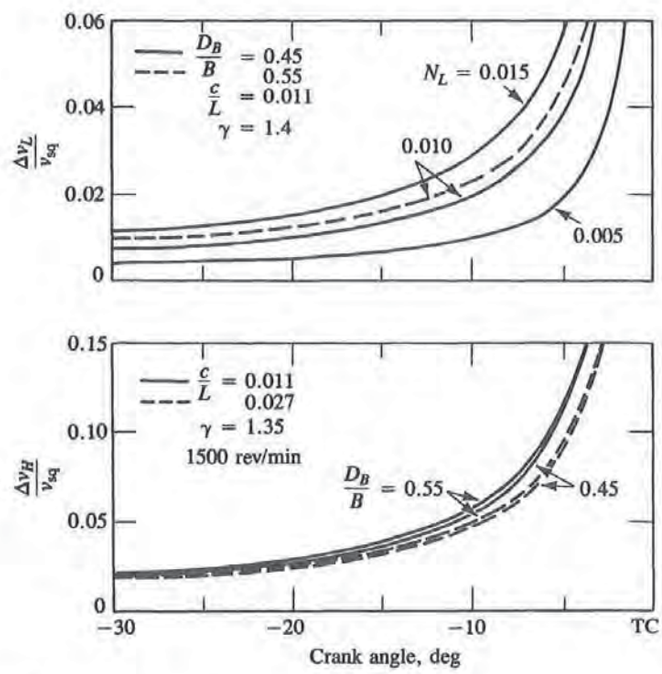


FIGURE 8-22
Values of squish velocity decrement due to leakage Δv_L and heat transfer Δv_H, normalized by the ideal squish velocity, as a function of crank angle.[35]

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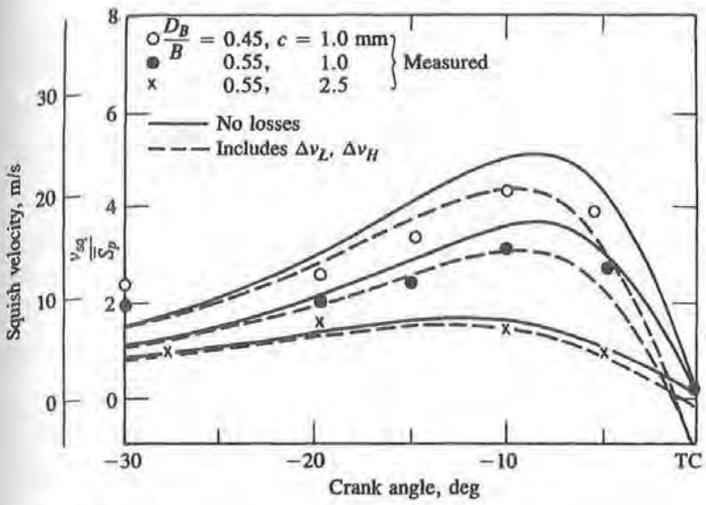


FIGURE 8-23
Comparison of measured squish velocities in bowl-in-piston combustion chambers, with different bowl diameter/bore ratios and clearance heights, to calculated ideal squish velocity (solid lines) and calculations corrected for leakage and heat transfer (dashed lines). Bore = 85 mm, stroke = 93 mm, 1500 rev/min.[35]

velocity due to heat transfer, Δv_H, has also been derived, using standard engine heat-transfer correlations (see Sec. 12.4). Values of $\Delta v_H/v_{sq}$ are also shown in Fig. 8-22. Again the effects are small in the region of maximum squish, though they become more important as the squish velocity decreases from its maximum value as the piston approaches TC.

Velocity measurements in engines provide good support for the above theory. The ideal theory adequately predicts the dependence on engine speed.[36] With appropriate corrections for leakage and heat-transfer effects, the above theory predicts the effects of the bowl diameter/bore ratio and clearance height on squish velocity (see Fig. 8-23). The change in direction of the radial motion as the piston moves through TC has been demonstrated under motored engine conditions. Under firing conditions, the combustion generated gas expansion in the open portion of the combustion chamber substantially increases the magnitude of the reverse squish motion after TC.[37]

8.5 PRECHAMBER ENGINE FLOWS

Small high-speed diesel engines use an auxiliary combustion chamber, or prechamber, to achieve adequate fuel-air mixing rates. The prechamber is connected to the main combustion chamber above the piston via a nozzle, passageway, or one or more orifices. Flow of air through this restriction into the prechamber during the compression process sets up high velocities in the prechamber at the time the fuel-injection process commences. This results in the required high fuel-air mixing rates. Figures 1-21 and 10-2 show examples of these prechamber or

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indirect-injection diesels. The two most common designs of auxiliary chamber are: the swirl chamber (Fig. 10-2*a*), where the flow through the passageway enters the chamber tangentially producing rapid rotation within the chamber, and the prechamber (Fig. 10-2*b*) with one or more connecting orifices designed to produce a highly turbulent flow but no ordered motion within the chamber. Auxiliary chambers are sometimes used in spark-ignition engines. The torch-ignition three-valve stratified-charge engine (Fig. 1-27) is one such concept. The prechamber is used to create a rich mixture in the vicinity of the spark plug to promote rapid flame development. An alternative concept uses the prechamber around the spark plug to generate turbulence to enhance the early stages of combustion, but has no mixture stratification.

The most critical phase of flow into the prechamber occurs towards the end of compression. While this flow is driven by a pressure difference between the main chamber above the piston and the auxiliary chamber, this pressure difference is small, and the mass flow rate and velocity at the nozzle, orifice, or passageway can be estimated using a simple gas displacement model. Assuming that the gas density throughout the cylinder is uniform (an adequate assumption toward the end of compression—the most critical period), the mass in the prechamber m_P is given by $m_c(V_P/V)$, where m_c is the cylinder mass, V the cylinder volume, and V_P the prechamber volume. The mass flow rate through the throat of the restriction is, therefore,

$$\dot{m} = \frac{dm_P}{dt} = -\frac{m_c V_P}{V^2}\frac{dV}{dt} \tag{8.39}$$

Using the relations $dV/dt = -(\pi B^2/4)S_p$ where S_p is the instantaneous piston speed, $V_d = \pi B^2 L/4$, and $\bar{S}_p = 2NL$, Eq. (8.39) can be written as

$$\frac{\dot{m}}{m_c N} = 2(r_c - 1)\left(\frac{V_P}{V_c}\right)\left(\frac{S_p}{\bar{S}_p}\right)\left(\frac{V_c}{V}\right)^2 \tag{8.40}$$

where V_c is the clearance volume, $S_p/\bar{S}_p$ is given by Eq. (2.11), and V/V_c is given by Eq. (2.6). The gas velocity at the throat v_T can be obtained from $\dot{m}$ via the relation $\rho v_T A_T = \dot{m}$, the density $\rho = m_c/V$, and Eq. (8.40):

$$\frac{v_T}{\bar{S}_p} = \left(\frac{V_P}{V_c}\right)\left(\frac{\pi B^2/4}{A_T}\right)\left(\frac{S_p}{\bar{S}_p}\right)\left(\frac{V_c}{V}\right) \tag{8.41}$$

where A_T is the effective cross-sectional area of the throat. The variation of $\dot{m}/(m_c N)$ and $v_T/\bar{S}_p$ with crank angle during the compression process for values of r_c, V_P/V_c, and $A_T/(\pi B^2/4)$ typical of a swirl prechamber diesel are shown in Fig. 8-24. The velocity reaches its peak value about 20° before TC: very high gas velocities, an order of magnitude or more larger than the mean piston speed, can be achieved depending on the relative effective throat area. Note that as the piston approaches TC, first the nozzle velocity and then the mass flow rate decrease to zero. After TC, in the absence of combustion, an equivalent flow in the reverse direction out of prechamber would occur. Combustion in the pre-

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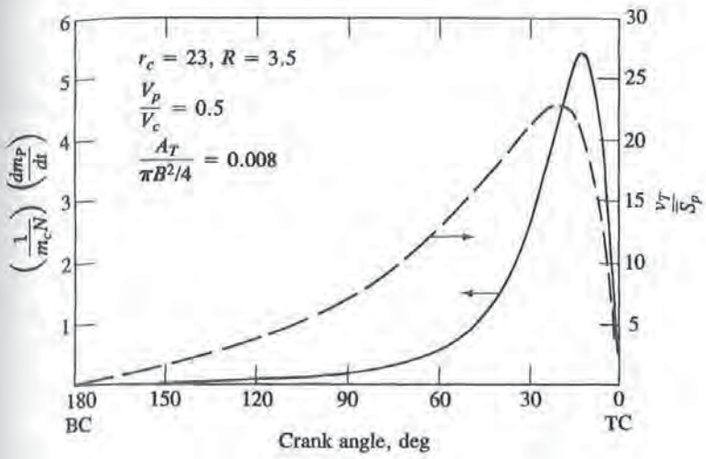


FIGURE 8-24
Velocity and mass flow rate at the prechamber nozzle throat, during compression, for a typical small swirl-prechamber automotive diesel.

chamber diesel usually starts just before TC, and the pressure in the prechamber then rises significantly above the main chamber pressure. The outflow from the prechamber is then governed by the development of the combustion process, and the above simple gas displacement model no longer describes the flow. This combustion generated prechamber gas motion is discussed in Sec. 14.4.4.

In prechamber stratified-charge engines, the flow of gas into the prechamber during compression is critical to the creation of an appropriate mixture in the prechamber at the crank angle when the mixture is ignited. In the concept shown in Fig. 1-27, a very rich fuel-air mixture is fed directly to the prechamber during intake via the prechamber intake valve, while a lean mixture is fed to the main chamber via the main intake valve. During compression, the flow into the prechamber reduces the prechamber equivalence ratio to a close-to-stoichiometric value at the time of ignition. Figure 8-25 shows a gas displacement calculation of this process and relevant data; the prechamber equivalence ratio, initially greater

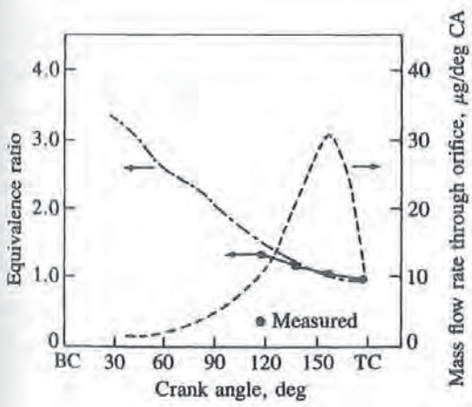


FIGURE 8-25
Effect of gas flow into the prechamber during compression on the prechamber equivalence ratio in a three-valve prechamber stratified-charge engine. Calculations based on gas displacement model.[38]

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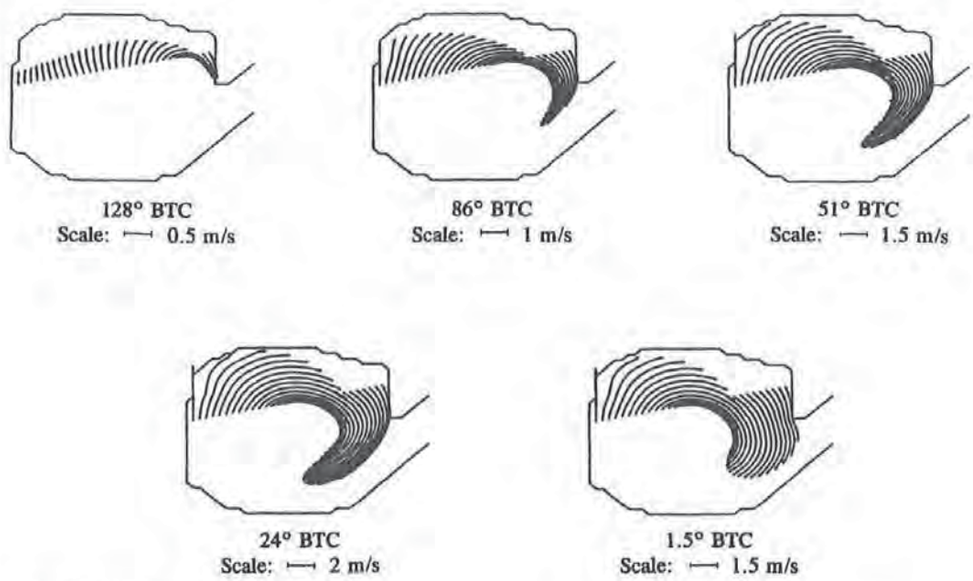


FIGURE 8-26
Calculations of developing flow field in (two-dimensional) swirl prechamber during compression process. Lines are instantaneous flow streamlines, analogous to streak photographs of flow field.[41]

than 3, is leaned out to unity as mass flows through the orifice into the prechamber (whose volume is 8.75 percent of the clearance volume).[38] Charts for estimating the final equivalence ratio, based on gas displacement, for this prechamber concept are available.[39]

The velocity field set up inside the prechamber during compression is strongly dependent on the details of the nozzle and prechamber geometry. Velocities vary linearly with mean piston speed.[40] In swirl prechambers, the nozzle flow sets up a vortex within the chamber. Figure 8-26 shows calculations of this developing flow field; instantaneous flow streamlines have been drawn in, with the length of the streamlines indicating how the particles of fluid move relative to each other.[41] The velocities increase with increasing crank angle as the compression process proceeds, and reach a maximum at about 20° before TC. Then, as the piston approaches TC and the flow through the passageway decreases to zero, the vortex in the swirl chamber expands to fill the entire chamber and mean velocities decay. Very high swirl rates can be achieved just before TC: local swirl ratios of up to 60 at intermediate radii and up to 20 at the outer radius have been measured. These high swirl rates produce large centrifugal accelerations.

8.6 CREVICE FLOWS AND BLOWBY

The engine combustion chamber is connected to several small volumes usually called *crevices* because of their narrow entrances. Gas flows into and out of these volumes during the engine operating cycle as the cylinder pressure changes.

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The largest crevices are the volumes between the piston, piston rings, and cylinder wall. Some gas flows out of these regions into the crankcase; it is called *blowby*. Other crevice volumes in production engines are the threads around the spark plug, the space around the plug center electrode, the gap around the fuel injector, crevices between the intake and exhaust valve heads and cylinder head, and the head gasket cutout. Table 8.1 shows the size and relative importance of these crevice regions in one cylinder of a production V-6 spark-ignition engine determined from measurements of cold-engine components. Total crevice volume is a few percent of the clearance volume, and the piston and ring crevices are the dominant contributors. When the engine is warmed up, dimensions and crevice volumes will change.

The important crevice processes occurring during the engine cycle are the following. As the cylinder pressure rises during compression, unburned mixture or air is forced into each crevice region. Since these volumes are thin they have a large surface/volume ratio; the gas flowing into the crevice cools by heat transfer to close to the wall temperature. During combustion while the pressure continues to rise, unburned mixture or air, depending on engine type, continues to flow into these crevice volumes. After flame arrival at the crevice entrance, burned gases will flow into each crevice until the cylinder pressure starts to decrease. Once the crevice gas pressure is higher than the cylinder pressure, gas flows back from each crevice into the cylinder.

The volumes between the piston, piston rings, and cylinder wall are shown schematically in Fig. 8-27. These crevices consist of a series of volumes (numbered 1 to 5) connected by flow restrictions such as the ring side clearance and ring gap. The geometry changes as each ring moves up and down in its ring groove, sealing either at the top or bottom ring surface. The gas flow, pressure distribution, and ring motion are therefore coupled. Figure 8-28 illustrates this behavior: pressure distributions, ring motion, and mass flow of gas into and out

TABLE 8.1
V-6 engine crevice data†[42]

	cm^3	%
Displaced volume per cylinder	632	
Clearance volume per cylinder	89	100
Volume above first ring	0.93	1.05
Volume behind first ring	0.47	0.52
Volume between rings	0.68	0.77
Volume behind second ring	0.47	0.52
Total ring crevice volume	2.55	2.9
Spark plug thread crevice	0.25	0.28
Head gasket crevice	0.3	0.34
Total crevice volume	3.1	3.5

† Determined for cold engine.

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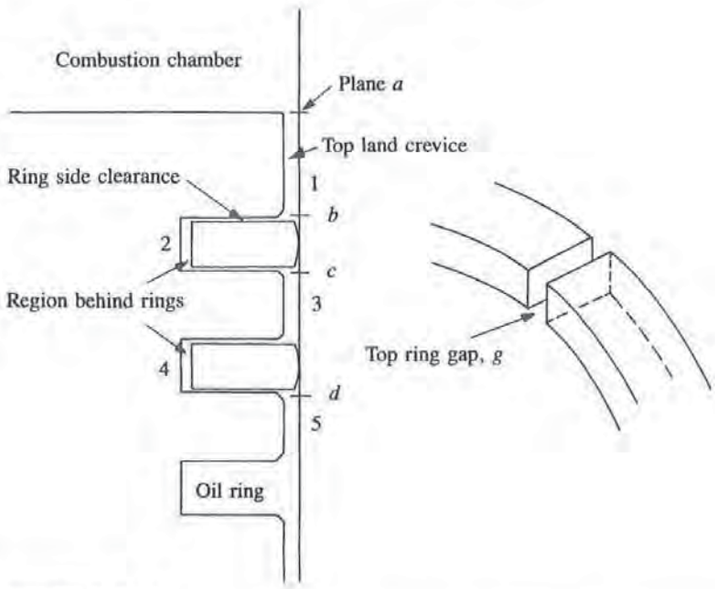


FIGURE 8-27
Schematic of piston and ring assembly in automotive spark-ignition engine.

of the regions defined by planes *a*, *b*, *c*, *d*, and through the ring gap *g* are plotted versus crank angle through compression and expansion. These results come from an analysis of these regions as volumes connected by passageways, with a prescribed cylinder pressure versus crank angle profile coupled with a dynamic model for ring motion, and assuming that the gas temperature equals the wall temperature.[42] During compression and combustion, the rings are forced to the groove lower surfaces and mass flows *into* all the volumes in this total crevice region. The pressure above and behind the first ring is essentially the same as the cylinder pressure; there is a substantial pressure drop across each ring, however. Once the cylinder pressure starts to decrease (after 15° ATC) gas flows out of regions 1 and 2 in Fig. 8-27 into the cylinder, but continues to flow *into* regions 3, 4, and 5 until the pressure in the cylinder falls below the pressure beneath the top ring. The top ring then shifts to seal with the upper grove surface and gas flows out of regions 2, 3, and 4 (which now have the same pressure), both into the cylinder and as blowby into the crackcase. Some 5 to 10 percent of the total cylinder charge is trapped in these regions at the time of peak cylinder pressure. Most of this gas returns to the cylinder; about 1 percent goes to the crankcase as blowby. The gas flow back into the cylinder continues throughout the expansion process. In spark-ignition engines this phenomenon is a major contributor to unburned hydrocarbon emissions (see Sec. 11.4.3). In all engines it results in a loss of power and efficiency.

There is substantial experimental evidence to support the above description of flow in the piston ring crevice region. In a special square-cross-section flow visualization engine, both the low-velocity gas expansion out of the volume above the first ring after the time of peak pressure and the jet-like flows through

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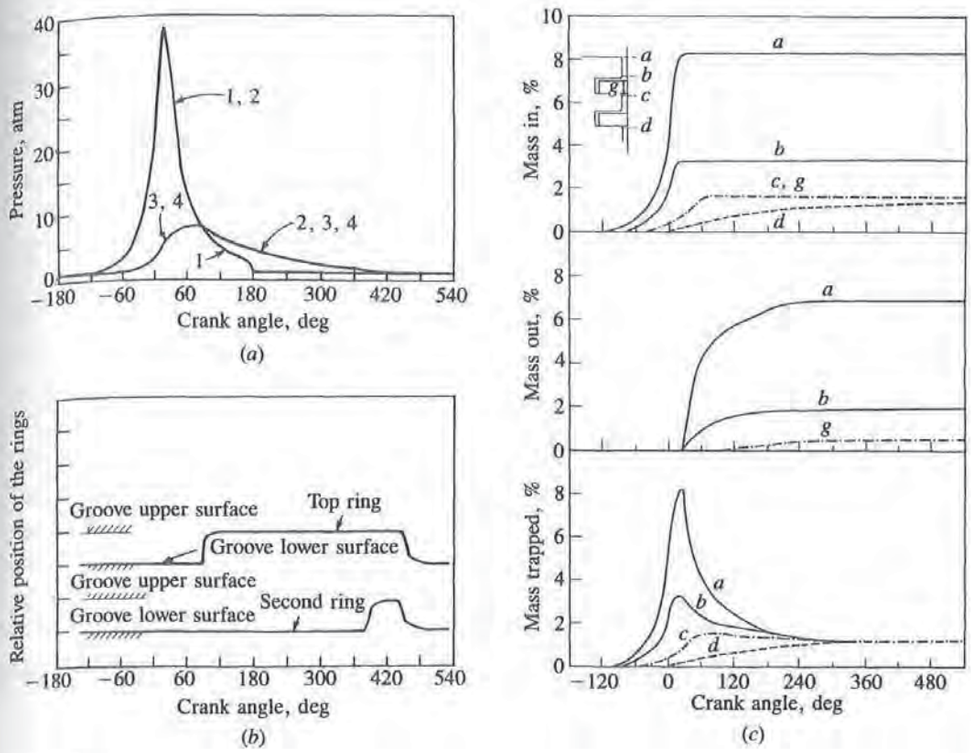


FIGURE 8-28
(*a*) Pressures in the combustion chamber (1), in region behind top ring (2), in region between rings (3), and behind second ring (4); (*b*) relative position of top and second rings; (*c*) percentage of total cylinder mass that flows into and out of the different crevice regions across planes *a*, *b*, *c*, and *d* and through the ring gap *g* in Fig. 8-27, and the percentage of mass trapped beneath these planes, as a function of crank angle. Automotive spark-ignition engine at wide-open throttle and 2000 rev/min.[42]

the top ring gap later in the expansion process when the pressure difference across the ring changes sign have been observed. Figure 8-29 shows these flows with explanatory schematics.

Blowby is defined as the gas that flows from the combustion chamber past the piston rings and into the crankcase. It is forced through any leakage paths afforded by the piston-bore-ring assembly in response to combustion chamber pressure. If there is good contact between the compression rings and the bore, and the rings and the bottom of the grooves, then the only leakage path of consequence is the ring gap. Blowby of gases from the cylinder to the crankcase removes gas from these crevice regions and thereby prevents some of the crevice gases from returning to the cylinder. Crankcase blowby gases used to be vented directly to the atmosphere and constituted a significant source of HC emissions. The crankcase is now vented to the engine intake system and the blowby gases are recycled. Blowby at a given speed and load is controlled primarily by the greatest flow resistance in the flow path between the cylinder and the crankcase. This is the smallest of the compression ring ring-gap areas. Figure 8-30 shows how measured blowby flow rates increase linearly with the smallest gap area.[43]

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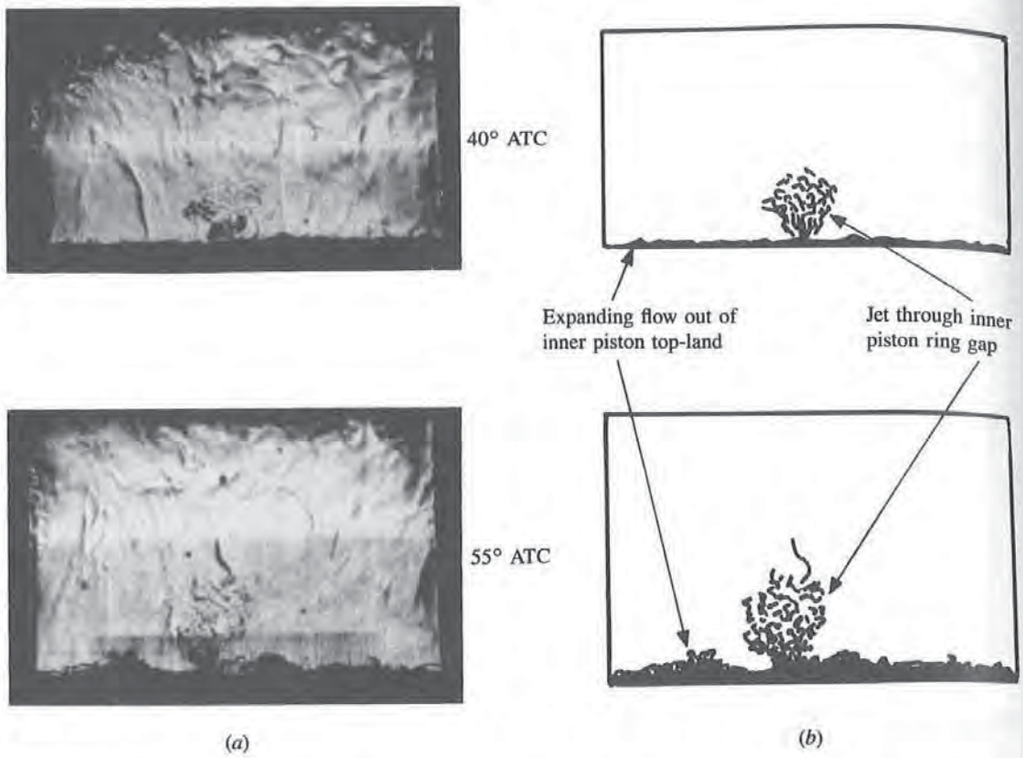


FIGURE 8-29
Schlieren photographs of the flow out of the piston–cylinder wall crevices during the expansion stroke. A production piston was inserted into the square cross-section piston of the visualization engine. Gas flows at low velocity out of the crevice entrance all around the production piston circumference once the cylinder pressure starts decreasing early in the expansion stroke. Gas flows out of the ring gap as a jet once the pressure above the ring falls below the pressure beneath the ring.[42]

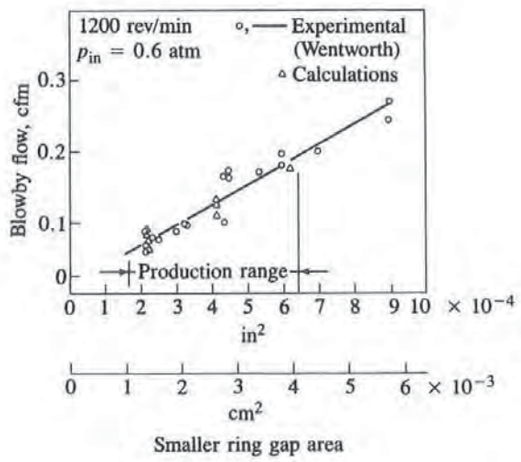


FIGURE 8-30
Measured blowby for one cylinder of an automobile spark-ignition engine as a function of the smallest ring gap area, compared with blowby calculations based on flow model described in text.[42, 43]

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Calculations of blowby based on the model described earlier are in good agreement.[42] Extrapolation back to the zero gap area gives nearly zero blowby. Note, however, that if the bore finish is rough, or if the rings do not contact the bore all around, or if the compression rings lift off the bottom of the groove, this linear relationship may no longer hold.

8.7 FLOWS GENERATED BY PISTON–CYLINDER WALL INTERACTION

Because a boundary layer exists on the cylinder wall, the motion of the piston generates unusual flow patterns in the corner formed by the cylinder wall and the piston face. When the piston is moving away from top-center a sink-type flow occurs. When the piston moves toward top-center a vortex flow is generated. Figure 8-31 shows schematics of these flows (in a coordinate frame with the piston face at rest). The vortex flow has been studied because of its effect on gas motion at the time of ignition and because it has been suggested as a mechanism for removing hydrocarbons off the cylinder wall during the exhaust stroke (see Sec. 11.4.3).

The vortex flow has been studied in cylinders with water as the fluid over the range of Reynolds numbers typical of engine operation.[44, 45] Laminar, transition, and turbulent flow regimes have been identified. It has been shown that a quasi-steady flow assumption is valid and that

$$\frac{A_V}{L^2} = f\left(\frac{v_w L}{\nu}\right)$$

where A_V is the vortex area (area inside the dashed line in Fig. 8-31), L is the stroke, v_w is the wall velocity in piston stationary coordinates ($v_w = S_p$ in the engine), ν is the kinematic viscosity, and $(v_w L/\nu)$ is a Reynolds number.

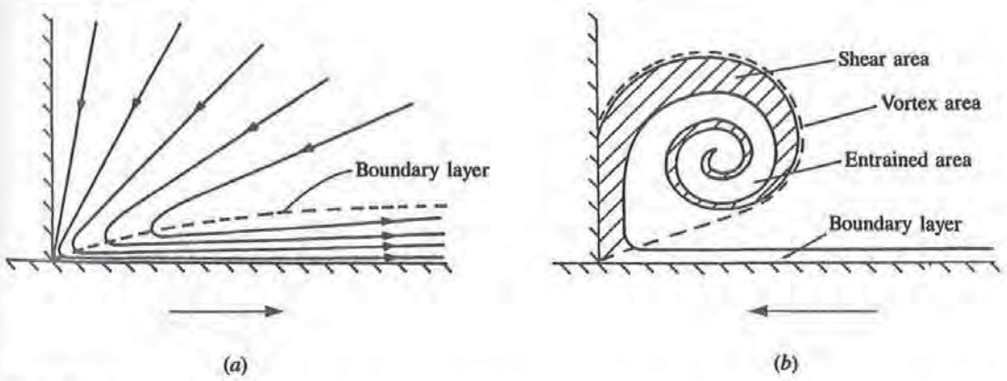


FIGURE 8-31
Schematics of the flow pattern set up in the piston face–cylinder wall corner, in piston-stationary coordinates, due to the boundary layer on the cylinder wall. Piston crown on left; cylinder wall at bottom. (*a*) Sink flow set up during intake and expansion; (*b*) vortex flow set up during compression and exhaust.[44] Arrow shows cylinder wall velocity relative to piston.

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For the laminar flow regime, a good assumption is that A_V is proportional to the shear area in the vortex (shown cross-hatched), which equals the boundary-layer area; this can be estimated from boundary-layer theory. In the turbulent flow regime, an entrainment theory was used, which assumed that the rate of change of vortex area was proportional to the product of the exposed perimeter of the vortex and the velocity difference between the vortex and the stationary fluid ($\approx v_w$). The relevant relationships are:

$$\text{For } (v_w L/\nu) \leq 2 \times 10^4: \qquad \frac{A_V}{L^2} = \left(\frac{v_w L}{\nu}\right)^{-1/2} \tag{8.42a}$$

$$\text{For } (v_w L/\nu) \geq 2 \times 10^4: \qquad \frac{A_V}{L^2} = 0.006 \tag{8.42b}$$

Figure 8-32 shows these two theories correlated against hydraulic analog data.

These theories are for constant values of ν. During compression, ν decreases substantially as the gas temperature and pressure increase (ν decreases by a factor of 4 for a compression ratio of 8). This will decrease the size of the vortex until the turbulent regime is reached. During the exhaust stroke following blowdown, ν will remain approximately constant as the pressure and temperature do not change significantly. Typical parameter values at 1500 rev/min are: $\bar{S}_p = 5$ m/s, $L = 0.1$ m; average values of ν are 1.2×10^{-5} and 1.4×10^{-4} m^2/s for compression and exhaust stroke, respectively. Hence a Reynolds number for the compression stroke is 4×10^4, $A_V/L^2 \approx 0.006$, and the vortex diameter $d_V \approx 0.09L$. For the exhaust stroke, the Reynolds number is 4×10^3, $A_V/L^2 \approx 0.015$, and $d_V \approx 0.14L$. Thus the vortex dimensions at the end of the upward stroke of the piston are comparable to the engine clearance height.

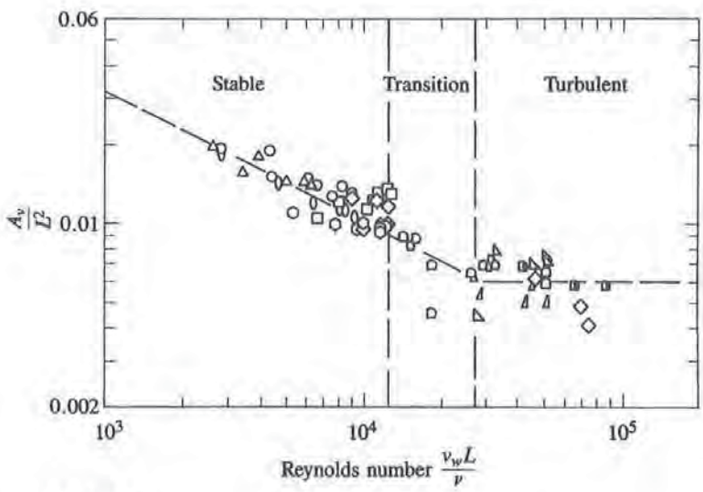


FIGURE 8-32
Ratio of area of vortex in piston face–cylinder wall corner to square of stroke, as a function of Reynolds number based on piston velocity, for piston moving toward the cylinder head.[44]

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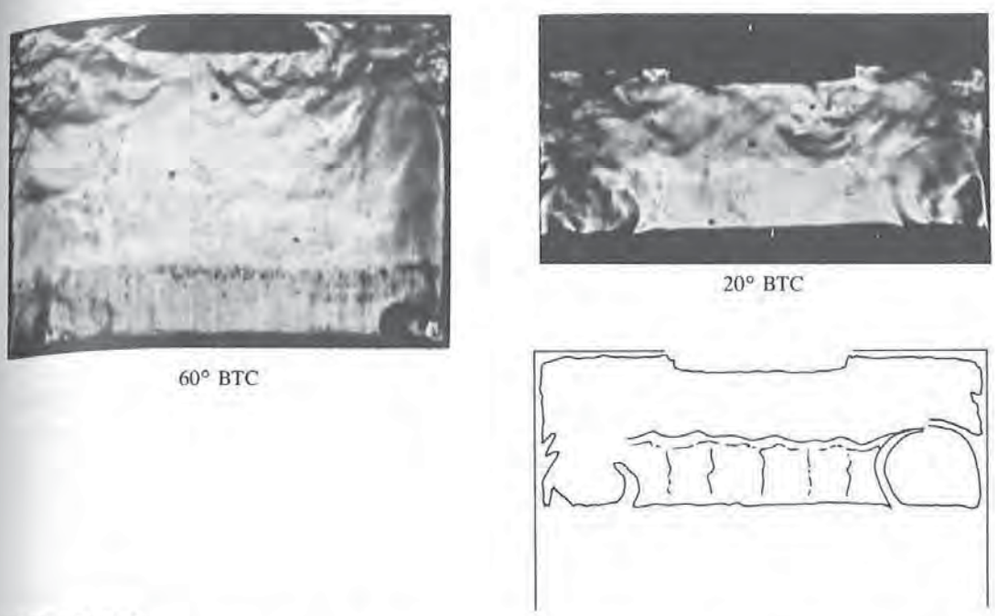


FIGURE 8-33
Schlieren photographs of in-cylinder flow during later stages of exhaust stroke. Growing vortex in the piston face–cylinder wall corner and turbulent outflow toward the valve are apparent at 60° BTC. At 20° BTC, the vortex has grown to of order $0.2B$ diameter.[42]

This vortex flow has been observed in an operating engine. Figure 8-33 shows schlieren photographs taken during the exhaust stroke in a special square-cross-section flow visualization spark-ignition engine. The accompanying schematic identifies the vortex structure which is visible in the photo because the cool boundary-layer gas is being scraped off the cylinder wall by the upward-moving piston and "rolled up." The vortex diameter as the piston approaches TC is about 20 percent of the bore.

PROBLEMS

8.1. (*a*) Estimate the ratio of the maximum gas velocity in the center of the hollow cone inlet jet to the mean piston speed from the data in Fig. 8-1.
(*b*) Compare this ratio with the ratio of inlet valve pseudo flow velocity determined from Fig. 6-15 to the mean piston speed at the same crank angle. The engine is that of Fig. 1-4.
(*c*) Are the engine velocity data in (*a*) consistent with the velocity calculated from the simple piston displacement model of (*b*)? Explain.

8.2. Given the relationship between turbulence intensity and mean piston speed [Eq. (8.23)] and that the turbulence integral scale is $\approx 0.2 \times$ clearance height, use Eqs. (8.14) and (8.15) to estimate the following quantities for a spark-ignition engine with bore = stroke = 86 mm, $r_c = 9$, at 1000 and 5000 rev/min and wide-open throttle:
(*a*) Mean and maximum piston speed, maximum gas velocity through the inlet valve (see Prob. 8.1)
(*b*) Turbulence intensity, integral length scale, micro length scale, and Kolmogorov length scale, all at TC

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8.3. The swirl ratio at the end of induction at 2000 rev/min in a direct-injection diesel engine of bore = stroke = 100 mm is 4.0. What is the average tangential velocity (evaluated at the inlet valve–axis radial location) required to give this swirl ratio? What is the ratio of this velocity to the mean piston speed and to the mean flow velocity through the inlet valve estimated from the average valve open area and open time?

8.4. (*a*) Derive a relationship for the depth (or height) h_B of a disc-shaped bowl-in-piston direct-injection diesel engine combustion chamber in terms of compression ratio r_c, bore B, stroke L, bowl diameter D_B, and top-center cylinder-head to piston-crown clearance c. For $B = L = 100$ mm, $r_c = 16$, $D_B = 0.5B$, $c = 1$ mm find the fraction of the air charge within the bowl at TC.

(*b*) If the swirl ratio at the end of induction at 2500 rev/min is 3 find the swirl ratio and average angular velocity in the bowl-in-piston chamber of dimensions given above. Assume the swirling flow is always a solid-body rotation. Compare the tangential velocity at the bowl edge with the mean piston speed. Neglect any friction effects.

(*c*) What would the swirl ratio be if the top-center clearance height was zero?

8.5. Using Eq. (8.37) and Fig. 8-20*b* plot the squish velocity divided by the mean piston speed at 10° BTC (the approximate location of the maximum) as a function of squish area expressed as a percentage of the cylinder cross section, $A_s/(\pi B^2/4) \times 100$, from 50 to 0 percent. $r_c = 10$, $c/B = 0.01$, $B/L = 1$, $R = l/a = 3.5$.

8.6. Figure 8-24 shows the velocity at the prechamber nozzle throat during compression for dimensions typical of a small swirl chamber indirect-injection diesel. Assuming that the swirl chamber shape is a disc of height equal to the diameter, that the nozzle throat is at 0.8 × prechamber radius, and that the flow enters the prechamber tangentially, estimate the swirl ratio based on the total angular momentum about the swirl chamber axis in the prechamber at top-center. Assume $B = L$; neglect friction.

8.7. The total crevice volume in an automobile spark-ignition engine is about 3 percent of the clearance volume. If the gas in these crevice regions is close to the wall temperature (450 K) and at the cylinder pressure, estimate the fraction of the cylinder mass within these crevice regions at these crank angles: inlet valve closing (50° ABC), spark discharge (30° BTC), maximum cylinder pressure (15° ATC), exhaust valve opening (60° BBC), TC of the exhaust stroke. Use the information in Fig. 1-8 for your input data, and assume the inlet pressure is 0.67 atm.

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CHAPTER 9

COMBUSTION IN SPARK-IGNITION ENGINES

9.1 ESSENTIAL FEATURES OF PROCESS

In a conventional spark-ignition engine the fuel and air are mixed together in the intake system, inducted through the intake valve into the cylinder, where mixing with residual gas takes place, and then compressed. Under normal operating conditions, combustion is initiated towards the end of the compression stroke at the spark plug by an electric discharge. Following inflammation, a turbulent flame develops, propagates through this essentially premixed fuel, air, burned gas mixture until it reaches the combustion chamber walls, and then extinguishes. Photographs of this process taken in operating engines illustrate its essential features. Figure 9-1 (color plate) shows a sequence of frames from a high-speed color movie of the combustion process in a special single-cylinder engine with a glass piston crown.[1] The spark discharge is at $-30°$. The flame first becomes visible in the photos at about $-24°$. The flame, approximately circular in outline in this

FIGURE 9-1 (On color plate opposite p. 498)
Color photographs from high-speed movie of spark-ignition engine combustion process, taken through glass piston crown. Ignition timing 30° BTC, light load, 1430 rev/min, $(A/F) = 19$.[1]

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view through the piston, then propagates outward from the spark plug location. The blue light from the flame is emitted most strongly from the front. The irregular shape of the turbulent flame front is apparent. At TC the flame diameter is about two-thirds of the cylinder bore. The flame reaches the cylinder wall farthest from the spark plug about 15° ATC, but combustion continues around parts of the chamber periphery for another 10°. At about 10° ATC, additional radiation—initially white, turning to pinky-orange—centered at the spark plug location is evident. This afterglow comes from the gases behind the flame which burned earlier in the combustion process, as these are compressed to the highest temperatures attained within the cylinder (at about 15° ATC) while the rest of the charge burns.[2, 3]

Additional features of the combustion process are evident from the data in Fig. 9-2, taken from several consecutive cycles of an operating spark-ignition engine. The cylinder pressure, fraction of the charge mass which has burned (determined from the pressure data, see Sec. 9.2), and fraction of the cylinder volume enflamed by the front (determined from photographs like Fig. 9-1) are shown, all as a function of crank angle.[4] Following spark discharge, there is a period during which the energy release from the developing flame is too small for the pressure rise due to combustion to be discerned. As the flame continues to grow and propagate across the combustion chamber, the pressure then steadily rises above the value it would have in the absence of combustion. The pressure reaches a maximum after TC but before the cylinder charge is fully burned, and then decreases as the cylinder volume continues to increase during the remainder of the expansion stroke.

The flame development and subsequent propagation obviously vary, cycle-by-cycle, since the shape of the pressure, volume fraction enflamed, and mass fraction burned curves for each cycle differ significantly. This is because flame growth depends on local mixture motion and composition. These quantities vary in successive cycles in any given cylinder and may vary cylinder-to-cylinder. Especially significant are mixture motion and composition in the vicinity of the spark plug at the time of spark discharge since these govern the early stages of flame development. Cycle-by-cycle and cylinder-to-cylinder variations in combustion are important because the extreme cycles limit the operating regime of the engine (see Sec. 9.4.1).

Note that the volume fraction enflamed curves rise more steeply than the mass fraction burned curves. In large part, this is because the density of the unburned mixture ahead of the flame is about four times the density of the burned gases behind the flame. Also, there is some unburned mixture behind the visible front to the flame: even when the entire combustion chamber is fully enflamed, some 25 percent of the mass has still to burn. From this description it is plausible to divide the combustion process into four distinct phases: (1) spark ignition; (2) early flame development; (3) flame propagation; and (4) flame termination. Our understanding of each of these phases will be developed in the remainder of this chapter.

The combustion event must be properly located relative to top-center to

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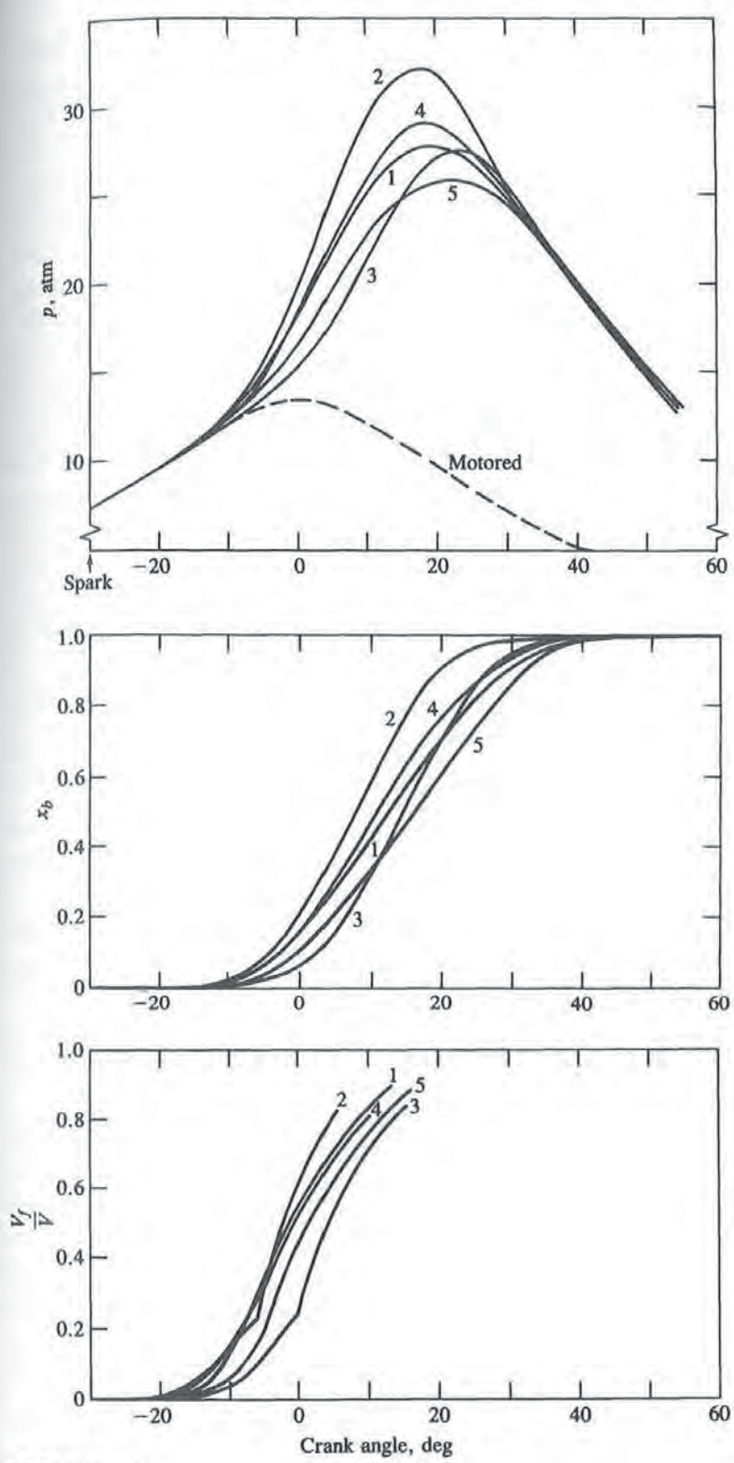


FIGURE 9-2
Cylinder pressure, mass fraction burned, and volume fraction enflamed for five consecutive cycles in a spark-ignition engine as a function of crank angle. Ignition timing 30° BTC, wide-open throttle, 1044 rev/min, $\phi = 0.98$.[4]

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obtain the maximum power or torque. The combined duration of the flame development and propagation process is typically between 30 and 90 crank angle degrees. Combustion starts before the end of the compression stroke, continues through the early part of the expansion stroke, and ends after the point in the cycle at which the peak cylinder pressure occurs. The pressure versus crank angle curves shown in Fig. 9-3*a* allow us to understand why engine torque (at given engine speed and intake manifold conditions) varies as spark timing is varied relative to TC. If the start of the combustion process is progressively advanced before TC, the compression stroke work transfer (which is *from* the piston *to* the cylinder gases) increases. If the end of the combustion process is progressively delayed by retarding the spark timing, the peak cylinder pressure occurs later in the expansion stroke and is reduced in magnitude. These changes reduce the expansion stroke work transfer *from* the cylinder gases *to* the piston. The optimum timing which gives the maximum brake torque—called *maximum brake torque, or MBT, timing*—occurs when the magnitudes of these two opposing trends just offset each other. Timing which is advanced or retarded from this optimum gives lower torque. The optimum spark setting will depend on the rate of flame development and propagation, the length of the flame travel path across the combustion chamber, and the details of the flame termination process after it reaches the wall. These depend on engine design and operating conditions, and the properties of the fuel, air, burned gas mixture. Figure 9-3*b* shows the effect of variations in spark timing on brake torque for a typical spark-ignition engine. The maximum is quite flat.

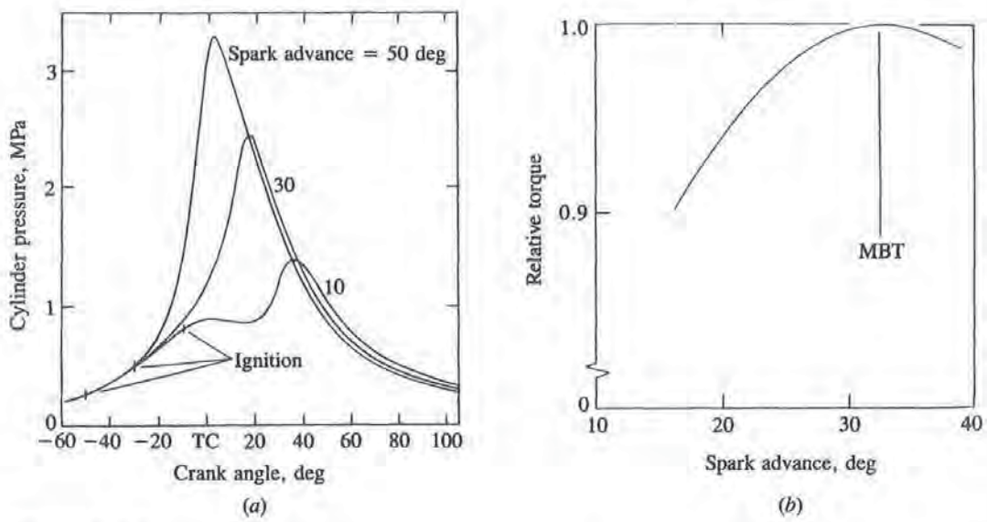


FIGURE 9-3
(*a*) Cylinder pressure versus crank angle for overadvanced spark timing (50°), MBT timing (30°), and retarded timing (10°). (*b*) Effect of spark advance on brake torque at constant speed and (*A*/*F*), at wide-open throttle. MBT is maximum brake torque timing.[5]

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Empirical rules for relating the mass burning profile and maximum cylinder pressure to crank angle at MBT timing are often used. For example, with optimum spark timing: (1) the maximum pressure occurs at about 16° after TC; (2) half the charge is burned at about 10° after TC. In practice, the spark is often retarded to give a 1 or 2 percent reduction in brake torque from the maximum value, to permit a more precise definition of timing relative to the optimum.

So far we have described normal combustion in which the spark-ignited flame moves *steadily* across the combustion chamber until the charge is fully consumed. However, several factors—e.g., fuel composition, certain engine design and operating parameters, and combustion chamber deposits—may prevent this normal combustion process from occurring. Two types of abnormal combustion have been identified: knock and surface ignition.

Knock is the most important abnormal combustion phenomenon. Its name comes from the noise that results from the autoignition of a portion of the fuel, air, residual gas mixture ahead of the advancing flame. As the flame propagates across the combustion chamber, the unburned mixture ahead of the flame—called the *end gas*—is compressed, causing its pressure, temperature, and density to increase. Some of the end-gas fuel-air mixture may undergo chemical reactions prior to normal combustion. The products of these reactions may then autoignite: i.e., spontaneously and rapidly release a large part or all of their chemical energy. When this happens, the end gas burns very rapidly, releasing its energy at a rate 5 to 25 times that characteristic of normal combustion. This causes high-frequency pressure oscillations inside the cylinder that produce the sharp metallic noise called knock.

The presence or absence of knock reflects the outcome of a race between the advancing flame front and the precombustion reactions in the unburned end gas. Knock will not occur if the flame front consumes the end gas before these reactions have time to cause the fuel-air mixture to autoignite. Knock will occur if the precombustion reactions produce autoignition before the flame front arrives.

The other important abnormal combustion phenomenon is *surface ignition.* Surface ignition is ignition of the fuel-air charge by overheated valves or spark plugs, by glowing combustion-chamber deposits, or by any other hot spot in the engine combustion chamber: it is ignition by any source other than normal spark ignition. It may occur before the spark plug ignites the charge (preignition) or after normal ignition (postignition). It may produce a single flame or many flames. Uncontrolled combustion is most evident and its effects most severe when it results from preignition. However, even when surface ignition occurs after the spark plug fires (postignition), the spark discharge no longer has complete control of the combustion process.

Surface ignition may result in knock. Knock which occurs following normal spark ignition is called *spark knock* to distinguish it from knock which has been preceded by surface ignition. Abnormal combustion phenomena are reviewed in more detail in Sec. 9.6.

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9.2 THERMODYNAMIC ANALYSIS OF SI ENGINE COMBUSTION

9.2.1 Burned and Unburned Mixture States

Because combustion occurs through a flame propagation process, the changes in state and the motion of the unburned and burned gas are much more complex than the ideal cycle analysis in Chapter 5 suggests. The gas pressure, temperature, and density change as a result of changes in volume due to piston motion. During combustion, the cylinder pressure increases due to the release of the fuel's chemical energy. As each element of fuel-air mixture burns, its density decreases by about a factor of four. This combustion-produced gas expansion compresses the unburned mixture ahead of the flame and displaces it toward the combustion chamber walls. The combustion-produced gas expansion also compresses those parts of the charge which have already burned, and displaces them back toward the spark plug. During the combustion process, the unburned gas elements move away from the spark plug; following combustion, individual gas elements move back toward the spark plug. Further, elements of the unburned mixture which burn at different times have different pressures and temperatures just prior to combustion, and therefore end up at different states after combustion. The thermodynamic state and composition of the burned gas is, therefore, non-uniform. A first law analysis of the spark-ignition engine combustion process enables us to quantify these gas states.

Consider the schematic of the engine cylinder while combustion is in progress, shown in Fig. 9-4. Work transfer occurs between the cylinder gases and the piston (*to the gas* before TC; *to the piston* after TC). Heat transfer occurs to the chamber walls, primarily from the burned gases. At the temperatures and pressures typical of spark-ignition engines it is a reasonable approximation to assume that the volume of the reaction zone *where combustion is actually occurring* is a negligible fraction of the chamber volume even though the thickness of the turbulent flame may not be negligible compared with the chamber dimensions (see Sec. 9.3.2). With normal engine operation, at any point in time or crank angle, the pressure throughout the cylinder is close to uniform. The condi-

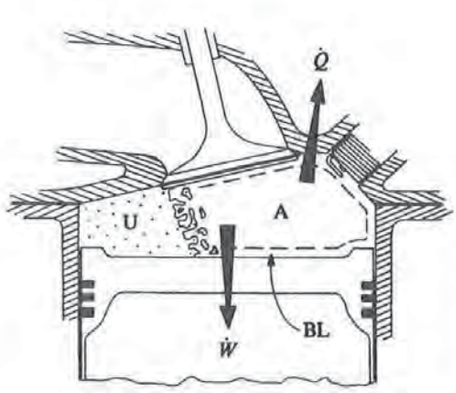


FIGURE 9-4
Schematic of flame in the engine cylinder during combustion: unburned gas (U) to left of flame, burned gas to right. A denotes adiabatic burned-gas core, BL denotes thermal boundary layer in burned gas, $\dot{W}$ is work-transfer rate to piston, $\dot{Q}$ is heat-transfer rate to chamber walls.

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tions in the burned and unburned gas are then determined by conservation of mass:

$$\frac{V}{m} = \int_0^{x_b} v_b \, dx + \int_{x_b}^1 v_u \, dx \tag{9.1}$$

and conservation of energy:

$$\frac{U_0 - W - Q}{m} = \int_0^{x_b} u_b \, dx + \int_{x_b}^1 u_u \, dx \tag{9.2}$$

where V is the cylinder volume, m is the mass of the cylinder contents, v is the specific volume, x_b is the mass fraction burned, U_0 is the internal energy of the cylinder contents at some reference point θ_0, u is the specific internal energy, W is the work done on the piston, and Q is the heat transfer to the walls. The subscripts u and b denote unburned and burned gas properties, respectively. The work and heat transfers are

$$W = \int_{V_0}^{V} p \, dV' \qquad Q = \int_{\theta_0}^{\theta} \left(\frac{\dot{Q}}{360N}\right) d\theta \tag{9.3}$$

where $\dot{Q}$ is the instantaneous heat-transfer rate to the chamber walls.

To proceed further, models for the thermodynamic properties of the burned and unburned gases are required. Several categories of models are described in Chap. 4. Accurate calculations of the state of the cylinder gases require an equilibrium model (or good approximation to it) for the burned gas and an ideal gas mixture model (of frozen composition) for the unburned gas (see Table 4.2). However, useful illustrative results can be obtained by assuming that the burned and unburned gases are different ideal gases, each with constant specific heats;[6] i.e.,

$$pv_b = R_b T_b \qquad u_b = c_{v,b} T_b + h_{f,b} \tag{9.4}$$

$$pv_u = R_u T_u \qquad u_u = c_{v,u} T_u + h_{f,u} \tag{9.5}$$

Combining Eqs. (9.1) to (9.5) gives

$$\frac{pV}{m} = x_b R_b \bar{T}_b + (1 - x_b) R_u \bar{T}_u \tag{9.6}$$

and

$$\frac{U_0 - W - Q}{m} = x_b(c_{v,b} \bar{T}_b + h_{f,b}) + (1 - x_b)(c_{v,u} \bar{T}_u + h_{f,u}) \tag{9.7}$$

where

$$\bar{T}_b = \frac{1}{x_b} \int_0^{x_b} T_b \, dx \qquad \bar{T}_u = \frac{1}{1 - x_b} \int_{x_b}^1 T_u \, dx$$

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are the mean temperatures of the burned and unburned gases. Equations (9.6) and (9.7) may now be solved to obtain

$$x_b = \frac{pV - p_0 V_0 + (\gamma_b - 1)(W + Q) + (\gamma_b - \gamma_u)mc_{v,u}(\bar{T}_u - T_0)}{m[(\gamma_b - 1)(h_{f,u} - h_{f,b}) + (\gamma_b - \gamma_u)c_{v,u}\bar{T}_u]} \tag{9.8}$$

and

$$\bar{T}_b = \frac{R_u}{R_b}\bar{T}_u + \frac{pV - mR_u\bar{T}_u}{mR_b x_b} \tag{9.9}$$

If we now assume the unburned gas is initially uniform and undergoes isentropic compression, then

$$\frac{\bar{T}_u}{T_0} = \left(\frac{p}{p_0}\right)^{(\gamma_u - 1)/\gamma_u} \tag{9.10}$$

This equation, with Eqs. (9.8) and (9.9) enables determination of both x_b and $\bar{T}_b$ from the thermodynamic properties of the burned and unburned gases, and known values of p, V, m, and $\dot{Q}$. Alternatively, if x_b is known then p can be determined. Mass fraction burned and cylinder gas pressure are uniquely related.

While Eq. (9.9) defines a mean burned gas temperature, the burned gas is not uniform. Mixture which burns early in the combustion process is further compressed after combustion as the remainder of the charge is burned. Mixture which burns late in the combustion process is compressed prior to combustion and, therefore, ends up at a different final state. A temperature gradient exists across the burned gas with the earlier burning portions at the higher temperature.[7,8] Two limiting models bracket what occurs in practice: (1) a *fully mixed* model, where it is assumed that each element of mixture which burns mixes instantaneously with the already burned gases (which therefore have a uniform temperature), and (2) an *unmixed* model, where it is assumed that no mixing occurs between gas elements which burn at different times.

In the fully mixed model the burned gas is uniform, $T_b = \bar{T}_b$, and the equations given above fully define the state of the cylinder contents. In the unmixed model, the assumption is made that no mixing occurs between gas elements that burn at different times, and each burned gas element is therefore isentropically compressed (and eventually expanded) after combustion.† Thus:

$$\frac{T_b(x_b', x_b)}{T_b(x_b')} = \left[\frac{p(x_b)}{p(x_b')}\right]^{(\gamma_b - 1)/\gamma_b} \tag{9.11}$$

† This model applies to burned gas regions of the chamber *away* from the walls. Heat transfer to the walls results in a thermal boundary layer on the walls which grows with time. The gas in the boundary layer is not isentropically compressed and expanded.

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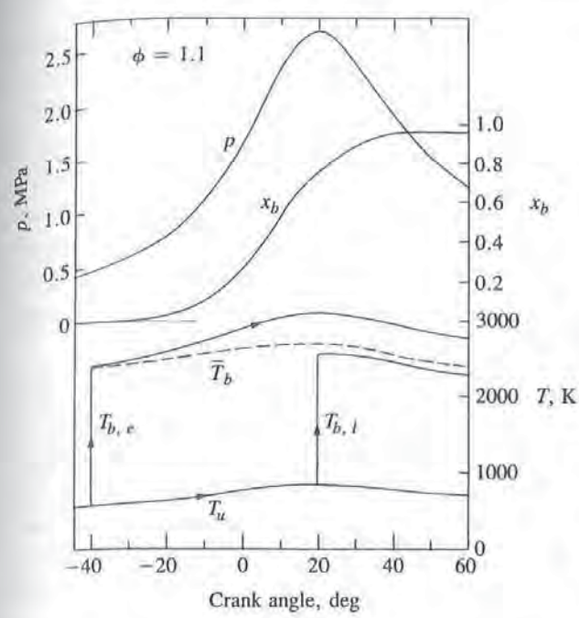


FIGURE 9-5
Cylinder pressure, mass fraction burned, and gas temperatures as functions of crank angle during combustion. T_u is unburned gas temperature, T_b is burned gas temperature, the subscripts e and l denote early and late burning gas elements, and $\bar{T}_b$ is the mean burned gas temperature.[9] (*Reprinted with permission. Copyright 1973, American Chemical Society.*)

where $T_b(x_b', x_b)$ is the temperature of the element which burned at the pressure $p(x_b')$ when the pressure is $p(x_b)$, and

$$T_b(x_b') = \frac{h_{f,u} - h_{f,b} + c_{p,u}\, T_u(x_b')}{c_{p,b}} \tag{9.12}$$

is the temperature resulting from isenthalpic combustion of the unburned gas at $T_u(x_b')$, $p(x_b')$. An example of the temperature distribution computed with this model is shown in Fig. 9-5. A mixture element that burns right at the start of the combustion process reaches, in the absence of mixing, a peak temperature after combustion about 400 K higher than an element that burns toward the end of the combustion process. The mean burned gas temperature is closer to the lower of these temperatures. These two models approximate respectively to situations where the time scale that characterizes the turbulent mixing process in the burned gases is (1) much less than the overall burning time (for the fully mixed model) or (2) much longer than the overall burning time (for the unmixed model). The real situation lies in between.

Measurements of burned gas temperatures have been made in engines using spectroscopic techniques through quartz windows in the cylinder head. Examples of measured temperatures are shown in Fig. 9-6. The solid lines marked A, B, and C are the burned gas temperatures measured by Rassweiler and Withrow[7] using the sodium line reversal technique in an L-head engine, for the spark plug end (A), the middle (B), and the opposite end (C) of the chamber, respectively. Curves labeled W_2 and W_3 were measured by Lavoie[8] through two different windows, W_2 and W_3 (with W_2 closer to the spark), again in an L-head engine. Each set of experimental temperatures shows a temperature gradient across the burned gas comparable to that predicted, and the two sets have similar shapes.

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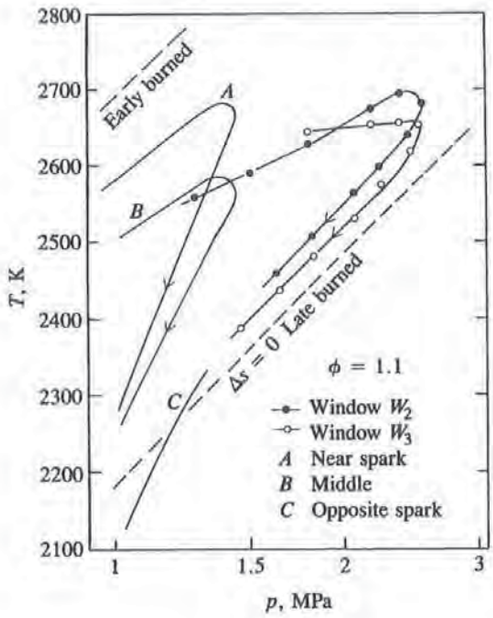


FIGURE 9-6
Burned gas temperatures measured using spectroscopic techniques through windows in the cylinder head, as a function of cylinder pressure. Temperatures measured closer to spark plug have higher values. Dashed lines show isentropic behavior.[7, 8]

In the unmixed model, the temperature of each burned gas element follows a different isentropic line as it is first compressed as p increases to p_{max} and then expanded as the pressure falls after p_{max}. The measured temperature curves in Fig. 9-6 do not follow the calculated isentropes because of gas motion past the observation ports. As has already been mentioned, the expansion of a gas element which occurs during combustion compresses the gas ahead of the flame and moves it away from the spark plug. At the same time, previously burned gas is compressed and moved back toward the spark plug. Defining this motion in an engine requires sophisticated flow models, because the combustion chamber shape is rarely symmetrical, the spark plug is not usually centrally located, and often there is a bulk gas motion at the time combustion is initiated. However, the gas motion in a spherical or cylindrical combustion bomb with central ignition which can readily be computed illustrates the features of the combustion-induced motion in an engine. Figure 9-7 shows calculated particle trajectories for a stoichiometric methane-air mixture, initially at ambient conditions, as a laminar

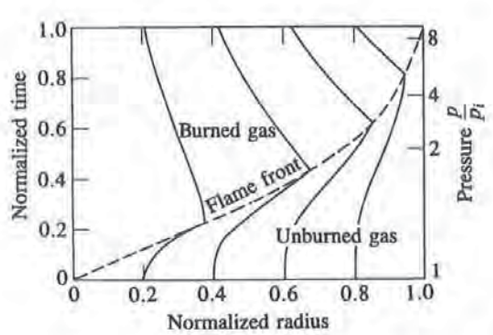


FIGURE 9-7
Particle trajectories in unburned and burned gas as flame propagates outward at constant velocity from the center of a spherical combustion bomb. Stoichiometric methane-air mixture initially at 1 atm and 300 K.

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flame with a constant burning velocity propagates outward from the center of a spherical container. Applying this gas motion model to an engine, it can be concluded that a window in the cylinder head initially views earlier burned gas (of higher temperature and entropy) and that as more of the charge burns, the window views later burned gas of progressively lower entropy. The experimental curves fit this description: they cross the constant entropy lines toward lower entropy. Note that the gradient in temperature persists well into the expansion stroke, indicating that the "unmixed" model is closer to reality than the "fully mixed" model.

More accurate calculations relating the mass fraction burned, gas pressure, and gas temperature distribution are often required. Note that the accuracy of such calculations depends on the accuracy with which the time-varying heat loss to the chamber walls can be estimated (see Sec. 12.4.3) and whether flows into and out of crevice regions are significant (see Sec. 8.6), as well as the accuracy of the models used to describe the thermodynamic properties of the gases. Appropriate more accurate models for the thermodynamic properties are: an equilibrium model for the burned gas, and specific heat models which vary with temperature for each of the components of the unburned mixture (see Secs. 4.1 and 4.7). In the absence of significant crevice effects, Eqs. (9.1) and (9.2) can be written as

$$\frac{V}{m} = \bar{v}_b x_b + \bar{v}_u(1 - x_b) \tag{9.13}$$

$$\frac{U_0 - W - Q}{m} = \bar{u}_b x_b + \bar{u}_u(1 - x_b) \tag{9.14}$$

where

$$\bar{v}_b = \frac{1}{x_b}\int_0^{x_b} v_b \, dx \qquad \text{and} \qquad \bar{v}_u = \frac{1}{1 - x_b}\int_{x_b}^{1} v_u \, dx$$

and similar definitions hold for $\bar{u}_b$ and $\bar{u}_u$. For a given equivalence ratio, fuel and burned gas fraction:

$$h_u = h_u(T_u) \qquad h_b = h_b(T_b, p) \tag{9.15a, b}$$

$$pv_u = \left(\frac{\tilde{R}}{M_u}\right)T_u \qquad pv_b = \left(\frac{\tilde{R}}{M_b}\right)T_b \tag{9.16a, b}$$

and

$$\bar{u}_u = \bar{h}_u - p\bar{v}_u \qquad \bar{u}_b = \bar{h}_b - p\bar{v}_b \tag{9.17a, b}$$

To simplify the calculations, it is convenient to assume that, for the burned gas, $\bar{u}_b = u_b(\bar{T}_b, p)$ and $\bar{v}_b = v_b(\bar{T}_b, p)$. This corresponds to the fully mixed assumption described above. The effect of neglecting the temperature distribution in the calculation of mass fraction burned is small. In addition, the heat losses from the unburned gas can usually be neglected; the unburned gas is then compressed isentropically. $\bar{T}_u$ is specified for some initial state of the unburned gas (where

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$x_b = 0$) by p_0, V_0, M_u, and the mass of charge m. Then, since for any isentropic process

$$\left(\frac{\partial T}{\partial p}\right)_s = \frac{v - (\partial h/\partial p)_T}{(\partial h/\partial T)_p} \tag{9.18}$$

$\bar{T}_u$ can be determined.

Equations (9.13) to (9.18) constitute a set of nine equations for the nine unknowns $\bar{v}_u$, $\bar{v}_b$, $\bar{u}_u$, $\bar{u}_b$, $\bar{h}_u$, $\bar{h}_b$, $\bar{T}_u$, $\bar{T}_b$, and x_b or p. One convenient solution method is to eliminate x_b from Eqs. (9.13) and (9.14) to obtain

$$\frac{(V/m) - \bar{v}_u}{\bar{v}_b - \bar{v}_u} - \frac{(U/m) - \bar{u}_u}{\bar{u}_b - \bar{u}_u} = f(\bar{T}_b, \bar{T}_u) = 0 \tag{9.19}$$

where $U = U_0 - W - Q$. $\bar{T}_u$ can be determined from Eq. (9.18). Equation (9.19) can then be solved using an appropriate iterative technique for $\bar{T}_b$, and x_b can be obtained from Eq. (9.13). An alternative formulation based on the rate of change of pressure $dp/d\theta$ and equations for $d\bar{T}_u/d\theta$, $d\bar{T}_b/d\theta$, $dm_b/d\theta$, and $dV_b/d\theta$ can be found in Ref. 10. Some examples of mass fraction burned curves obtained from measured pressure data, with gasoline and methanol fuels, are shown in Fig. 9-8. With accurate pressure versus crank angle records, values of final mass fraction burned should be close to but lower than unity, usually in the range 0.93 to 0.98; the difference from unity is the combustion inefficiency for lean mixtures (see Fig. 3-9) and incomplete oxygen utilization for rich mixtures (see Fig. 4-20).

More accurate burned gas temperature calculations need to account for the presence of a thermal boundary layer (of order 1 mm thick) around the combustion chamber walls (see Sec. 12.6.5). The burned gas region in Fig. 9-4 can be

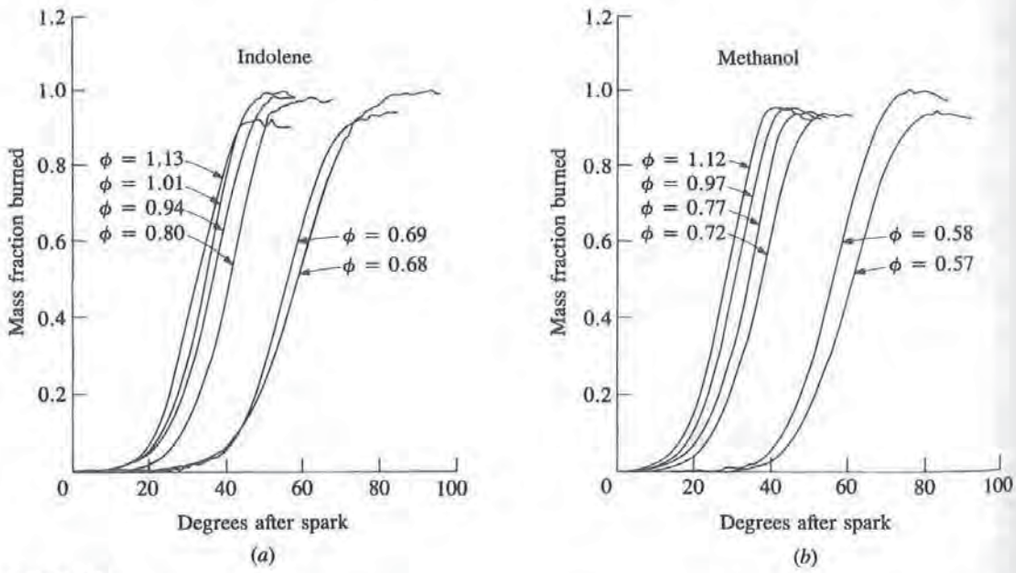


FIGURE 9-8
Mass fraction burned curves determined from measured cylinder pressure data using two-zone combustion model: (*a*) gasoline; (*b*) methanol. ϕ = fuel/air equivalence ratio.[11]

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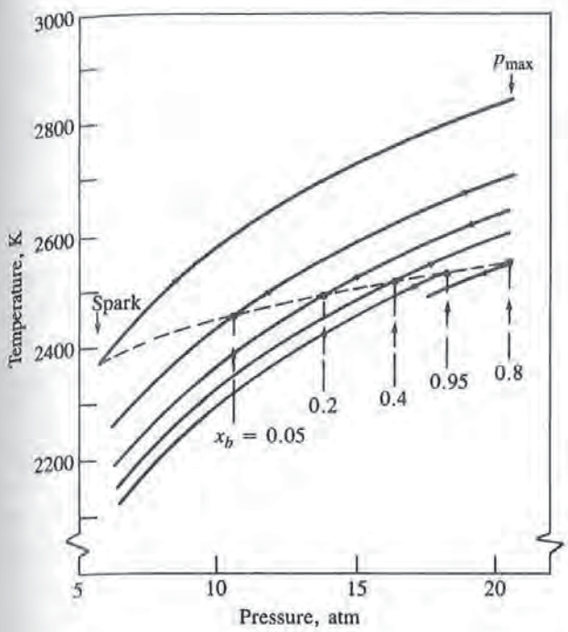


FIGURE 9-9
Calculated temperature distribution in the adiabatic core of the burned gas zone for the unmixed model assuming thermodynamic equilibrium. $\phi = 1.0$. Dashed line is temperature of each element just after it burns.

divided into an adiabatic core and a boundary layer that grows in thickness with time. In the adiabatic core, in the absence of mixing between gas elements that burn at different times, burned gas is compressed and then expanded isentropically. The burned gas temperature distribution can be calculated as follows. Given the pressure versus crank angle data, the unburned mixture state can be determined using Eq. (9.18) above. Each small element of unburned mixture burns in a constant-enthalpy constant-pressure process. So the burned state of an element of unburned charge, which burns at $p = p_i$, can be obtained from the relation

$$h_b(T_{b,i}, p_i) = h_u(T_{u,i}, p_i)$$

After combustion, this element which burned at $p = p_i$ is compressed and expanded along the isentropic:

$$s_b(T_b, p) = s_b(T_{b,i}, p_i)$$

An example of the temperature distribution computed in this manner for this unmixed model in the burned gas adiabatic core is shown in Fig. 9-9. The element ignited by the spark is compressed to the highest peak temperature at p_{max}. The temperature difference across the bulk of the charge ($0.05 < x_b < 0.95$) is about 200 K.

9.2.2 Analysis of Cylinder Pressure Data

Cylinder pressure changes with crank angle as a result of cylinder volume change, combustion, heat transfer to the chamber walls, flow into and out of crevice regions, and leakage. The first two of these effects are the largest. The effect of

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volume change on the pressure can readily be accounted for; thus, combustion rate information can be obtained from accurate pressure data provided models for the remaining phenomena can be developed at an appropriate level of approximation. The previous section has developed the fundamental basis for such calculations.

Cylinder pressure is usually measured with piezoelectric pressure transducers. This type of transducer contains a quartz crystal. One end of the crystal is exposed through a diaphragm to the cylinder pressure; as the cylinder pressure increases, the crystal is compressed and generates an electric charge which is proportional to the pressure. A charge amplifier is then used to produce an output voltage proportional to this charge. Accurate cylinder pressure versus crank angle data can be obtained with these systems provided the following steps are carried out: (1) the correct reference pressure is used to convert the measured pressure signals to absolute pressures; (2) the pressure versus crank angle (or volume) phasing is accurate to within about 0.2°; (3) the clearance volume is estimated with sufficient accuracy; (4) transducer temperature variations (which can change the transducer calibration factor) due to the variation in wall heat flux during the engine cycle are held to a minimum. Log p versus log V plots can be used to check the quality of cylinder pressure data. The first three of the above requirements can be validated using log p–log V diagrams for a motored engine. If the effects of thermal cycling are significant, the expansion stroke on the log p–log V plot for a firing engine shows excessive curvature.[12]

Figure 9-10 shows pressure-volume data from a firing spark-ignition engine on both a linear p–V and a log p–log V diagram. On the log p–log V diagram the compression process is a straight line of slope 1.3. The start of combustion can be identified by the departure of the curve from the straight line. The end of com-

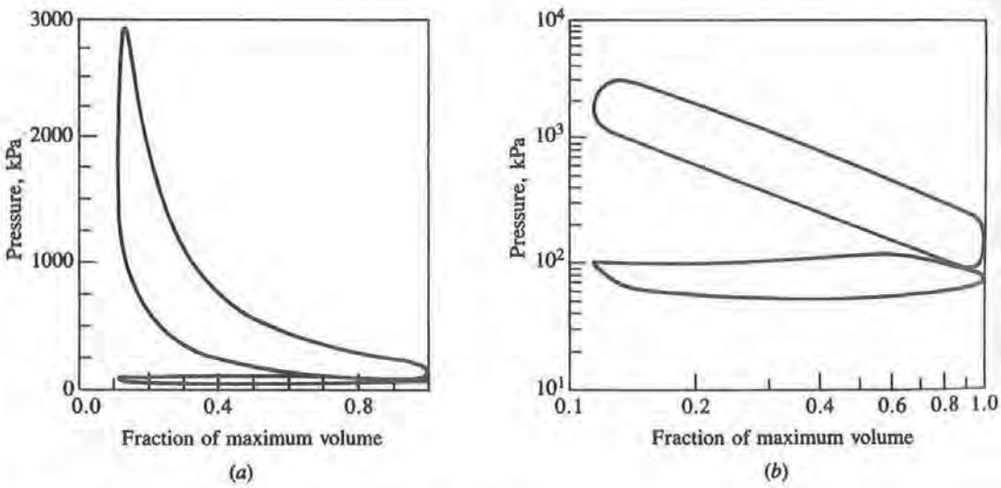


FIGURE 9-10
(*a*) Pressure-volume diagram; (*b*) log p–log (V/V_{max}) plot. 1500 rev/min, MBT timing, imep = 513 kPa, $\phi = 0.8$, $r_c = 8.72$, propane fuel.[12]

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bustion can be located approximately in similar fashion; the expansion stroke following combustion is essentially linear with slope 1.33. Since both the compression of the unburned mixture prior to combustion and the expansion of the burned gases following the end of combustion are close to adiabatic isentropic processes (for which $pV^\gamma = \text{constant}$; $\gamma = c_p/c_v$), the observed behavior is as expected. More extensive studies[12, 13] show that the compression and expansion processes are well fitted by a polytropic relation:

$$pV^n = \text{constant} \tag{9.20}$$

The exponent n for the compression and expansion processes is 1.3 (± 0.05) for conventional fuels. It is comparable to the average value of γ_u for the unburned mixture over the compression process, but is larger than γ_b for the burned gas mixture during expansion due to heat loss to the combustion chamber walls (see Figs. 4-13 and 4-16).

Log p–log V plots such as Fig. 9-10 approximately define the start and end of combustion, but do not provide a mass fraction burned profile. One well-established technique for estimating the mass fraction burned profile from the pressure and volume data is that developed by Rassweiler and Withrow.[2] They correlated cylinder pressure data with flame photographs, and showed how Eq. (9.20) could be used to account for the effect of cylinder volume change on the pressure during combustion. Assuming that the unburned gas filling the volume V_u ahead of the flame at any crank angle during combustion has been compressed polytropically by the advancing flame front, then the volume $V_{u,0}$ it occupied at time of spark is

$$V_{u,0} = V_u\left(\frac{p}{p_0}\right)^{1/n} \tag{9.21}$$

Similarly, the burned gas behind the flame filling the volume V_b would, at the end of combustion, fill a volume $V_{b,f}$ given by

$$V_{b,f} = V_b\left(\frac{p}{p_f}\right)^{1/n} \tag{9.22}$$

The mass fraction burned x_b is equal to $1 - (V_{u,0}/V_0)$ and to $V_{b,f}/V_f$, where V_0 amd V_f are the total cylinder volumes at time of spark and at the end of combustion, respectively. Since $V = V_u + V_b$, Eqs. (9.21) and (9.22) then give:

$$x_b = \frac{p^{1/n}V - p_0^{1/n}V_0}{p_f^{1/n}V_f - p_0^{1/n}V_0} \tag{9.23}$$

This method is widely used, though it contains several approximations. Heat-transfer effects are included only to the extent that the polytropic exponent n in Eq. (9.22) differs appropriately from γ. The pressure rise due to combustion is proportional to the amount of fuel chemical energy released rather than the mass of mixture burned. Also, the polytropic exponent n is not constant during combustion. Selecting an appropriate value for n (whether n is assumed to be con-

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stant or to vary through the combustion process) is the major difficulty in applying this pressure data analysis procedure.

The effects of heat transfer, crevices, and leakage can be explicitly incorporated into cylinder pressure data analysis by using a "heat release" approach based on the first law of thermodynamics. A major advantage of such an approach is that the pressure changes can be related directly to the amount of fuel chemical energy released by combustion, while retaining the simplicity of treating the combustion chamber contents as a single zone. Figure 9-11 shows the appropriate open-system boundary for the combustion chamber.[14] The first law for this open system is

$$\delta Q_{\text{ch}} = dU_s + \delta Q_{\text{ht}} + \delta W + \sum h_i \, dm_i \tag{9.24}$$

The change in sensible energy of the charge dU_s is separated from that due to change in composition: the term δQ_{ch} represents the "chemical energy" released by combustion. The work is piston work and equal to $p\,dV$. δQ_{ht} is heat transfer to the chamber walls. The mass flux term represents flow across the system boundary. In the absence of fuel injection, it represents flow into and out of crevice regions (see Sec. 8.6).

The accuracy with which this energy balance can be made depends on how adequately each term in the above equation can be quantified. Assuming that U_s is given by $mu(T)$, where T is the mean charge temperature and m is the mass within the system boundary, then

$$dU_s = mc_v(T)\,dT + u(T)\,dm$$

Note that this mean temperature determined from the ideal gas law is close to the mass-averaged cylinder temperature during combustion since the molecular weights of the burned and unburned gases are essentially identical. Crevice effects can usually be modeled adequately by flow into and out of a single volume at the cylinder pressure, with the gas in the crevice at a substantially lower temperature. Leakage to the crankcase can usually be neglected. Then Eq. (9.24), on substituting for dU_s and dm_i ($=dm_{\text{cr}} = -dm$), becomes

$$\delta Q_{\text{ch}} = mc_v\,dT + (h' - u)dm_{\text{cr}} + p\,dV + \delta Q_{\text{ht}} \tag{9.25}$$

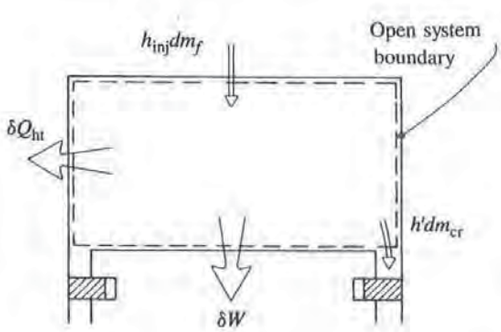


FIGURE 9-11
Open system boundary for combustion chamber for heat-release analysis.

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where

$dm_{cr} > 0$ when flow is out of the cylinder into the crevice
$dm_{cr} < 0$ when flow is from the crevice to the cylinder
h' is evaluated at cylinder conditions when $dm_{cr} > 0$ and
at crevice conditions when $dm_{cr} < 0$

Use of the ideal gas law (neglecting the change in gas constant R) with Eq. (9.25) then gives

$$\delta Q_{ch} = \left(\frac{c_v}{R}\right) V\, dp + \left(\frac{c_v}{R} + 1\right) p\, dV + (h' - u + c_v T)\, dm_{cr} + \delta Q_{ht} \qquad (9.26)$$

This equation can be used in several ways. When the heat or energy release term, δQ_{ch}, is combined with the heat-transfer and crevice terms, the combination is termed *net heat release*—the combustion energy release less heat lost to the walls. It is equal to the first two terms on the right-hand side of Eq. (9.26) which, together, represent the sensible energy change and work transfer to the piston. While heat losses during combustion are a small fraction of the fuel energy (10 to 15 percent), the distributions of heat release and heat transfer with crank angle are different; heat transfer becomes more important as the combustion process ends and average gas temperatures peak. The net heat-release profile obtained from integrating the first two terms on the right-hand side of Eq. (9.26), normalized to give unity at its maximum value, is often interpreted as the burned mass fraction (or, more correctly, the energy-release fraction) versus crank angle profile.

Use of Eq. (9.26) requires a value for c_v/R $[= 1/(\gamma - 1)]$. The ratio of specific heats γ for both unburned and burned gases decreases with increasing temperature and varies with composition (see Figs. 4-13, 4-16, and 4-18). As the mean charge temperature increases during compression and combustion and then decreases during expansion, γ should vary. An approximate approach, modeling $\gamma(T)$ with a linear function of temperature fitted to the appropriate curves in Figs. 4-13, 4-16, and 4-18 and with γ constant during combustion, has been shown to give adequate results.[15]

The convective heat-transfer rate to the combustion chamber walls can be calculated from the relation

$$\frac{dQ_{ht}}{dt} = A h_c (T - T_w)$$

where A is the chamber surface area, T is the mean gas temperature, T_w is the mean wall temperature, and h_c is the heat-transfer coefficient (averaged over the chamber surface area). h_c can be estimated from engine heat-transfer correlations (see Sec. 12.4.3). Since crevice effects are usually small, a sufficiently accurate model for their overall effect is to consider a single aggregate crevice volume where the gas is at the same pressure as the combustion chamber, but at a different temperature. Since these crevice regions are narrow, an appropriate assumption is that the crevice gas is at the wall temperature. Inserting this crevice model

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into Eq. (9.26), with $\gamma(T) = a + bT$, gives the *chemical energy-* or *gross heat-release rate*:

$$\frac{dQ_{ch}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + V_{cr}\left[\frac{T'}{T_w} + \frac{T}{T_w(\gamma - 1)} + \frac{1}{bT_w} \ln\left(\frac{\gamma - 1}{\gamma' - 1}\right)\right] \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \quad (9.27)$$

An example of the use of Eq. (9.27) to analyze an experimental pressure versus crank angle curve for a conventional spark-ignition engine is shown in Fig. 9-12. The integrated heat release is plotted against crank angle. The lowest curve shown is the net heat release. The addition of heat transfer and crevice models gives the chemical energy release. The curve at the top of the figure is the mass of fuel within the combustion chamber times its lower heating value. It decreases slightly as p_{max} is approached due to flow into crevices. The difference between the final value of Q_{ch} and $(m_f Q_{LHV})$ should equal the combustion inefficiency (which is a few percent of $m_f Q_{LHV}$). The combustion inefficiency can be determined from the exhaust gas composition (see Sec. 4.9.4). Inaccuracies in the cylinder pressure data and the heat-release calculation will also contribute to this difference. An important advantage of a heat-release analysis that relates the pressure changes to the amount of fuel chemical energy within the cylinder is that this error can be determined. In the example in Fig. 9-12, the measured combustion inefficiency was close to the amount shown.

Two-zone models (one zone representing the unburned mixture ahead of the flame and one the burned mixture behind the flame) are used to calculate the mass fraction burned profile from measured cylinder pressure data.[10] Figure 9-8 shows results from such an analysis, using the methodology described in Sec. 9.2.1. The advantage of a two-zone analysis is that the thermodynamic properties of the cylinder contents can be quantified more accurately. The disadvantages are that the unburned and the burned zone heat-transfer areas must both now be estimated, and a model for the composition of the gas flowing into the crevice

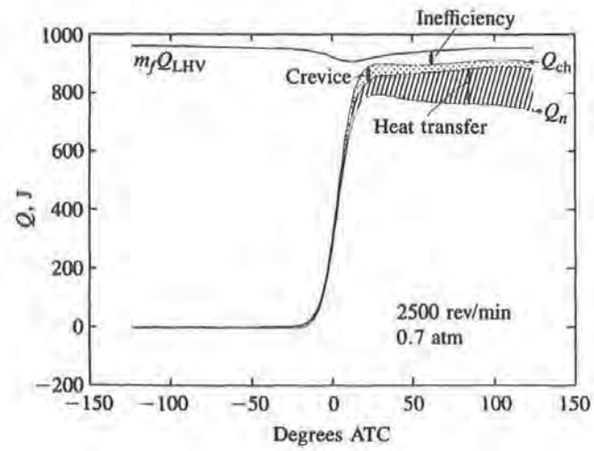


FIGURE 9-12
Results of heat-release analysis showing the effects of heat transfer, crevices, and combustion inefficiency.[14]

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region must be developed. Due to this complexity, crevice models are usually omitted despite the fact that their impact can be significant.

9.2.3 Combustion Process Characterization

The mass fraction burned profiles as a function of crank angle in each individual cycle shown in Fig. 9-2 and the chemical energy- or gross heat-release curve in Fig. 9-12 have a characteristic S-shape. The rate at which fuel-air mixture burns increases from a low value immediately following the spark discharge to a maximum about halfway through the burning process and then decreases to close to zero as the combustion process ends. It proves convenient to use these mass fraction burned or energy-release fraction curves to characterize different stages of the spark-ignition engine combustion process by their duration in crank angles, thereby defining the fraction of the engine cycle that they occupy. The flame development process, from the spark discharge which initiates the combustion process to the point where a small but measurable fraction of the charge has burned, is one such stage. It is influenced primarily by the mixture state, composition, and motion in the vicinity of the spark plug (see Sec. 9.3). The stage during which the major portion of the charge burns as the flame propagates to the chamber walls is next. This stage is obviously influenced by conditions throughout the combustion chamber. The final stage, where the remainder of the charge burns to completion, cannot as easily be quantified because energy-release rates are comparable to other energy-transfer processes that are occurring.

The following definitions are most commonly used to characterize the energy-release aspects of combustion:

Flame-development angle $\Delta\theta_d$. The crank angle interval between the spark discharge and the time when a small but significant fraction of the cylinder mass has burned or fuel chemical energy has been released. Usually this fraction is 10 percent, though other fractions such as 1 and 5 percent have been used.†

Rapid-burning angle $\Delta\theta_b$. The crank angle interval required to burn the bulk of the charge. It is defined as the interval between the end of the flame-development stage (usually mass fraction burned or energy-release fraction of 10 percent) and the end of the flame-propagation process (usually mass fraction burned or energy-release fraction of 90 percent).‡

Overall burning angle $\Delta\theta_o$. The duration of the overall burning process. It is the sum of $\Delta\theta_d$ and $\Delta\theta_b$.

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† This angle is sometimes called the *ignition delay*. Since the flame starts to propagate outward immediately following the spark discharge there is no delay, and the terminology used here is preferred (see Sec. 9.3).

‡ An alternative definition for $\Delta\theta_b$ uses the maximum burning rate to define an angle or time characteristic of the bulk charge burning process[4] (see Fig. 9-13). $\Delta\theta_b$ and $\Delta\theta_b^*$ are usually closely comparable.

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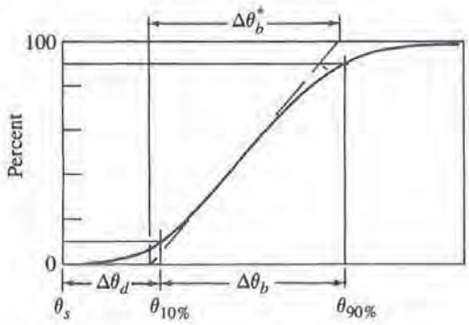


FIGURE 9-13
Definition of flame-development angle, $\Delta\theta_d$, and rapid-burning angle, $\Delta\theta_b$, on mass fraction burned versus crank angle curve.

Figure 9-13 illustrates these definitions on a mass fraction burned, or fraction of fuel energy released, versus crank angle plot. While the selection of the 10 and 90 percent points is arbitrary, such a choice avoids the difficulties involved in determining accurately the shape of the curve at the start and end of combustion. These angles can be converted to times (in seconds) by dividing by $6N$ (with N in revolutions per minute).

A functional form often used to represent the mass fraction burned versus crank angle curve is the Wiebe function:

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{9.28}$$

where θ is the crank angle, θ_0 is the start of combustion, $\Delta\theta$ is the total combustion duration ($x_b = 0$ to $x_b = 1$), and a and m are adjustable parameters. Varying a and m changes the shape of the curve significantly. Actual mass fraction burned curves have been fitted with $a = 5$ and $m = 2$.[16]

9.3 FLAME STRUCTURE AND SPEED

9.3.1 Experimental Observations

The combustion process in the spark-ignition engine takes place in a turbulent flow field. This flow field is produced by the high shear flows set up during the intake process and modified during compression, as described in Chap. 8. The importance of the turbulence to the engine combustion process was recognized long ago through experiments where the intake event, and the turbulence it generates, was eliminated and the rate of flame propagation decreased substantially. Understanding the structure of this engine flame as it develops from the spark discharge and the speed at which it propagates across the combustion chamber, and how that structure and speed depend on charge motion, charge composition, and chamber geometry, are critical to engine optimization. This section reviews experimental evidence that describes the essential features of the flame development and propagation processes.

Direct flame photographs such as those in Fig. 9-1 indicate the location and shape of the actual reaction zone which radiates in the blue region of the visible spectrum. An irregular front is apparent. Further insight into the structure of the

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flame can be obtained from photographs taken with techniques that are sensitive to density changes in the flow field, such as schlieren and shadowgraph. With these techniques, a parallel light beam is passed through the combustion chamber. Portions of the beam which pass through regions where density gradients normal to the beam exist are deflected, due to the refractive index gradients that result from the density gradients. In the schlieren technique, the beam is focused on a knife edge; the deflected parts of the beam are displaced relative to the knife edge and produce lighter or darker regions when subsequently refocused onto film. With the shadowgraph technique, the parallel beam emerging from the combustion chamber is photographed directly; deflected parts of the beam produce lighter and darker regions on the film. With these techniques, details of flame structure can be discerned.

Figure 9-14 shows a set of photographs from one engine cycle, from a high-speed schlieren movie taken in a special visualization spark-ignition engine operating at 1400 rev/min and 0.5 atm inlet pressure. Also shown are the cylinder pressure versus crank angle data, and the mass fraction burned profile calculated from the pressure data using the method of Rassweiler and Withrow[2] (see Sec. 9.2.2). This engine had a square-cross-section cylinder with two quartz walls to permit easy optical access, but otherwise operated normally.[17] Visualization of

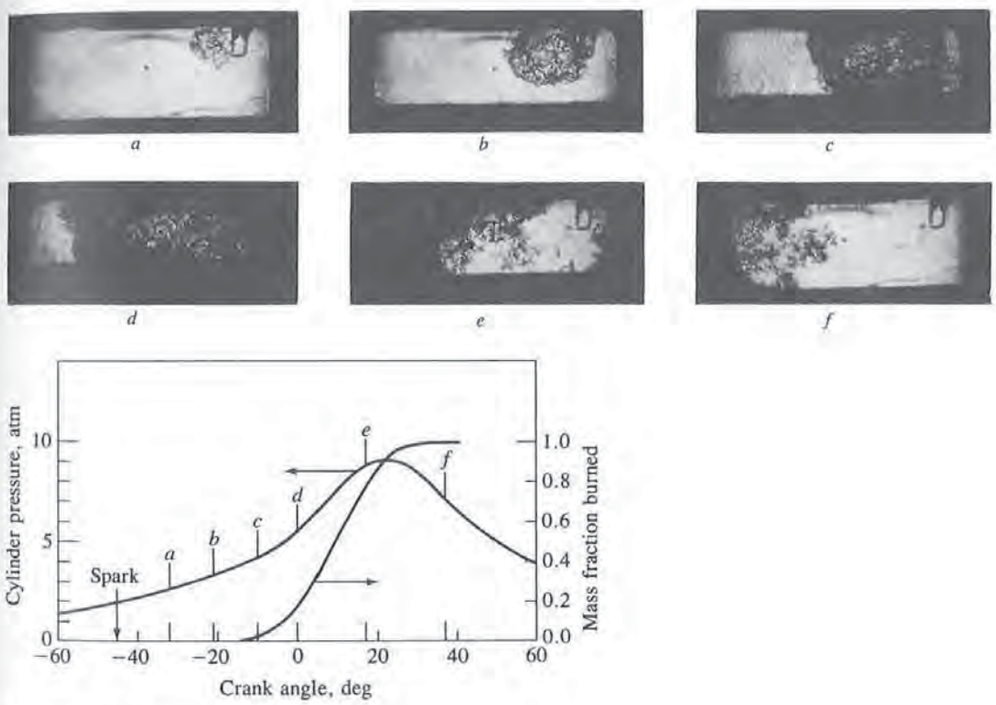


FIGURE 9-14
Sequence of movie frames from one engine cycle in a square-cross-section cylinder, single-cylinder, engine with two glass walls, and corresponding pressure and mass fraction burned curves. 1400 rev/min, 0.5 atm inlet pressure.

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the flame is especially important during the early stages of flame development when the pressure rise due to combustion is too small to be detected.

These photographs show how the flame "ball," roughly spherical in shape, grows steadily from the time of spark discharge. The effect of turbulence is already visible in the convoluted flame surface in Fig. 9-14*a*. The volume enflamed behind the front continues to grow in a roughly spherical manner, except where intersected by the chamber walls, as seen in Fig. 9-14*b* and *c*. The mass fraction burned and the associated pressure rise due to combustion become significant by the time the flame front has traversed two-thirds to three-quarters of the field of view. Note that the fraction of the cylinder filled with enflamed charge is less than is suggested by the photos because the front of the flame is approximately spherical and the cylinder has a square cross section. Maximum cylinder pressure occurs close to the time the flame makes contact with the far wall, as seen in Fig. 9-14*e*. Finally, the unburned mixture ahead of and within the front burns out and the density gradients associated with the flame reaction zone disappear, clearing the field of view as shown in Fig. 9-14*e* and *f*.

A useful relationship between the mass fraction burned, $x_b(=m_b/m)$, and the volume fraction occupied by the burned gas, $y_b(=V_b/V)$, can be obtained from the identities

$$m = m_u + m_b \qquad V = V_u + V_b$$

and the ideal gas law:

$$x_b = \left[1 + \frac{\rho_u}{\rho_b}\left(\frac{1}{y_b} - 1\right)\right]^{-1} \tag{9.29}$$

While the density ratio (ρ_u/ρ_b) does depend on the equivalence ratio, burned gas fraction in the unburned mixture, gas temperature, and pressure, its value is close to 4 for most spark-ignition engine operating conditions. Thus, the plot of x_b against y_b has a universal form,[10] as shown in Fig. 9-15. This curve is an important aid in interpreting flame geometry information.

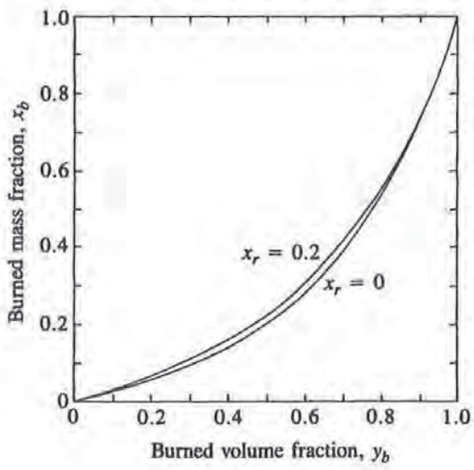


FIGURE 9-15
Relation between mass fraction burned x_b and volume fraction burned y_b. x_r is residual mass fraction.[4]

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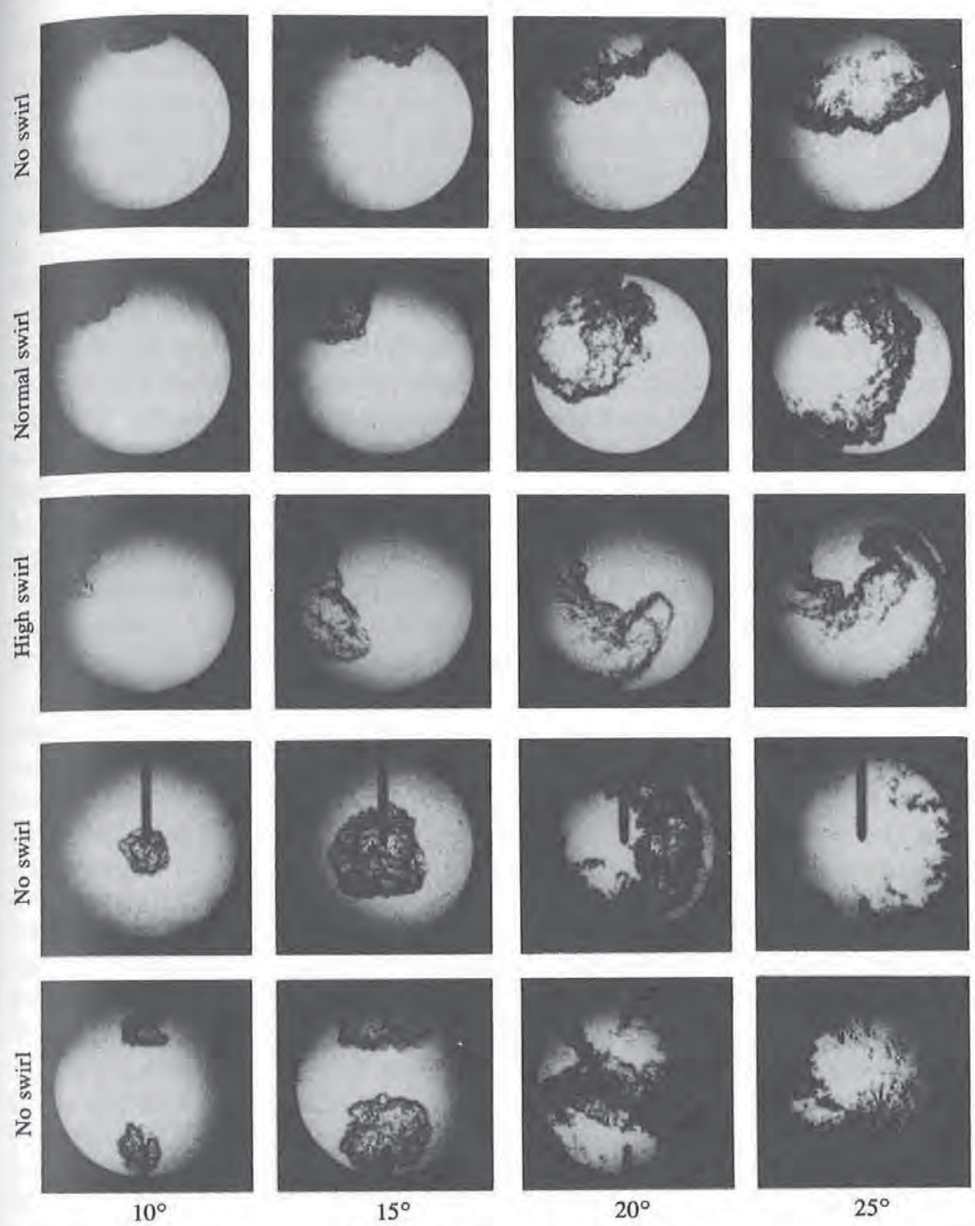


FIGURE 9-16
Laser shadowgraph photographs of engine combustion process taken in single-cylinder engine with transparent cylinder head. From top to bottom: side plug without swirl; side plug with normal swirl; side plug with high swirl; central plug without swirl; two plugs without swirl.[18]

The above-described features of the developing and propagating flame are common to almost all engine geometries and operating conditions. Figure 9-16 shows shadowgraph photographs of the flame at fixed crank angle intervals after ignition, taken through a transparent cylinder head with different geometric and flow configurations.[18] The approximately spherical development of the flame

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from the vicinity of the spark plug, except where it intercepts the chamber walls, is evident for side and center ignition with one plug, and for ignition with two plugs in the absence of any intake generated swirl. With normal levels of swirl, the flame center is convected with the swirling flow, but the flame front as it grows is still approximately spherical in shape. Only with unusually high levels of swirl and aerodynamic stabilization of the flame at the spark plug location does the flame become stretched out and distorted by the flow in a major way.[18]

At any given flame radius, the geometry of the combustion chamber and the spark plug location govern the flame front surface area—the area of the approximately spherical surface corresponding to the leading edge of the flame contained by the piston, cylinder head, and cylinder wall. The larger this surface area, the greater the mass of fresh charge that can cross this surface and enter the flame zone. The photos in Fig. 9-16 illustrate the importance of flame area. The center plug location gives approximately twice the flame area of the side plug geometry at a given flame radius, and burns about twice as fast (the fraction of the cylinder volume enflamed is about twice the size, at a fixed crank angle interval after spark). The arrangement with two spark plugs at opposite sides of the chamber is not significantly different in enflamed volume from the single center plug because, once the flame fronts are intersected by the cylinder wall, the flame front areas are comparable.

Mixture burning rate is strongly influenced by engine speed. It is well established that the duration of combustion in crank angle degrees only increases slowly with increasing engine speed.[19] Figure 9-17 shows how the interval between the spark discharge and 10 percent mass fraction burned, the flame development angle $\Delta\theta_d$, and the interval between the spark and 90 percent mass fraction burned, the overall burning angle $\Delta\theta_d + \Delta\theta_b$ (see Sec. 9.2.3), vary with engine speed.[20] Both intervals increase by a factor of about 1.6 for a factor of 4 increase in engine speed; i.e., the burning rate throughout the combustion process increases almost, though not quite, as rapidly as engine speed. Additionally, at a given engine speed, increasing in-cylinder gas velocities (e.g., with intake generated swirl) increases the burning rate: the flame size for the swirling flows in Fig. 9-16 is larger than for the quiescent case with the same plug location at the crank angle intervals after spark shown. Increasing engine speed and introducing swirl both increase the levels of turbulence in the engine cylinder at the time of combustion (see Sec. 8.2.2). Increased turbulence increases the rate of development and propagation of the turbulent premixed engine flame.

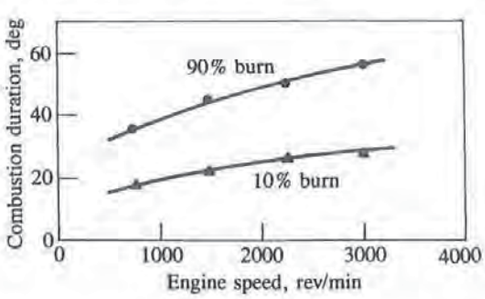


FIGURE 9-17
Effect of engine speed on flame-development angle (0 to 10 percent burned) and overall burning angle (0 to 90 percent burned). $\phi = 1.0$, intake pressure 0.54 atm, spark 30° BTC.[20]

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It is also well established that unburned mixture composition and state affect the burning rate. Reducing the inlet pressure (and maintaining the ratio of exhaust to inlet pressure fixed to hold the residual gas fraction constant) increases both the flame development and rapid burning angles.[19] The fuel/air equivalence ratio affects the burning rate. Both flame development and burning angles show a minimum for slightly rich mixtures ($\phi \approx 1.2$) and increase significantly as the mixture becomes substantially leaner than stoichiometric.[19, 20] The burned gas fraction in the unburned mixture, due to the residual gas fraction *and* any recycled exhaust gases, affects the burning rate: increasing the burned gas fraction slows down both flame development and propagation.[20] Fuel composition changes can be significant also. While mixtures of isooctane or conventional gasolines with air and burned gases (at identical conditions) have closely comparable burning rates, propane, methane, methanol, and ethanol mixtures exhibit modest differences in burning rate and hydrogen-air mixtures substantial differences. The basic combustion chemistry of the fuel, air, burned gas mixture influences the combustion process. However, the relative importance of combustion chemistry effects depends on combustion chamber design and burn rate. Faster burning engines (which have higher turbulence) are less sensitive to changes in mixture composition, pressure, and temperature than are slower burning engines (which have lower turbulence). The effects of chamber geometry, gas motion, and gas composition and state are interrelated.[21]

9.3.2 Flame Structure

Laminar flames in premixed fuel, air, residual gas mixtures are characterized by a laminar flame speed S_L and a laminar flame thickness δ_L (see Sec. 9.3.3). The laminar flame speed is the velocity at which the flame propagates into quiescent premixed unburned mixture ahead of the flame. There are several ways to define the thickness of a laminar flame.[22] Given the molecular diffusivity D_L (see Sec. 4.8), dimensional arguments give the most commonly used definition: $\delta_L = D_L/S_L$. Turbulent flames are also characterized by the root mean square velocity fluctuation, the turbulence intensity u' [Eq. (8.3)], and the various length scales of the turbulent flow ahead of the flame. The integral length scale l_I [Eq. (8.8)] is a measure of the size of the large energy-containing structures of the flow. The Kolmogorov scale l_K [Eq. (8.11)] defines the smallest structures of the flow where small-scale kinetic energy is dissipated via molecular viscosity.

Several dimensionless parameters are used to characterize turbulent premixed flames. The dimensionless parameter used to define the turbulence is the turbulent Reynolds number, $\mathrm{Re}_T = u' l_I/\nu$. For homogeneous and isotropic (no preferred direction) turbulence, the integral and Kolmogorov scales are related by Eq. (8.14): $l_K/l_I = \mathrm{Re}_T^{-3/4}$. A characteristic turbulent eddy turnover time τ_T can be defined as

$$\tau_T = \frac{l_I}{u'}$$

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A characteristic chemical reaction time is the residence time in a laminar flame:

$$\tau_L = \frac{\delta_L}{S_L}$$

The ratio of the characteristic eddy turnover time to the laminar burning time is called the *Damköhler number*:

$$\mathrm{Da} = \frac{\tau_T}{\tau_L} = \left(\frac{l_I}{\delta_L}\right)\left(\frac{S_L}{u'}\right) \tag{9.30}$$

It is an inverse measure of the influence of the turbulent flow on the chemical processes occurring in the flame. Other ratios are of interest. The ratio δ_L/l_K is a measure of the stretch or local distortion to which a laminar flame is subjected by the turbulent flow. Unless $l_I/\delta_L \gg 1$ the concept of a localized flame region has little significance. The ratio u'/S_L is a measure of the relative strength of the turbulence.[22]

Different regimes of turbulent flames are apparent in the plot of Damköhler number versus turbulent Reynolds number in Fig. 9-18.[22] It has been assumed that $D_L \approx \nu$ and that the relationships for homogeneous isotropic turbulence are valid. Two regimes—distributed reactions and reaction sheets—are normally identified. In the distributed reaction regime, chemical reactions proceed in distributed reaction zones and thin-sheet flames do not occur. A sufficient condition for this regime is $l_I \ll \delta_L$. In the reaction sheet regime, propagating reaction fronts are wrinkled and convoluted by the turbulence. A sufficient condition for the existence of reaction sheets is $l_K \gg \delta_L$. For $\mathrm{Re}_T > 1$, there is a region in Fig. 9-18 where $l_I > \delta_L > l_K$: the characteristics of flames in this regime are unclear.

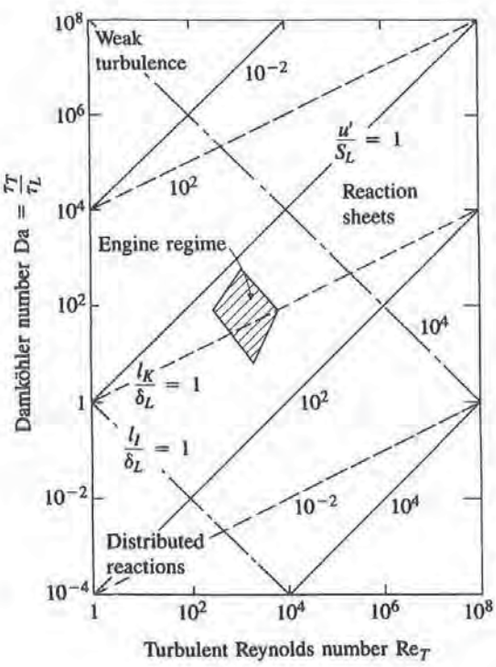


FIGURE 9-18
Different turbulent flame regimes on plot of Damköhler number versus turbulence Reynolds number, u' is turbulence intensity; S_L is laminar flame speed; l_I, l_K, and δ_L are integral scale, Kolmogorov scale, and laminar flame thickness. (*From Abraham* et al.[22])

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Values of Da and Re_T for a typical spark-ignition engine (the cross-hatched region in Fig. 9-18) lie predominantly in the reaction sheet flame regime. Engine operation at high speed (the lower right boundary) and low load (the lower left boundary) gives values of Da and Re_T which fall below the $l_K/\delta_L = 1$ line. This is largely due to the low values of laminar flame speed that result from the high amounts of residual gas and EGR under these conditions (see Sec. 9.3.4 and Fig. 6-19). Whether the flame structure under these conditions is significantly different is not known. Observations of engine flames to date, described below, lie above the $l_K/\delta_L = 1$ line, within the reaction sheet regime. One would expect, then, the structure of the flame in a spark-ignition engine, once developed, to be that of a thin reaction sheet wrinkled and convoluted by the turbulent flow.

Detailed observations have been made of flame structure from ignition to flame extinguishing at the far cylinder wall. A flame develops from the spark discharge which causes ignition as follows. In the initial breakdown phase of ignition, a cylindrical discharge between the spark plug electrodes is established.[23] As electrical energy is fed into the discharge, the arc expands and exothermic chemical reactions capable of sustaining a propagating flame develop. Figure 9-19 shows how this development of a flame kernel occurs, with a set of shadowgraph photographs taken at 40-μs intervals of the spark plug electrode gap in one cylinder of a 2-liter conventional engine. The first photograph is between 20 and 50 μs after the spark breakdown occurred. The complete sequence ($\sim$200 μs) corresponds to 1.3 crank angle degrees. The outer boundary of this developing flame kernel is approximately spherical and is smooth with modest irregularities, corresponding to a thin reaction zone with high-temperature gases inside. As this developing sheetlike flame grows it interacts with the turbulent flow field in the vicinity of the spark plug: the flame outer surface becomes increasingly convoluted and the flame center can be convected away from the plug in a direction and with a velocity that can vary substantially cycle-by-cycle, as seen in Fig. 9-20.[4, 17]

The structure of the flame continues to develop as it propagates across the chamber. Evidence, largely from schlieren photographs and studies of flame structure with laser diagnostics, shows that early in the burning process the flame

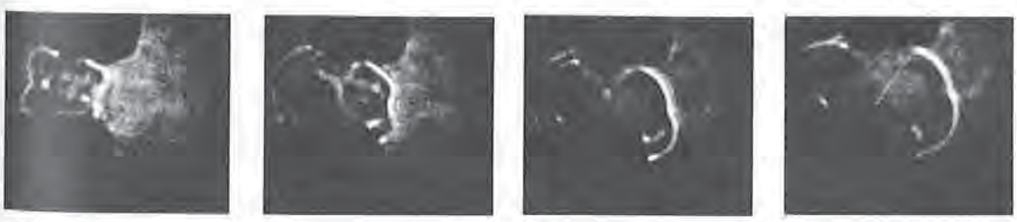


FIGURE 9-19
Shadowgraph photographs of spark-generated kernel between the spark plug electrodes. First photograph on left, 20 to 50 μs after breakdown; 40 μs between photos. Stoichiometric mixture, 1100 rev/min. (*Courtesy A. Douad, Institut Francais du Petrole.*)

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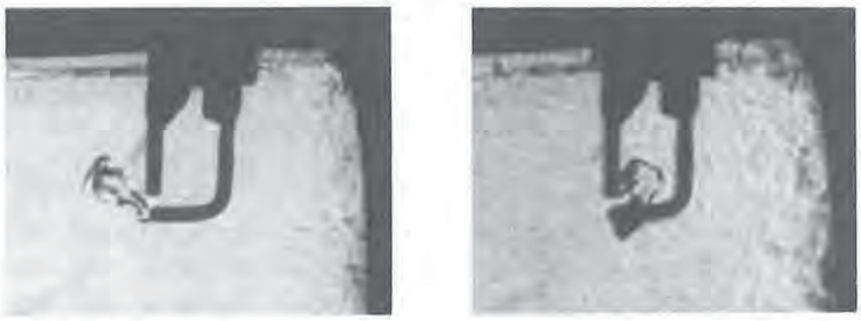


FIGURE 9-20
Schlieren photographs of developing flame, 5° after spark discharge, showing different convection processes in two different cycles. Spark plug wires 0.8 mm diameter. 1400 rev/min; 0.5 atm inlet pressure; propane fuel; $\phi = 0.9$; spark advance 45° BTC.[17]

is a thin, moderately wrinkled but simply connected, front or reaction sheet between unburned and burned gas. The thickness of the front is about 0.1 mm which is comparable to the thickness of a laminar flame under the prevailing conditions. The scale of the wrinkles is typically about 2 mm at engine speeds of 1000 to 2000 rev/min. As the flame propagates across the chamber, the thickness of the reaction sheet front remains roughly constant, the flame front becomes more convoluted, and the scale of the wrinkles tends to decrease with time.[24]

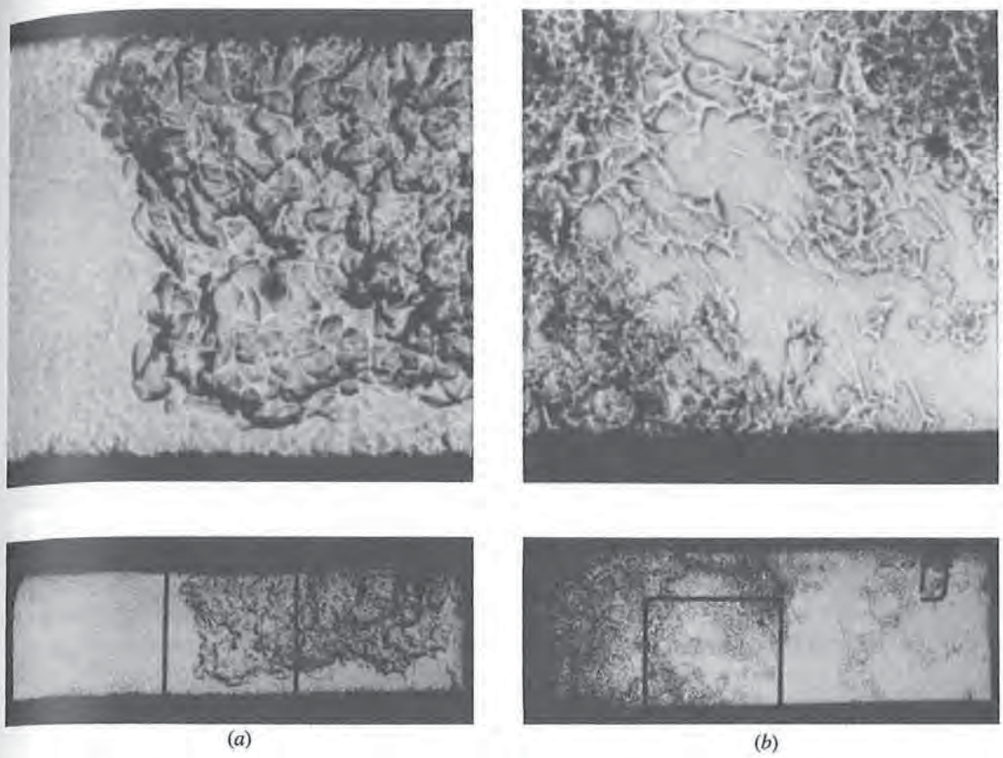
Further evidence that the thin reaction sheet front becomes highly wrinkled and convoluted by the turbulent flow field into a thick turbulent flame "brush" is provided by the schlieren photographs in Fig. 9-21. These show a flame propagating across the combustion chamber of a square-cross-section single-cylinder engine with a special transparent piston crown containing a 13-mm wide channel to isolate a small section of the flame.[25] The flame on the left is for a propane-air mixture; that on the right is hydrogen-air. The energy density per unit volume of mixture and the flow field are comparable for each fuel. The effective thickness of this turbulent flame—the average distance between the region ahead of the flame, where only unburned mixture exists, and the region behind the flame, where only



(*a*) (*b*)

FIGURE 9-21
Schlieren photographs of flame in square-cross-section cylinder engine, with narrow channel in piston crown (at bottom of pictures) which permits observation of 13-mm wide section of flame. (*a*) Propane fuel, spark timing 36° BTC, photograph at 14° ATC, flame thickness ≈ 4.6 mm. (*b*) Hydrogen fuel, spark timing 3° BTC, photograph at 10° ATC, flame thickness ≈ 1.5 mm. Stoichiometric mixtures, 1380 rev/min, 0.5 atm inlet pressure.[25]

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(*a*) (*b*)

FIGURE 9-22
Enlarged schlieren photographs of (*a*) flame front and (*b*) flame back in square-cross-section cylinder engine with two glass side walls. 1400 rev/min, 0.5 atm inlet pressure, propane fuel, $\phi = 0.9$.[17]

burned mixture exists—is apparent.† It was 4 to 5 mm for propane and 1.5 mm for hydrogen. The difference is due to the substantially higher laminar flame speed for the hydrogen, air, burned gas mixture (see Sec. 9.3.4) which increases the Damköhler number and shifts the flame toward the weak-turbulence (i.e., less wrinkled) flame regime in Fig. 9-18.

Additional insight into the structure of the developed engine flame can be obtained by enlarging photographs of the leading and trailing edges of the flame, obtained with the schlieren or shadowgraph technique. Figure 9-22*a* shows the front of the flame 40° after the spark, when it has propagated about halfway across the chamber. It shows the irregular but smoothly curved surfaces which comprise the leading edge of the flame. Figure 9-22*b* shows the back of the flame 70° after the spark, when the front of the flame has just reached the wall of the combustion chamber farthest from the spark plug. It shows large clear regions of burned gas behind the flame and smaller clear regions connected by a lacelike

† These photographs were selected from a large number to give the minimum flame thickness corresponding to the flame front perpendicular to the channel length.

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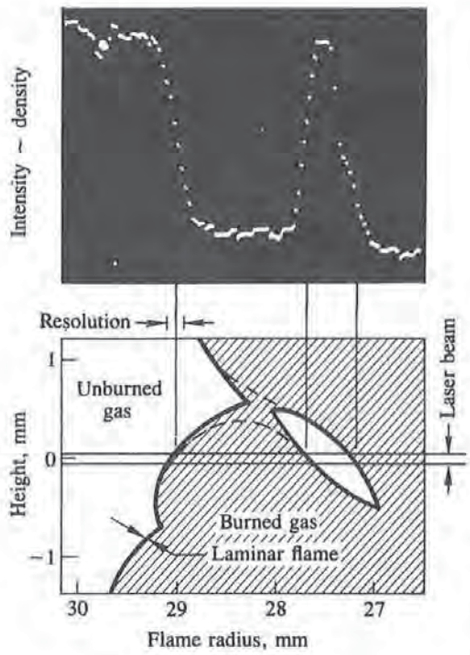


FIGURE 9-23
Upper picture: oscillogram of output of optical multichannel analyzer showing the intensity of light scattered from a narrow laser beam as a function of distance through the flame. Intensity is a measure of gas density. Lower picture: schematic of flame structure corresponding to this signal. 600 rev/min, 1 atm inlet pressure, propane fuel, $\phi = 1.0$.[24]

structure within which the remaining regions of unburned mixture are being consumed.[17] The analogy with a crumpled sheet of paper is appropriate.

Laser scattering experiments, where "snapshots" of the density profile along a laser beam passed through the flame were obtained using Rayleigh scattered light from the gas molecules, provide explicit evidence of this structure.[26] Figure 9-23 shows an oscillogram of the output of the optical analyzer. Each dot represents the intensity of the scattered light which is a measure of the gas density. The signal on the left corresponds to unburned gas. The flame is propagating from right to left. The oscillogram shows a thin transition zone of width 0.25 mm between unburned and burned gas, followed at a distance of 1.5 mm by an "island" of unburned gas. The fraction of oscillograms showing such "islands" varied from 0 at 300 rev/min to 20 percent at 1800 rev/min. An interpretation of this signal consistent with the available photographic evidence is shown underneath.[24]

The above results suggest that increasing engine speed, which increases turbulence levels in the unburned charge, increasingly convolutes and probably multiply-connects the thin reaction sheet flame front. Enlarged schlieren photographs of a 9-mm diameter section of the developed engine flame, viewed normal to the flame surface, indicate that increasing turbulence intensity and decreasing turbulence scales result in increasingly finely wrinkled flame structures. Figure 9-24 shows a set of such photographs, arranged in order of increasing turbulent Reynolds number. Increases in turbulence intensity, which was measured, were achieved by increases in engine speed and by modifying the inlet valve. Relevant parameters for each photograph are given in Table 9.1.

The above theoretical discussion and experimental evidence indicates that

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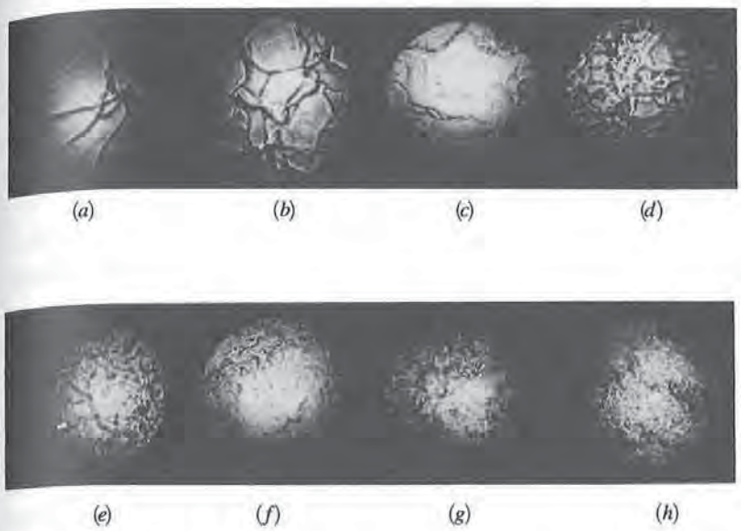


FIGURE 9-24
Shadowgraph photographs, with light beam normal to flame front, of 10-mm diameter section of flame. Photographs, arranged in order of increasing turbulent Reynolds number, suggest this is an appropriate scaling parameter (see Table 9.1).[27]

developed turbulent flames in spark-ignition engines, under normal operating conditions, are highly wrinkled and probably multiply-connected thin reaction sheets. The overall thickness of the turbulent flame "brush," front to back, is of order 1 cm. The thickness of the thin reaction sheet is comparable to estimates of the laminar flame thickness under the prevailing unburned mixture conditions which are of order 0.1 mm. The scale of the wrinkles is of order 1 mm.[24] Direct evidence to date is limited to the low to mid engine speed and low to high engine

TABLE 9.1
Parameters for shadowgraph photographs in Fig. 9-24

Photograph	Engine speed, rev/min	Turbulence intensity, m/s	Valve	Re_M	Re_T
a	300	0.44	S	106	229
b	600	0.88	S	157	503
c	300	1.07	US	173	611
d	900	1.33	S	193	760
e	1200	1.80	S	224	1024
f	600	1.95	US	234	1117
g	900	2.90	US	285	1658
h	1200	4.0	US	333	2263

Note: Valve: S, shrouded; US, unshrouded (produced higher turbulence due to less ordered flow). $Re_M = l_M u'/\nu$; $Re_T = l_I u'/\nu$ (see Sec. 8.2.1).[27]

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load ranges. Whether the structure becomes significantly different at high engine speed is not known. Models of this turbulent flame development and propagation process are reviewed in Chap. 14.

9.3.3 Laminar Burning Speeds

An important intrinsic property of a combustible fuel, air, burned gas mixture is its laminar burning velocity. This burning velocity is defined as the velocity, relative to and normal to the flame front, with which unburned gas moves into the front and is transformed to products under laminar flow conditions. Some details of flame structure help explain the significance of this quantity. A flame is the result of a self-sustaining chemical reaction occurring within a region of space called the flame front where unburned mixture is heated and converted into products. The flame front consists of two regions: a preheat zone and a reaction zone. In the preheat zone, the temperature of the unburned mixture is raised mainly by heat conduction from the reaction zone: no significant reaction or energy release occurs and the temperature gradient is concave upward ($\partial^2 T/\partial x^2 > 0$). Upon reaching a critical temperature, exothermic chemical reaction begins. The release of chemical energy as heat results in a zone where the temperature gradient is concave downward ($\partial^2 T/\partial x^2 < 0$). The region between the temperature where exothermic chemical reaction begins and the hot boundary at the downstream equilibrium burned gas temperature is called the reaction zone. The thicknesses of the preheat and reaction zones can be calculated for one-dimensional flames from conservation equations of mass and energy. The thickness of the preheat zone $\delta_{L,\,\mathrm{ph}}$ is

$$\delta_{L,\,\mathrm{ph}} = \frac{4.6\bar{k}}{\bar{c}_p \rho_u S_L} \tag{9.31}$$

where $\bar{k}$ and $\bar{c}_p$ are the mean thermal conductivity and specific heat at constant pressure in the preheat zone and S_L is the laminar burning velocity.[28] Thus, the factors which govern the laminar burning velocity of a specific unburned mixture—the velocity at which this flame structure propagates relative to the unburned gas ahead of it—are the temperature and species concentration gradients within the flame and the mixture transport and thermodynamic properties.

Laminar burning velocities at pressures and temperatures typical of unburned mixture in engines are usually measured in spherical closed vessels by propagating a laminar flame radially outward from the vessel center. The laminar burning velocity is then given by

$$S_L = \frac{dm_b/dt}{A_f \rho_u} \tag{9.32}$$

where the mass burning rate is determined from the rate of pressure rise in the vessel and A_f is the flame area. Because the laminar flame thickness [e.g., given by Eq. (9.31)] under engine conditions is of order 0.2 mm[26] and is therefore much less than characteristic vessel dimensions, in applying Eq. (9.32) the flame can be

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treated as negligibly thin. Laminar burning velocities for methane, propane, isooctane, methanol, gasoline, and hydrogen—premixed with air—at pressures, temperatures, and equivalence ratios which occur in engines have been measured using this technique.[29–33] Also, the effect of a burned gas diluent on laminar burning velocity with gasoline-air mixtures has been determined.[32] Correlations derived from these data are the most accurate means available for estimating laminar burning velocities for mixtures and conditions relevant to spark-ignition engines.

The effect of the mixture fuel/air equivalence ratio on laminar burning velocity for several hydrocarbon fuels and methanol is shown in Fig. 9-25. The burning velocity peaks slightly rich of stoichiometric for all the fuels shown. The values for isooctane and gasoline are closely comparable. Data at higher pressures and temperatures have been fitted to a power law of the form:

$$S_L = S_{L,0}\left(\frac{T_u}{T_0}\right)^{\alpha}\left(\frac{p}{p_0}\right)^{\beta} \tag{9.33}$$

where $T_0 = 298$ K and $p_0 = 1$ atm are the reference temperature and pressure, and $S_{L,0}$, α, and β are constants for a given fuel, equivalence ratio, and burned gas diluent fraction. For propane, isooctane, and methanol, these constants can be represented by

$$\alpha = 2.18 - 0.8(\phi - 1) \tag{9.34a}$$

$$\beta = -0.16 + 0.22(\phi - 1) \tag{9.34b}$$

and

$$S_{L,0} = B_m + B_\phi(\phi - \phi_m)^2 \tag{9.35}$$

where ϕ_m is the equivalence ratio at which $S_{L,0}$ is a maximum with value B_m.

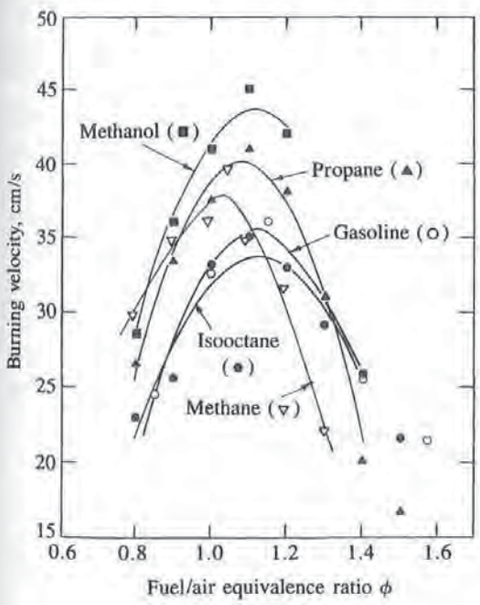


FIGURE 9-25
Laminar burning velocity for several fuels as function of equivalence ratio, at 1 atm and 300 K. Lines are least-squares polynomial fits to data.[29, 30]

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TABLE 9.2
Parameters ϕ_m, B_m, and B_ϕ for Eq. (9.35)

Fuel	ϕ_m	B_m, cm/s	B_ϕ, cm/s	Ref.
Methanol	1.11	36.9	−140.5	30
Propane	1.08	34.2	−138.7	30
Isooctane	1.13	26.3	−84.7	30
Gasoline	1.21	30.5	−54.9	32

Note: Values of $S_{L,0}$ given by Eq. (9.35) are obtained from least-squares fits of Eq. (9.33) to data over the range $p = 1$–8 atm, $T_u = 300$–700 K. They do not correspond exactly to the laminar flame speed data at 1 atm and 298 K in Fig. 9-25.

Values of ϕ_m, B_m, and B_ϕ are given in Table 9.2.[30] For gasoline (a reference gasoline with average molecular weight of 107 and an H/C ratio of 1.69) additional data were available and were correlated by[32]

$$\alpha_g = 2.4 - 0.271\phi^{3.51} \tag{9.36a}$$

$$\beta_g = -0.357 + 0.14\phi^{2.77} \tag{9.36b}$$

For methane, simple equations such as (9.34*a*, *b*) do not adequately correlate the data over the range of p and T_u relevant to engines. However, laminar burning velocity data from a spherical constant-volume bomb experiment have been obtained along an unburned gas isentropic path, as the pressure in the bomb rises during combustion. Variation in laminar burning velocity along such unburned gas isentropes does correlate with a power law:

$$S_{L,s} = S_{L,0}\left(\frac{\rho_u}{\rho_{u0}}\right)_s^{\varepsilon} \tag{9.37}$$

Values for $S_{L,0}$ and ε from the literature are summarized in Table 9.3.

TABLE 9.3
Parameters for methane-air laminar burning velocity correlation [Eq. (9.37)]

ϕ	p_i, atm	$S_{L,0}$,† cm/s	ε	Ref.
1.0	0.5	49	0.51	31
1.0	1.0	35	0.2	31
0.8–1.2	1–8	‡	0.17–0.19	33

† At 298 K initial temperature.
‡ See Fig. 9-25.

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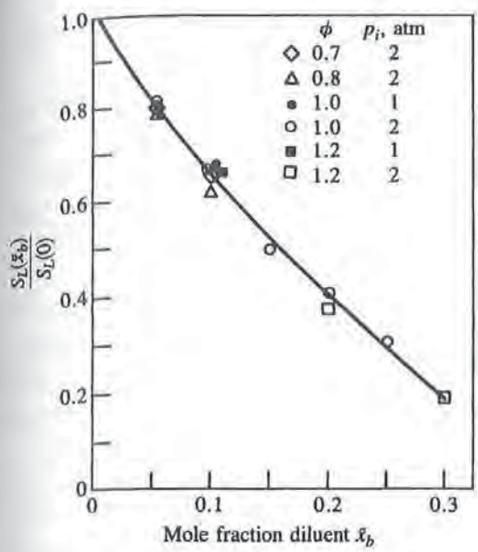


FIGURE 9-26
Effect of burned gas mole fraction $\tilde{x}_b$ in unburned mixture on laminar burning velocity. Fuel: gasoline.[32]

The presence of burned gas in the unburned cylinder charge due to residual gases and any recycled exhaust gases causes a substantial reduction in the laminar burning velocity. Any burned gas in the unburned mixture reduces the heating value per unit mass of mixture and, thus, reduces the adiabatic flame temperature. It acts as a diluent. The effect of increasing burned gas or diluent fraction on laminar flame speed is shown in Fig. 9-26. The diluent used was a mixture of CO_2 and N_2, chosen to match the heat capacity of actual gasoline-air combustion products.† The proportional reduction in laminar burning velocity is essentially independent of the unburned mixture equivalence ratio, pressure, and temperature over the range of interest in engines. The data in Fig. 9-26 are correlated by the relation:

$$S_L(\tilde{x}_b) = S_L(\tilde{x}_b = 0)(1 - 2.06\tilde{x}_b^{0.77}) \tag{9.38}$$

where $\tilde{x}_b$ is the mole fraction of burned gas diluent. Other studies corroborate the magnitude of this burned gas effect.[32]

Note that for equal heat capacity added to the unburned mixture, burned gases have a much larger effect on laminar burning velocity than does excess air. For example, the laminar burning velocity of a stoichiometric mixture as it is leaned to $\phi = 0.8$ is reduced by 23 percent. The excess air required has a heat capacity of about 0.2 times that of the combustion products of the undiluted mixture. Adding the same heat capacity by adding stoichiometric burned gases (which requires a burned gas mole fraction of 0.175) reduces the laminar burning

† The water in actual residual and exhaust gas was omitted. A mixture of 80 percent N_2 and 20 percent CO_2, by volume, was used.

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velocity by 55 percent.[32] Proper allowance for the burned gas fraction in estimating laminar burning velocities for spark-ignition engines is most important.

The above correlations define the laminar burning velocity as a function of unburned mixture thermodynamic properties and composition, only. It has been assumed that flame thickness and curvature effects are negligible.[28] Our interest in laminar burning velocity is twofold: first, it is used to define the characteristic chemical reaction time of the mixture in Eq. (9.30); second, a presumed consequence of the wrinkled thin-reaction-sheet turbulent-flame structure is that, locally, the sheet propagates at the laminar burning velocity. The above correlations adequately characterize a quiescent burning process. However, laminar flame propagation can be influenced by the local flow field in the unburned gas. If the flame thickness is less than the Kolmogorov scale, the primary effect is one of straining which affects both the flame area (usually referred to as flame stretching for an area increase) and the local (laminar) burning velocity. While this problem is not yet well understood, it is known that straining can affect the laminar burning velocity and can cause flame extinction. The laminar burning velocity decreases with increasing strain rate, and the Lewis number of the unburned mixture has a significant influence on this rate of decrease. The Lewis number is the ratio of diffusivities of heat and mass. For stoichiometric mixtures it is close to one; it increases above about unity as the unburned fuel-air mixture is leaned out. Thus the local flow field may have a discernable effect on the local burning velocity of the thin laminarlike reaction-sheet flame, especially for lean or dilute mixtures.[34]

9.3.4 Flame Propagation Relations

If the heat-release or mass burning rate analysis of Sec. 9.2.2 is coupled with an analysis of flame geometry data, substantial additional insight into the behavior of spark-ignition engine flames is obtained. Flame photographs (such as those in Figs. 9-1, 9-14, and 9-16 and Refs. 4 and 35) effectively define the position of the front or leading edge of the turbulent engine flame. The "shadow" of the enflamed zone, under normal engine conditions, is close to circle: only in the presence of very high swirl does substantial distortion of the flame shape occur.[18] Thus, to a good approximation, the surface which defines the leading edge of the turbulent flame (ahead of which only unburned mixture exists) is a portion of the surface of a sphere. Figure 9-27 indicates the geometrical parameters which define this flame surface: r_c, α_c, z_c, the coordinates of the flame center; r_f, the radius of the best-fit circle to the flame front silhouette; and the geometry of the combustion chamber walls. The flame is initiated at the spark plug; however, it may move away from the plug during the early stages of its development as shown. We define the *flame front area* A_f as the spherical surface of radius r_f coinciding with the leading edge of the flame contained within the combustion chamber, and the *enflamed volume* V_f as the volume within the chamber behind this flame front.

The thermodynamic analysis of cylinder pressure data allows us to define additional geometrical parameters. The *burned gas radius* r_b is the radius of the

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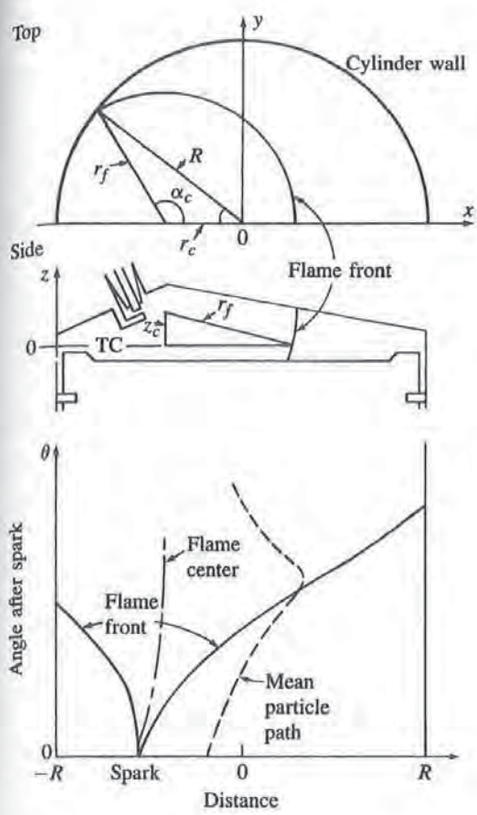


FIGURE 9-27
Schematic of spherical flame front in engine combustion chamber identifying parameters which define flame geometry. (*From Beretta* et al.[4])

spherical surface within the combustion chamber which would contain all the burned gas behind it; i.e.,

$$V_b(r_b, r_c, \alpha_c, z_c) = V_b(p, \theta) \tag{9.39}$$

The *spherical burning area* A_b is the area of this spherical surface; i.e.,

$$A_b = \frac{\partial V_b(r_b, r_c, \alpha_c, z_c)}{\partial r_b} \tag{9.40}$$

The *laminar burning area* A_L is the surface area the flame would have if it burned at the laminar flame speed, i.e.,

$$A_L = \frac{dm_b/dt}{\rho_u S_L} \tag{9.41}$$

where S_L is the laminar flame speed in the unburned mixture ahead of the flame (see Sec. 9.3.3).

Several velocities can be defined. The *mean expansion speed* of the front u_f is given by

$$u_f = \frac{dA_S/dt}{L_S} \tag{9.42}$$

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where A_S is the "shadow" area enclosed by the "best-fit" circle through the leading edge of the flame and

$$L_S = \frac{\partial A_S}{\partial r_f}$$

is the arc length within the chamber of this "best-fit" circle. The *mean expansion speed* of the *burned gas* u_b is

$$u_b = \frac{\partial V_b/\partial t}{A_b} \tag{9.43}$$

This derivative is taken with the piston position fixed since only burned volume changes due to combustion are of interest. The *burning speed* S_b is defined by

$$S_b = \frac{dm_b/dt}{\rho_u A_b} \tag{9.44}$$

The *mean gas speed* just ahead of the flame front u_g is

$$u_g = u_b - S_b \tag{9.45}$$

Note that combining Eqs. (9.41) and (9.44) gives the relation

$$S_b A_b = S_L A_L \tag{9.46}$$

Also, it follows from Eqs. (9.29), (9.43), and (9.44) that

$$\frac{u_b}{S_b} = \frac{\rho_u}{\rho_b}(1 - y_b) + y_b = \frac{\rho_u/\rho_b}{[(\rho_u/\rho_b) - 1]x_b + 1} \tag{9.47}$$

As x_b and $y_b \to 0$, u_b/S_b approaches the expansion ratio ρ_u/ρ_b. As x_b and $y_b \to 1$, u_b/S_b approaches unity.

The variation of the above quantities during the engine combustion process, coupled with the photographs and discussion in Sec. 9.3.2, provide substantial insight into the flame development and propagation process. Figure 9-28 shows results from an analysis of cylinder pressure data and the corresponding flame front location information (determined from high-speed movies through a window in the piston) of several individual engine operating cycles. The combustion chamber was a typical wedge design with a bore of 102 mm and a compression ratio of 7.86. The flame radius initially grows at a rate that increases with time and exhibits substantial cycle-by-cycle variation in its early development (Fig. 9-28*a*). Later ($r_f \gtrsim 30$ mm) the growth rate, which approximates the expansion speed u_b, reaches an essentially constant value. The flame radius r_f is initially equal to the burned gas radius r_b; it increases above r_b as the flame grows and becomes increasingly distorted by the turbulent flow field (Fig. 9-28*b*). Eventually $r_f - r_b$ goes to an essentially constant value of about 6 mm for $r_b \gtrsim 30$ mm. This difference, $r_f - r_b$, is approximately half the thickness of the turbulent flame brush.

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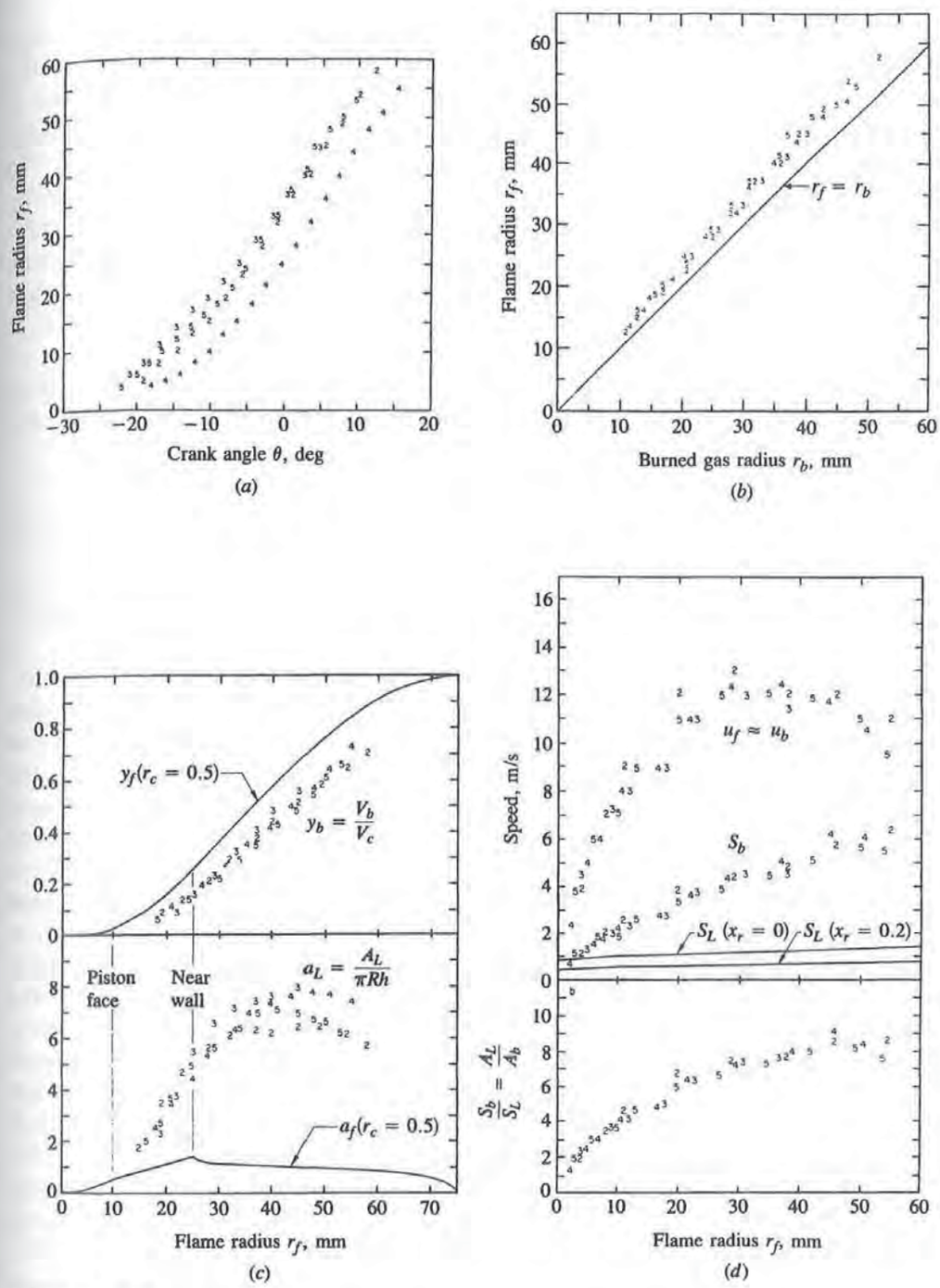


FIGURE 9-28
Variation of flame geometry and velocity parameters during four individual combustion cycles at 1044 rev/min, $\phi = 0.98$, 1 atm inlet pressure: (*a*) flame radius r_f versus crank angle; (*b*) flame radius r_f versus burned gas radius r_b; (*c*) normalized enflamed volume y_f, burned volume y_b, normalized flame front area a_f, and laminar area a_L versus flame radius; (*d*) front expansion speed u_b, burning speed S_b, and laminar flame speed S_L versus flame radius. (*From Beretta* et al.[4])

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Normalized enflamed and burned volumes, and flame front area and laminar burning area, are shown in Fig. 9-28*c*. Volumes are normalized by the cylinder volume, and areas by πRh, where h is the average clearance height and R the cylinder radius. Discontinuities occur in the flame area a_f at the points where the flame front contacts first the piston face and then the near cylinder wall. The laminar area A_L is initially close to the flame area A_f and then increases rapidly as the flame grows beyond 10 mm in radius. During the rapid burning combustion phase ($y_f \gtrsim 0.2$) the value of y_f is significantly greater than y_b. During this phase, the laminar area exceeds the flame area by almost an order of magnitude. These observations indicate the existence of substantial pockets of unburned mixture behind the leading edge of the flame.[4]

The ratio of the volume of the unburned mixture within the turbulent flame zone ($V_f - V_b$) to the reaction-sheet area within the flame zone ($A_L - A_f$) defines a characteristic length

$$l_T = \frac{V_f - V_b}{A_L - A_f} \tag{9.48}$$

which can be thought of as the scale of the pockets of unburned mixture within the flame. For the data set of Fig. 9-28, l_T is approximately constant and of order 1 mm.[24]

These flame geometry results would be expected from the previous photographic observations of how the flame grows from a small approximately spherical smooth-surfaced kernal shortly after ignition to a highly wrinkled reaction-sheet turbulent flame of substantial overall thickness. Initially, the amount of unburned gas within the enflamed volume is small. During the rapid burning phase of the combustion process, however, a significant fraction (some 25 percent; see Fig. 9-2) of the gas entrained into the flame zone is unburned.

The front expansion speed u_f, burning speed S_b, and laminar flame speed S_L are shown in Fig. 9-28*d*. The expansion speed increases as the flame develops to a maximum value that is several times the mean piston speed of 3.1 m/s and is comparable to the mean flow velocity through the inlet valve of 18 m/s.[24] The burning speed increases steadily from a value close to the laminar flame speed at early times to almost an order of magnitude greater than S_L during the rapid burning phase. During this rapid burning phase, since $(r_f - r_b)$ is approximately constant, the flame front expansion speed and the mean burned gas expansion speed are essentially equal. The difference between $u_b \approx u_f$ and S_b is the unburned gas speed u_g just ahead of the flame front. Note that the ratio u_f/S_b ($\approx u_b/S_b$) decreases monotonically from a value equal to the expansion ratio (ρ_u/ρ_b) at spark to unity as the flame approaches the far wall, as required by Eq. (9.47).

The effect of flame propagation on the flow field in the unburned mixture ahead of the flame is important because it is the turbulence just ahead of the flame that determines the local burning velocity. Measurements of mean velocities, rms fluctuation velocities, and turbulence intensities have been made using laser doppler anemometry (see Sec. 8.2.2) at different locations within engine combustion chambers (e.g., Refs. 36 and 37). Such data are difficult to interpret

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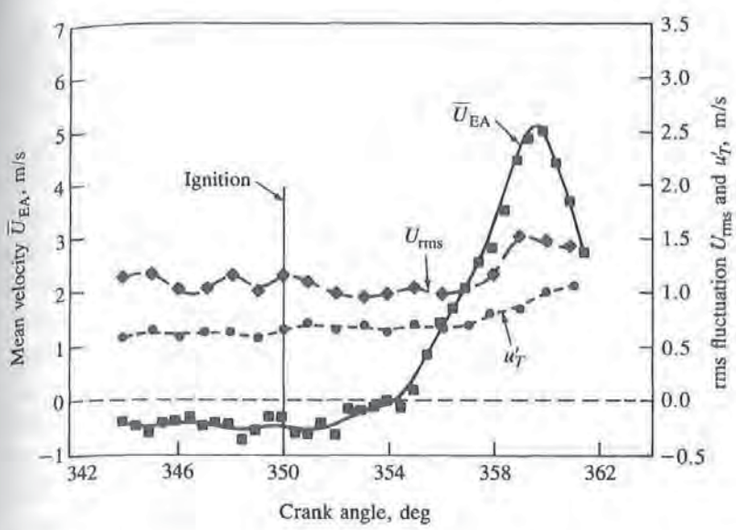


FIGURE 9-29
Laser doppler anemometer measurements of ensemble-averaged mean velocity $\bar{U}_{EA}$ [Eq. (8.20)], rms fluctuation in individual-cycle mean velocity U_{rms} [Eq. (8.21)] and turbulence intensity u'_T [Eq. (8.22)], close to the cylinder axis, from before ignition to after flame arrival. Disc-shaped chamber, spark plug in cylinder wall, measurement at $x/B = 0.57$, $B = 76$ mm, 300 rev/min. $\bar{S}_p = 0.83$ m/s.[36]

because the mean flow varies cycle-by-cycle, the turbulence is not homogeneous, and the flame motion and shape show substantial cyclic variations. The results in Fig. 9-29 were taken in a special single-cylinder engine with a disc-shaped combustion chamber where the spark plug was located in the cylinder liner. Shown are the ensemble-averaged mean velocity, cyclic variation in mean velocity, and the turbulence intensity, normal to the front, during the major portion of the combustion process, close to the chamber center.

The mean velocity normal to the front increases steadily from shortly after ignition, as the combustion-produced gas expansion displaces unburned mixture toward the wall. It peaks as the flame arrives at the measurement location. The cyclic variation in mean velocity and the turbulence intensity normal to the front remain essentially constant until a few degrees before the flame arrival. These two quantities are comparable in magnitude; thus the turbulence intensity is lower than the rms fluctuation velocity (in this case by about a factor of 2) (see Sec. 8.2.2). Whether the increase in turbulence as the flame approaches is due to rapid distortion resulting from the compression of the unburned mixture which occurs during combustion or is the result of inadequate resolution of cycle-to-cycle flow variations is unclear. Rapidly imposed distortions of a turbulent flow field, such as those imposed by combustion-produced gas expansion, would lead to an increase in vorticity and turbulence intensity. Other studies, e.g., Ref. 37, indicate there is little or no increase in turbulence intensity ahead of the flame.

The variation of burning speed with engine speed has also been carefully examined in a study where flame position was determined from high-speed movies, mass burning rates from cylinder pressure, and turbulence information

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from experiments at equivalent motored conditions.[35] The turbulence quantity obtained during motoring experiments was the ensemble-averaged root mean square velocity fluctuation defined by Eq. (8.18). Values of S_b/S_L and u'_F/S_L were determined at two points in the combustion process: at a flame radius of 30 mm (the end of the flame development process) and at mass fraction burned equal to 0.5 (halfway through the rapid burning phase). To correct the motored turbulence data for the higher pressure levels corresponding to engine firing conditions, a simple rapid distortion model (see Sec. 14.4.2) based on conservation of angular momentum in turbulent eddies was used. A linear correlation between S_b and u'_F results, as shown in Fig. 9-30, for the rapid burning combustion phase. Note that as u'_F/S_L goes to zero, S_b/S_L approaches a value close to unity.

Once the flame front reaches the far cylinder wall (see Fig. 9-27) the front can no longer propagate: however, combustion continues behind the front until all the unburned mixture entrained into the enflamed region is consumed. This final burning or termination phase of the combustion process can be approximated by an exponential decay in the mass burning rate with a characteristic time constant τ_b of order 1 ms. Since these "islands" or "pockets" of unburned mixture behind the leading edge of the flame have a characteristic scale l_T based on the laminar flame area [Eqs. (9.41) and (9.48)], it follows that

$$l_T = \tau_b S_L \tag{9.49}$$

In summary, the above flame data analysis procedures show that the relationships between r_f and r_b, V_f and V_b, A_f and A_L, u_f, S_b and S_L are distinctly different in the three phases of combustion: (1) the development phase, where a highly wrinkled reaction-sheet "thick"-overall turbulent flame evolves from the essentially spherical flame kernal established by the spark discharge; (2) the rapid-burning phase, where this thick "developed" turbulent flame propagates across the combustion chamber to the far wall, during which most of the mass is burned; and (3) the termination phase after the flame front has reached the far wall and propagation of the front is no longer possible, when the remaining unburned mixture within the flame burns up. The burning velocity, in the rapid-burning phase of the combustion process, scales with turbulence intensity, which in turn scales with engine speed.

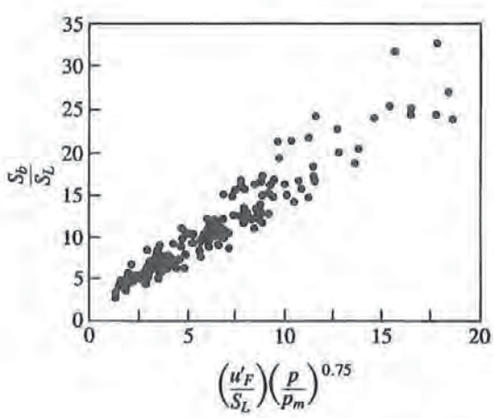


FIGURE 9-30
Variation of burning speed with turbulence intensity. The ensemble-averaged rms velocity fluctuation was measured during motoring engine operation. The ratio p/p_m (firing pressure/motoring pressure) corrects for the effect of additional compression on the turbulence intensity. Range of engine speeds and spark timings.[35]

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9.4 CYCLIC VARIATIONS IN COMBUSTION, PARTIAL BURNING, AND MISFIRE

9.4.1 Observations and Definitions

Observation of cylinder pressure versus time measurements from a spark-ignition engine, for successive operating cycles, shows that substantial variations on a cycle-by-cycle basis exist. Since the pressure development is uniquely related to the combustion process, substantial variations in the combustion process on a cycle-by-cycle basis are occurring. In addition to these variations in each individual cylinder, there can be significant differences in the combustion process and pressure development between the cylinders in a multicylinder engine. Cyclic variations in the combustion process are caused by variations in mixture motion within the cylinder at the time of spark cycle-by-cycle, variations in the amounts of air and fuel fed to the cylinder each cycle, and variations in the mixing of fresh mixture and residual gases within the cylinder each cycle, especially in the vicinity of the spark plug. Variations between cylinders are caused by differences in these same phenomena, cylinder-to-cylinder.

Cycle-by-cycle variations in the combustion process are important for two reasons. First, since the optimum spark timing is set for the "average" cycle, faster-than-average cycles have effectively overadvanced spark-timing and slower-than-average cycles have retarded timing, so losses in power and efficiency result. Second, it is the extremes of the cyclic variations that limit engine operation. The fastest burning cycles with their overadvanced spark timing are most likely to knock. Thus, the fastest burning cycles determine the engine's fuel octane requirement and limit its compression ratio (see Sec. 9.6.3). The slowest burning cycles, which are retarded relative to optimum timing, are most likely to burn incompletely. Thus these cycles set the practical lean operating limit of the engine or limit the amount of exhaust gas recycle (used for NO emissions control) which the engine will tolerate. Due to cycle-by-cycle variations, the spark timing and average air/fuel ratio must always be compromises, which are not necessarily the optimum for the average cylinder combustion process. Variations in cylinder pressure have been shown to correlate with variations in brake torque which directly relate to vehicle driveability.

An example of the cycle-by-cycle variations in cylinder pressure and the variations in mixture burning rate that cause them are shown in Fig. 9-31. Pressure and gross heat-release rate [calculated from the cylinder pressure using Eq. (9.27)] for several successive cycles at a mid-load, mid-speed point are shown as a function of crank angle. The maximum heat-release rate and the duration of the heat release or burning process vary by a factor of two from the slowest to the fastest burning cycle shown. The peak cylinder pressure varies accordingly. The faster burning cycles have substantially higher values of maximum pressure than do the slower burning cycles; with the faster burning cycles peak pressure occurs closer to TC.

The heat-release rate data in Fig. 9-31 show that there are cycle-by-cycle variations in the early stages of flame development (from zero to a few percent of

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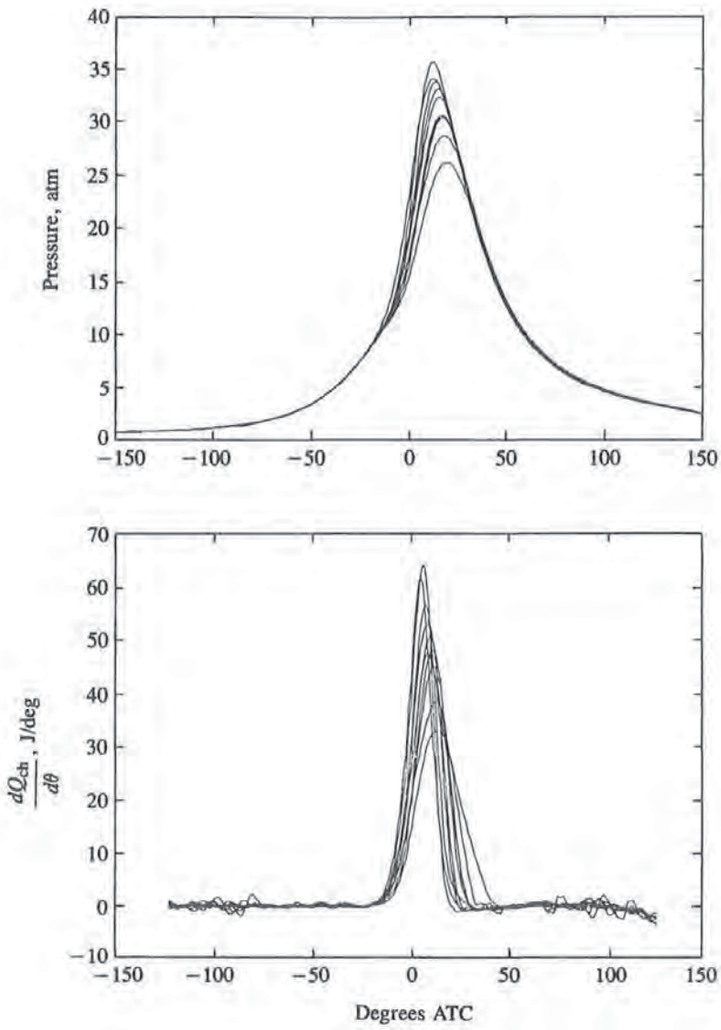


FIGURE 9-31
Measured cylinder pressure and calculated gross heat-release rate for ten cycles in single-cylinder spark-ignition engine operating at 1500 rev/min, $\phi = 1.0$, $p_{inlet} = 0.7$ atm, MBT timing 25° BTC.[14]

the total heat release) and in the major portion of the combustion process—the rapid-burning phase—indicated by the variations in the maximum burning rate.

As the mixture becomes leaner with excess air or more dilute with a higher burned gas fraction from residual gases or exhaust gas recycle, the magnitude of cycle-by-cycle combustion variations increases. Eventually, some cycles become sufficiently slow burning that combustion is not completed by the time the exhaust opens: a regime where *partial burning* occurs in a fraction of the cycles is encountered. For even leaner or more dilute mixtures, the *misfire* limit is reached. At this point, the mixture in a fraction of the cycles fails to ignite. While spark-ignition engines will continue to operate with a small percentage of the cycles in the partial-burn or misfire regimes, such operation is obviously undesirable from

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the point of efficiency, hydrocarbon emissions, torque variations, and roughness. The partial-burn and misfire regimes are discussed in Sec. 9.4.3.

Various measures of cycle-by-cycle combustion variability are used. It can be defined in terms of variations in the cylinder pressure between different cycles, or in terms of variations in the details of the burning process which cause the differences in pressure. The following quantities have been used:

1. *Pressure-related parameters.* The maximum cylinder pressure p_{max}; the crank angle at which this maximum pressure occurs θ_{pmax}; the maximum rate of pressure rise $(dp/d\theta)_{max}$; the crank angle at which $(dp/d\theta)_{max}$ occurs; the indicated mean effective pressure [which equals $\int p\, dV/V_d$, see Eqs. (2.14), (2.15), and (2.19)].
2. *Burn-rate-related parameters.* The maximum heat-release rate (net or gross, see Sec. 9.2.2); the maximum mass burning rate; the flame development angle $\Delta\theta_d$ and the rapid burning angle, $\Delta\theta_b$ (see Sec. 9.2.3).
3. *Flame front position parameters.* Flame radius, flame front area, enflamed or burned volume, all at given times; flame arrival time at given locations.

Pressure-related quantities are easiest to determine; however, the relation between variations in combustion rate and variations in cylinder pressure is complex.[38] Equation (9.26) defines the factors that govern this relationship. Because the rate of change of pressure is substantially affected by the rate of change of cylinder volume as well as rate of burning, changes in the phasing of the combustion process relative to TC (e.g., which result from changes in flame development angle) as well as changes in the shape and magnitude of the heat-release rate profile affect the pressure. Figure 9-32 illustrates how the magnitude of the maximum cylinder pressure p_{max} and the crank angle at which it occurs θ_{pmax} vary as the crank angle at which combustion effectively starts (e.g., θ at which 1 percent of the cylinder mass has burned) and the burning rate are varied. Curve $CABE$ shows how p_{max} and θ_{pmax} vary for a fixed fast-burning heat-release profile (the duration of the heat-release process and its maximum value are held

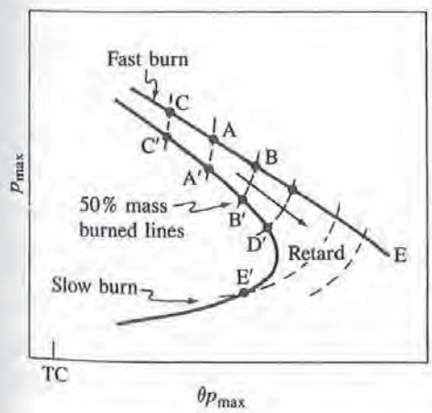


FIGURE 9-32
Schematic of variation in maximum cylinder pressure and crank angle at which it occurs, in individual cycles. $CABE$ typical of fast heat-release process; $C'A'B'D'E'$ typical of slow heat-release process. (*From Matekunas.*[38])

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constant), as the phasing of this combustion process relative to TC is varied. A corresponds to MBT timing where the start of combustion is phased to give maximum brake torque, B corresponds to retarded timing, and C to over-advanced timing. $C'A'B'D'E'$ is a similar curve for a slow-burning heat-release profile. A' corresponds to MBT timing, and B' and D' to increasingly retarded timing. Note that with a sufficiently slow-burning heat-release profile, beyond D', $\theta_{p\max}$ decreases as the burn process is increasingly retarded. This occurs when the rate of increase of pressure due to combustion becomes so low that it is more than offset by the pressure decrease due to volume increase: eventually for extremely slow and late burning, the maximum pressure approaches the motored pressure at TC. The dashed lines show the constant start-of-combustion timing *relative* to MBT for each burn rate curve. Note that $\theta_{p\max}$ for constant relative timing varies little as the heat-release profile or burn rate varies.[38]

We can now explain the effects of variations in the heat-release profile (both in the development stage of the burning process, which effectively changes the location of the start of combustion, and in the rate of burning throughout the process) on $p_{\max}$ and $\theta_{p\max}$, when the spark timing occurs at a fixed crank angle. For a fixed burning rate profile (duration of burn and maximum burning rate) as the start of combustion is delayed to be closer to TC, $p_{\max}$ decreases and $\theta_{p\max}$ initially increases (A to B or A' to B'). This is the effect of a change in relative timing or phase of the burning process due to a slower initial rate of flame development with fixed spark timing. If, in addition to the flame development being slower, the heat-release rate throughout the burning process is lower, then that combustion process is even more retarded from the optimum and $p_{\max}$ decreases and $\theta_{p\max}$ increases further, to their values at D'. The effect of a faster initial flame development and faster burning rate, with fixed spark time, is the opposite. The magnitudes of the changes in $p_{\max}$ and $\theta_{p\max}$ depend, obviously, on the extent of the cyclic variations; they also depend on whether the average burn process is fast or slow. For fast-burning engines, a larger fraction of the heat release occurs near TC when the chamber volume is changing relatively slowly. Thus pressure variations are mainly due to combustion variations. With slow-burning engines, where a significant fraction of the energy release occurs well after TC, the effect of volume change also becomes significant and augments the effect of combustion variations. For large variations and a slow average burning process, $p_{\max}$ can fall below E', and $\theta_{p\max}$ then decreases. A fast-burning combustion process significantly reduces the impact of cyclic combustion variations on engine performance.[39]

We can now evaluate the various measures of combustion variability. The maximum pressure variation has been shown to depend on both changes in phasing and burning rate. The magnitude of this variation depends on whether the combustion chamber is faster or slower burning, on average. It also depends on whether the burning process is substantially retarded relative to MBT. It depends, too, on cyclic cylinder fuel and air charging variations. Thus the interpretation of variations in $p_{\max}$ [or in the maximum rate of pressure rise $(dp/d\theta)_{\max}$] in terms of variations in the rate and phasing of the burning process

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must be done with care. The location of maximum pressure $\theta_{p\max}$ also depends on relative phasing of combustion and on the burn rate profile. In addition, for slow-burning chambers and retarded timing (around E') variations produce little change in $\theta_{p\max}$. However, for fast-burning chambers, with MBT or only slightly retarded timing, the location of peak pressure depends essentially on the phasing of each combustion process relative to its MBT phasing, and is independent of charging variations. For these reasons, $\theta_{p\max}$ is a useful measure of variability in combustion event phasing.[38]

One important measure of cyclic variability, derived from pressure data, is the *coefficient of variation* in *indicated mean effective pressure*. It is the standard deviation in imep divided by the mean imep, and is usually expressed in percent:

$$\mathrm{COV}_{\mathrm{imep}} = \frac{\sigma_{\mathrm{imep}}}{\mathrm{imep}} \times 100 \tag{9.50}$$

It defines the cyclic variability in indicated work per cycle, and it has been found that vehicle driveability problems usually result when $\mathrm{COV}_{\mathrm{imep}}$ exceeds about 10 percent.

Figure 9-33 illustrates the relationships between $p_{\max}$, $\theta_{p\max}$, and imep for 120 cycles of an engine cylinder at fixed operating conditions and three different spark timings.[38] The MBT timing data show a spread in imep at a fixed value of $\theta_{p\max}$. This imep data band is relatively flat and is centered around $\theta_{p\max} \approx 16°$; only at later values of $\theta_{p\max}$ does imep fall off significantly. The vertical spread in imep around $\theta_{p\max} = 16°$ is due to variations in the amount of fuel entering the cylinder each cycle; normal variations in the burn profile under these conditions, which effectively change the phasing of the combustion process, produce only modest reductions in imep. For early $\theta_{p\max}$ (the extreme upper left of Fig. 9-33*a*), the variations in $p_{\max}$ are also due mainly to these fuel-charging variations, cycle-by-cycle; these are the fastest burning cycles with the most advanced phasing. As

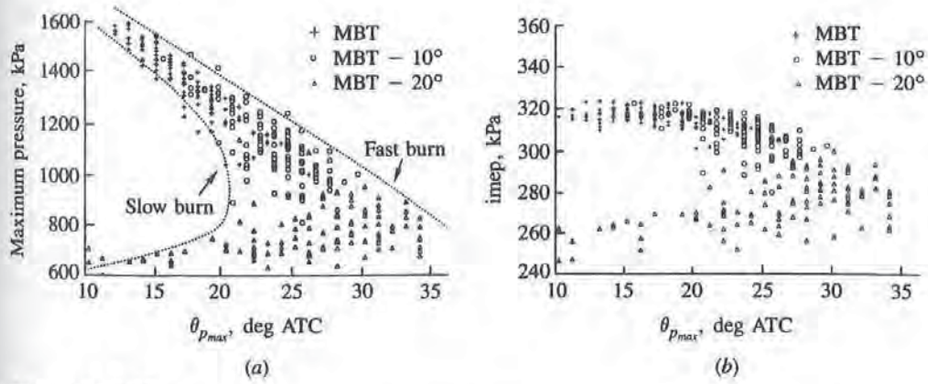


FIGURE 9-33
(*a*) Individual-cycle maximum pressure versus crank angle at which $p_{\max}$ occurs. (*b*) Individual-cycle indicated mean effective pressure versus $\theta_{p\max}$.[38]

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$\theta_{p\max}$ increases, the dispersion increases as cyclic variations in phasing and burning rate have increasing impact.

An important issue is whether variations in the early stages of flame development and variations in subsequent portions of the burning process are independent of each other or are correlated. Plots of early flame development angle (spark to 1 percent mass burned) against the burning angle (1 to 90 percent burned) from individual cycles for several different combustion chambers indicate the following. There is a trend with increasing flame development angle for the burning angle to increase (or the burning rate to decrease); however, there is much scatter about this trend (for a given value of flame development angle different cycles show a substantial range in burning angle), and the quantitative aspects of the trend depend on operating conditions and on combustion chamber design. In addition, as the mean rapid burning angle increases (due to changing operating conditions or a slower-burning chamber design) the mean flame development angle, the cyclic variation in the flame development angle, and the cyclic variation in the rapid burning angle all increase.[40] This topic is discussed more fully in the following section.

The shapes of the frequency distributions in individual-cycle pressure data (e.g., in $p_{\max}$, $(dp/d\theta)_{\max}$, $\theta_{p\max}$) and in burn rate data such as $\Delta\theta_d$, $\Delta\theta_b$, $(dQ/d\theta)_{\max}$ depend on whether the combustion process is fast and "robust" (e.g., with close-to-stoichiometric mixtures at higher loads at optimum timing—well away from the lean operating limit of the engine) or slower and less repeatable, closer to the lean or dilute-mixture operating limit. Under robust combustion conditions these distributions are close to normal distributions.[41–43] When the combustion process is much slower, the cyclic variability becomes large and the distribution becomes skewed toward the slower burning cycles which have low imep (due to the substantial retard of these slower cycles). When partial burning and then misfire occur, the low-pressure tail of the distribution approaches the motored pressure value at TC.[44] Examples of the frequency distributions of imep in these two combustion variability regimes are shown later in Fig. 9-36.

Cylinder pressure data are often averaged over many cycles to obtain the mean cylinder pressure at each crank angle. The primary use of this average pressure versus crank angle data is in calculating the average indicated mean effective pressure (which is a linear function of p). Since combustion parameters are not linearly related to the cylinder pressure [see Eq. (9-27)], analysis of the *average* pressure data will not necessarily yield accurate values of average combustion parameters. The error will be most significant when the combustion variability is largest. It is best to determine mean combustion parameters by averaging their values obtained from a substantial number of individual cycle analysis results. The number of cycles which must be averaged to obtain the desired accuracy depends on the extent of the combustion variability. For example, while 40 to 100 cycles may define imep to within a few percent when combustion is highly repeatable, several hundred cycles of data may be required when cyclic combustion variations are large.[12]

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9.4.2 Causes of Cycle-by-Cycle and Cylinder-to-Cylinder Variations

Cycle-by-cycle combustion variations are evident from the beginning of the combustion process. Analysis of flame photographs from many engine cycles taken in special research engines with windows in the combustion chamber has shown that dispersion in the fraction of the combustion chamber volume inflamed is present from the start of combustion (e.g., see Refs. 3 and 23). Dispersion in burning rate is also evident throughout the combustion process (see Figs. 9-2 and 9-31). Three factors have been found to influence this dispersion:[45]

1. The variation in gas motion in the cylinder during combustion, cycle-by-cycle
2. The variation in the amounts of fuel, air, and recycled exhaust gas supplied to a given cylinder each cycle
3. Variations in mixture composition within the cylinder each cycle—especially near the spark plug—due to variations in mixing between air, fuel, recycled exhaust gas, and residual gas

The relative importance of these factors is not yet fully defined, and depends on engine design and operating variables. The variation in the velocity field within the engine cylinder throughout the cycle, and from one cycle to the next, has been reviewed in Sec. 8.2.2. Toward the end of the compression stroke, the ensemble-averaged rms velocity fluctuation is of comparable magnitude to the mean piston speed, and may be larger than the mean flow velocity if there is no strongly directed local mean flow pattern (see Figs. 8-8 and 8-9). This ensemble-averaged velocity fluctuation combines both cycle-by-cycle variation in the mean flow and the turbulent velocity fluctuations. During compression, these two components are of comparable magnitude (see Figs. 8-9 and 9-29). While this data base is limited, it indicates that substantial variations in the mean flow exist, cycle-by-cycle, both in the vicinity of the spark plug *and* throughout the combustion chamber. Velocity variations contribute in a major way to variations in the initial motion of the flame center as it grows from the kernel established by the spark, and in the initial rate of growth of the flame; they can also affect the burning rate once the flame has developed to fill a substantial fraction of the combustion chamber. Variations in gas motion near the spark plug convect the flame in its early stages in different directions and at different velocities, cycle-by-cycle. This affects the flame's interaction with the cylinder walls, changing the flame area development with time. Variations in the turbulent velocity fluctuations near the spark plug will result in variations in the rate at which the small initially laminarlike flame kernel develops into a turbulent flame. Variations in the mean flow throughout the chamber will produce differences in flame front shape; also, they may produce differences in turbulence which affect the propagation velocity of the front (see Fig. 9-30 for the relation between mean burning speed and turbulence intensity).

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It is well known that, on a time-averaged basis, the fuel, air, and recycled exhaust gas flows into each cylinder of a multicylinder engine are not identical. These flow rate differences are typically a few percent (see Secs. 7.6.2 and 7.6.3). It is also known that the flow patterns within the different cylinders are not necessarily identical due to differences between the individual intake manifold runner and port geometries in many production engines. All these factors contribute to cylinder-to-cylinder variations in the combustion process: there can be significant differences in the mean burn rate parameters as well as in the cyclic variations in these parameters.[41] Also, the limited data available on the variation in mixture composition within each cylinder for each cycle indicates that cyclic charging variations in individual cylinders are comparable in magnitude to cylinder-to-cylinder differences (i.e., of order ± 5 percent[46]). Whether the amount of residual gas left in the cylinder varies significantly, cycle-by-cycle, is not known. At higher loads, where the combustion process is more repeatable (and always completed relatively early in the expansion stroke) and the residual gas fraction smaller (see Sec. 6.4), variations in the total amount of residual are not expected to be significant. At light loads (particularly at idle), where combustion variability is much higher and partial-burning cycles may occur, and especially with high valve overlap engine designs, variations in the residual gas mass and its composition may become important.

In addition, mixing of fuel, air, recycled exhaust, and residual is not complete: nonuniformities in composition exist within the cylinder at the start of combustion. Composition variations, cycle-by-cycle, in the vicinity of the spark plug electrode gap will affect the early stages of flame development, especially as the flame grows through the laminarlike burning phase following the creation of a small flame kernel by the spark discharge (see Sec. 9.5.1). Figure 9-34 indicates the extent of these composition nonuniformities. The available data comes from experiments where a small rapid-acting sampling valve located in the spark plug center electrode was used to extract gas from the vicinity of the electrode gap, close to the start of combustion, for individual cycle composition analysis. Figure 9-34*a* shows the cycle-by-cycle air/fuel ratio fluctuations in the burned gases sampled from one cylinder of a four-cylinder gasoline-fueled carbureted engine just after combustion has started. The standard deviation was typically 2 to 6 percent of the mean (A/F).[46–48] Figure 9-34*b* shows the relationship between total hydrocarbon and CO_2 concentrations in unburned mixture, sampled just before spark discharge. The CO_2 concentration is a measure of the burned gas fraction in the sampled unburned mixture; hence on average it correlates inversely with the total hydrocarbon concentration. However, there is substantial fluctuation in CO_2 concentration about the mean value, at a given fuel fraction, indicating significant fluctuations, cycle-by-cycle, in the mixing of fresh mixture with residual gas. Nonuniformities in EGR distribution between cylinders and EGR mixing within the cylinder would also increase the variations in burned gas fraction locally at the spark gap, cycle-by-cycle.[48]

Experiments in a multicylinder production SI engine, where the fuel/air ratio nonuniformities and the nonuniform mixing of fresh mixture with residual

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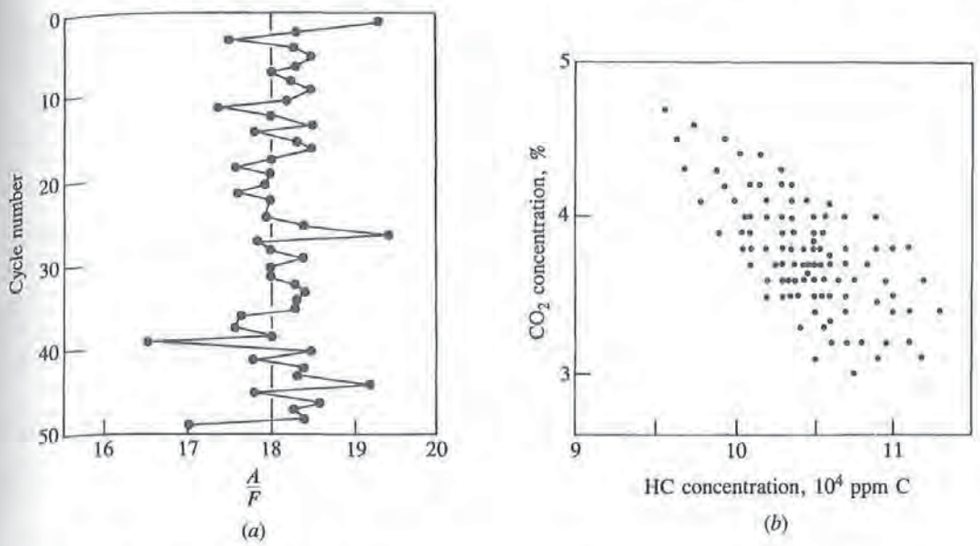


FIGURE 9-34
(*a*) Air/fuel ratio in 50 consecutive cycles, in vicinity of spark plug, measured just after ignition with a rapid-acting sampling valve located in the plug center electrode. Engine operated at 1400 rev/min, MBT timing, imep = 314 kPa.[47] (*b*) CO_2 and unburned HC concentrations in gas sampled in individual cycles from the vicinity of the spark plug just prior to ignition. Engine operated at 1200 rev/min, $\phi = 0.98$, $p_{inlet} = 0.5$ atm, gasoline fuel.[48]

gas were removed in turn as contributors to cycle-by-cycle variations (by comparing premixed propane operation with conventional carbureted operation with gasoline, and by removing residual gas by purging with nonfiring cycles), showed that the three contributing factors to cyclic combustion variations—velocity variations, fuel/air ratio variations, and residual gas mixing variations—are of comparable importance at road-load conditions.[45]

An explanation for cycle-by-cycle variations can be developed from the description of the turbulent flame propagation process in Sec. 9.3. Conditions in the *vicinity* of the spark plug will influence the initial stages of the flame propagation process—establishing a stable kernel and its development into a turbulent flame. During the developed flame propagation phase, the *average* conditions in the bulk gas within the combustion chamber will be the determining factors since the flame front spans the chamber, effectively averaging out local nonuniformities. By conditions are meant the turbulent velocity fluctuations and length scales in the flow, proportions of fuel, air, and burned gas in the mixture, and the mixture state.

Using the turbulent combustion model described in Sec. 14.4.2 (which is based on the description of the flame development and propagation process in Sec. 9.3), the flame development angle $\Delta\theta_d$ (the time to burn a few large eddies and establish a developed turbulent flame) can be expressed as[20]

$$\Delta\theta_d = C\left(\frac{l_I}{u'}\right)^{1/3}\left(\frac{l_M}{S_L}\right)^{2/3} \tag{9.51}$$

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The turbulent flow field influences $\Delta\theta_d$ through l_I, u', and l_M, the integral scale, the turbulence intensity, and the microscale, respectively. The mixture composition influences $\Delta\theta_d$ through the laminar flame speed S_L. There is therefore a variability in the flame development period, since all of these quantities can vary in the vicinity of the spark plug on a cycle-by-cycle basis.

The pressure development during the rapid-burning developed turbulent flame propagation phase, when the flame spans the combustion chamber, depends on the average rate of burning in the flame. Thus, variations in the turbulent flow field and mixture composition across the gas entering the flame front are averaged out and are not important. However, variation in chamber-average quantities are significant. The combustion model in Sec. 14.4.2 leads to the following expression for the maximum burning rate:

$$\left(\frac{dm_b}{dt}\right)_{\max} = \frac{Cm_f(h^*/B)(\rho_u^*/\rho_i)^{10/9}[(\bar{u}'^* S_L^*)/h_i]^{2/3}}{\nu^{*1/3}} \tag{9.52}$$

Here, m_f is the mass of fuel in the chamber, h is the instantaneous (mean) clearance height, B is the bore, ρ the density, $\bar{u}'$ the average turbulence intensity across the flame front, and ν the kinematic viscosity; the * denotes the value at the time of the maximum burning rate and the subscript i the value at spark; C is a constant depending on engine design. It can be seen from Eq. (9.52) that cycle-by-cycle variations in the maximum burning rate can result from variations in the overall flow pattern within the combustion chamber (which vary $\bar{u}'^*$) and from variations in the amount of fuel (m_f) that enters the cylinder each cycle.† Also, it can be seen that variations in the flame development process will result in variations in the maximum burning rate because the crank angle at which the maximum burning rate occurs is shifted, and all the starred parameters in Eq. (9.52) will have different values.

From the discussion of flame development and structure in Sec. 9.3, and Eqs. (9.51) and (9.52) above, we would expect that mixture conditions and motion leading to slower flame development rates (longer flame development angles, $\Delta\theta_d$)—lower turbulence intensities and more dilute mixtures—would also give lower burning rates (longer rapid burning angles, $\Delta\theta_b$). Data from many different engines and a wide range of operating cases show that this is the case, on average, though there is substantial variation about the mean trend. Figure 9-35 shows these trends; it also shows that the standard deviation of $\Delta\theta_d$ and the standard deviation of $\Delta\theta_b$ for a given chamber and operating condition generally increase as the average burning process becomes slower.[40]

One final factor of importance is how variations in flame development and burning rate affect engine torque. With fixed spark timing, such variations in the combustion process cycle-by-cycle result in slower developing and/or burning

† Variations in the total amount of air, recycled exhaust gas, and residual in the chamber could also, for some operating regimes, be significant.

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cycles being retarded and faster developing and/or burning cycles being overadvanced. The curve of torque versus combustion timing (relative to optimum timing), Fig. 9-3*b*, is almost independent of the burning rate; i.e., a given magnitude retard (of say 10°) relative to optimum timing gives almost the same reduction in torque for a very fast burn as it does for a very slow burn. This is because the burning process, for optimum timing, is centered at about 10° ATC independent of the burn rate, and retard or advance shifts this "center" by equal amounts for all burn rates.[16] One of the major advantages of fast-burn engines is now apparent. The magnitude of the variations in the flame development process and subsequent flame propagation rate are decreased as the burning rate is increased (see Fig. 9-35): the ratio of standard deviation in $\Delta\theta_d$ and $\Delta\theta_b$ to the mean values remains approximately constant. Thus, these smaller combustion variations in fast-burn engines, which correspond to modest retard and overadvance in nonaverage burn rate cycles, have little effect on torque. In contrast, the larger combustion variations of slow-burning engines result in significant cyclic torque variations.

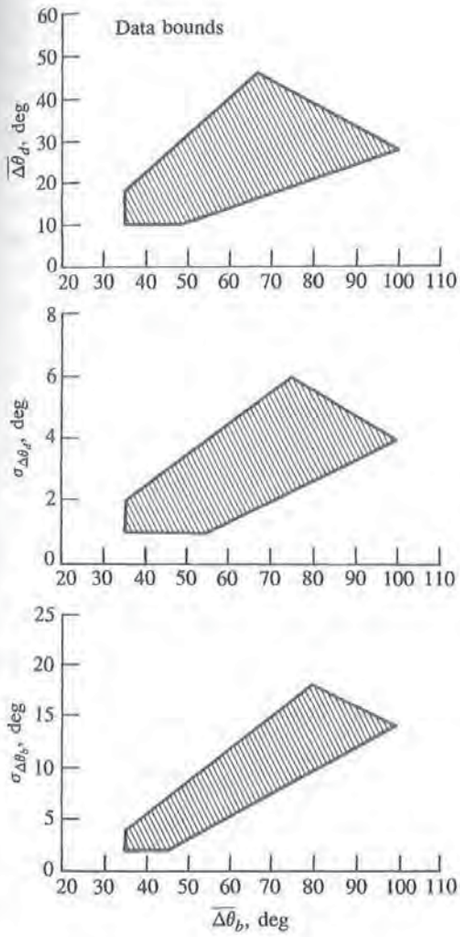


FIGURE 9-35
Variation in mean value of flame development angle $\overline{\Delta\theta_d}$ (spark to 1 percent mass burned) and standard deviations of flame development angle and rapid burning angle $\Delta\theta_b$ (1 to 90 percent mass burned) with mean rapid burning angle $\overline{\Delta\theta_b}$. Range of combustion chamber geometries and engine operating conditions.[40]

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9.4.3 Partial Burning, Misfire, and Engine Stability

As the unburned mixture in a spark-ignition engine is leaned out with excess air or is diluted with increasing amounts of burned residual gas and exhaust gas recycle, the flame development period, the duration of the rapid burning phase, and the cycle-by-cycle fluctuations in the combustion process all increase. Eventually a point is reached where engine operation becomes rough and unstable, and hydrocarbon emissions increase rapidly. The point at which these phenomena occur effectively defines the engine's *stable operating limit.*† These phenomena result from the lengthening of all stages of the combustion process as the unburned mixture is diluted. With increasing dilution, first a fraction of the cycles burns so slowly that combustion is only just completed prior to exhaust valve opening. Then as burning lengthens further, in some cycles there is insufficient time to complete combustion within the cylinder; also, flame extinguishment before the exhaust valve opens and before the flame has propagated across the chamber may start to occur in some cycles. Finally, misfiring cycles where the mixture never ignites may start to occur. The proportion of partial burning or nonburning cycles increases rapidly if the mixture is made even more lean or dilute, and the point is soon reached where the engine will not run at all.

The impact on engine stability of increasing combustion variability, due to increased exhaust gas recycle at part-load, is shown in Fig. 9-36. Figure 9-36*a* shows the distributions of individual-cycle indicated mean effective pressure values for 0, 20, and 28 percent EGR. Without EGR at these conditions, the spread in imep is narrow. Increasing EGR widens the distribution significantly and cycles with low imep, and eventually zero imep, occur. Figure 9-36*b* shows how the coefficient of variation of imep and hydrocarbon emissions increase as EGR is increased. Slow burn, then partial burn, and then misfire cycles occur with increasing frequency. In the slow-burn cycles, combustion is complete but ends after 80° ATC and the indicated mean effective pressure is low (between 85 and 46 percent of the mean value). Imep in partial-burn cycles was less than 46 percent of the mean. In misfiring cycles, imep < 0. Empirically, it has been found that COV_{imep} [see Eq. (9.50)] is about 10 percent at the engine's stable operating limit, which here occurs just before the onset of partial-burning cycles[49].

An explanation of combustion phenomenon at the engine stable operating limit has been developed by Quader.[50] It involves the following terms:

> **Ignition-limited spark timing** or the **ignition limit.** The spark timing [advanced from maximum brake torque (MBT) timing] at which *misfire* (i.e., failure of flame initiation) first occurs at a given mixture composition, in a given small but arbitrary fraction of cycles (e.g., 0.5 to 1 percent).

† This limit has often been called the *lean operating limit.* Since what limits engine operation in practice is excessive torque fluctuations, cycle-by-cycle, and high hydrocarbon emissions, resulting from the use of mixtures made overly dilute with either air or burned gases (or with both), *stable operating limit* is a more appropriate term.

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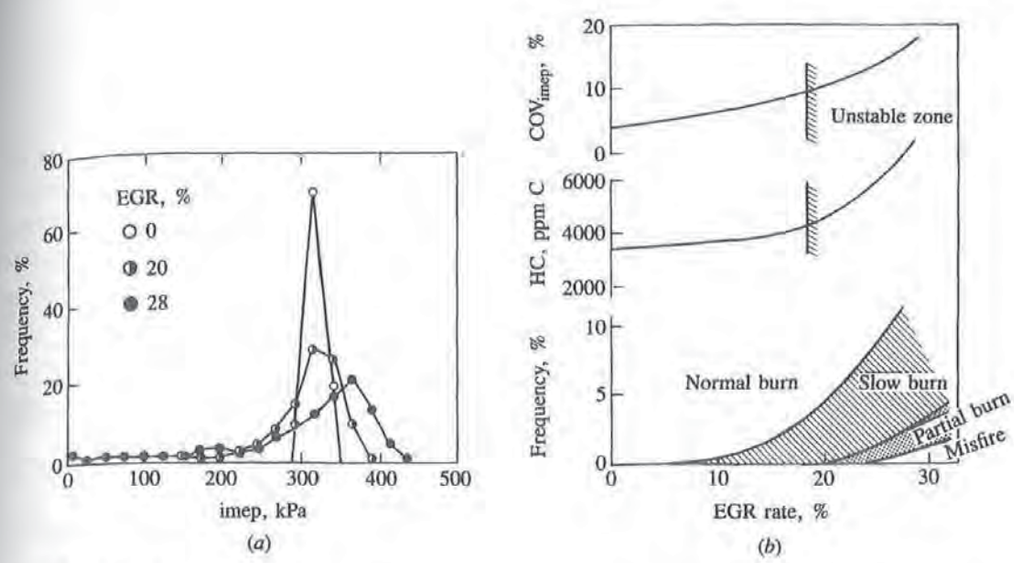


FIGURE 9-36
(*a*) Frequency distributions in indicated mean effective pressure at different EGR rates; 0 percent gave excellent engine stability, 20 percent acceptable stability, and 28 percent poor stability. (*b*) Coefficient of variation in imep, HC emissions, and percentage of normal, slow, partial-burn, and misfire cycles. Engine conditions: 1400 rev/min, $\phi = 1.0$, MBT timing, imep = 324 kPa.[49]

Partial-burn-limited spark timing or the **partial-burn limit.** The spark timing (retarded from MBT) at which incomplete flame propagation occurs at a given mixture composition in a given small percent of the cycles (again, this frequency is selected arbitrarily for experimental convenience).

Lean misfire limit at MBT spark. The leanest mixture stoichiometry at which the engine could be stabilized to operate at MBT spark timing with a misfire frequency below a specified value (again, this frequency, usually a percent or less of the cycles, is selected arbitrarily for convenience). A *dilute misfire limit*, the maximum amount of exhaust gas recycle that can be absorbed at a given stoichiometry for stable engine operation, can be similarly defined.

Engine experiments have defined the locations of the ignition limit line and the partial-burn limit line; they are shown qualitatively in Fig. 9-37. At a given spark timing, on this spark timing versus equivalence ratio plot, progressive leaning of the mixture fed to the engine will lead to the onset of misfire or to the onset of partial burning, depending on the location of the lines and the spark timing selected. The individual figures show the possible interactions of the maximum brake torque (MBT) timing line—leaner mixtures require greater advance—with the ignition limit and partial-burn limit lines. At MBT timing, the partial-burn limit may or may not be reached prior to misfire or the ignition limit. It will depend on the engine and ignition system design and operating conditions. For spark timings retarded relative to MBT, partial burning and not failure of flame initiation is the primary cause of unstable engine operation.

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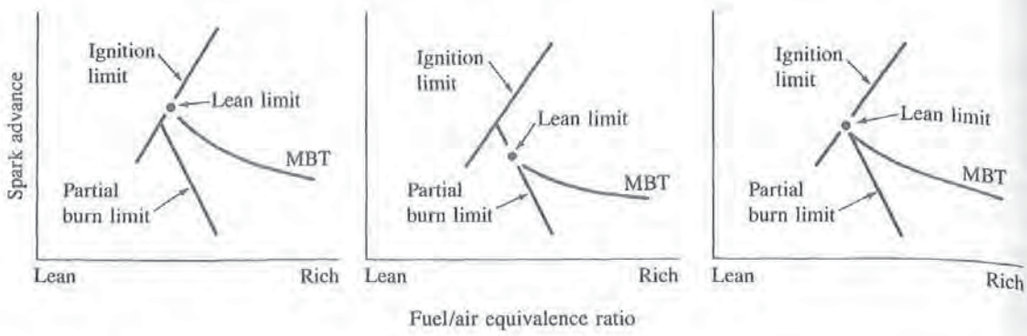


FIGURE 9-37
Schematics of three possible combinations of ignition limit, partial-burn limit, and MBT timing curves as function of fuel/air equivalence ratio. (*From Quader.*[50])

An example of these limiting combustion regimes for lean engine operation is shown in Fig. 9-38. With MBT timing, as the mixture is leaned out (at constant air flow rate), complete combustion in all cycles changes to partial burning in some cycles which changes to no ignition in some cycles at the ignition limit. In the partial-burning regime, the most common type of incomplete-combustion cycle was a slow-burning cycle which required more time to complete burning than was available: flame extinguishment during expansion was much less common. Engine performance measurements showed that the engine stability limit—evidenced by minimum fuel consumption and onset of rapid increase in HC emissions—occurred at $\phi = 0.65$, just before the partial-burn limit line where some slow-burning cycles occur but combustion is still complete in all cycles.[44]

From the above it is clear that flame initiation is a necessary but not sufficient condition for complete combustion. Too-slow flame development and prop-

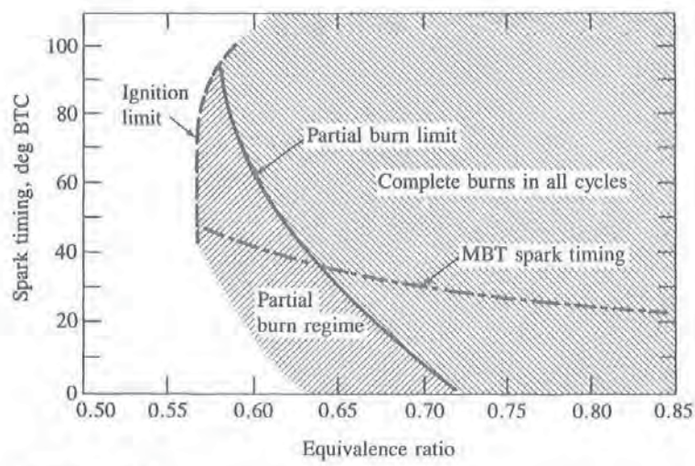


FIGURE 9-38
Actual limiting combustion regimes for lean-operating engine. 1200 rev/min, volumetric efficiency ≈ 60 percent, methane fuel, 40 mJ spark energy, 2.5 ms spark duration.[44]

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agation following successful ignition is usually the factor which limits engine operation with dilute mixtures. Experiments have shown that a limited interval, of order 80 crank angle degrees (depending on engine geometry and spark plug location), is available during the engine cycle when conditions are favorable for complete flame propagation.[49, 51] Outside of this interval, the mixture pressure and temperature are too low, and the turbulence intensity is too low to sustain a sufficiently rapid rate of combustion. Thus, it is factors which increase the flame development and propagation rates which primarily extend the partial-burn limit.

9.5 SPARK IGNITION

In spark-ignition engines, the electrical discharge produced between the spark plug electrodes by the ignition system starts the combustion process close to the end of the compression stroke. The high-temperature plasma kernel created by the spark develops into a self-sustaining and propagating flame front—a thin reaction sheet where the exothermic combustion chemical reactions occur. The function of the ignition system is to initiate this flame propagation process, in a repeatable manner cycle-by-cycle, over the full load and speed range of the engine at the appropriate point in the engine cycle. Shadowgraph and schlieren photographs of the kernel created by the discharge between the plug electrodes, the growth of that kernal, and its transition to a propagating flame have already been presented in Figs. 9-19 and 9-20, and described in the accompanying text. A spark can arc from one electrode to another when a sufficiently high voltage is applied. Ignition systems commonly used to provide this spark are: battery ignition systems where the high voltage is obtained with an ignition coil (coil ignition systems); battery systems where the spark energy is stored in a capacitor and transferred as a high-voltage pulse to the spark plug by means of a special transformer (capacitive-discharge ignition systems); and magneto ignition systems where the magneto—a rotating magnet or armature—generates the current used to produce a high-voltage pulse.

This section reviews our basic understanding of electrical discharges in inflammable gas mixtures relevant to engine ignition (Sec. 9.5.1), the major design and operating characteristics of conventional engine ignition systems (Sec. 9.5.2), and, briefly, some alternative approaches to generating a propagating flame (Sec. 9.5.3).

9.5.1 Ignition Fundamentals

A spark can arc from one plug electrode to the other only if a sufficiently high voltage is applied. In a typical spark discharge, the electrical potential across the electrode gap is increased until breakdown of the intervening mixture occurs. Ionizing streamers then propagate from one electrode to the other. The impedance of the gap decreases drastically when a streamer reaches the opposite electrode, and the current through the gap increases rapidly. This stage of the

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discharge is called the *breakdown phase.* It is followed by the *arc phase*, where the thin cylindrical plasma expands largely due to heat conduction and diffusion and, with inflammable mixtures, the exothermic reactions which lead to a propagating flame develop. This may be followed by a *glow discharge phase* where, depending on the details of the ignition system, the energy storage device (e.g., the ignition coil) will dump its energy into the discharge circuit.[52, 53]

Figure 9-39 shows the behavior of the discharge voltage and current as a function of time for a conventional coil ignition system. Typical values are shown; actual values depend on the details of the electrical components. The breakdown phase is characterized by a high-voltage (~10 kV), high-peak current (~200 A), and an extremely short duration (~10 ns). A narrow (~40 μm diameter) cylindrical ionized gas channel is established very early. The energy supplied is transferred almost without loss to this plasma column. The temperature and pressure in the column rise very rapidly to values up to about 60,000 K and a few hundred atmospheres, respectively. A strong shock or blast wave propagates outward, the channel expands, and, as a result, the plasma temperature and pressure fall. Some 30 percent of the plasma energy is carried away by the shock wave; however, most of this is regained since spherical blast waves transfer most of their energy to the gas within a small (~2 mm diameter) sphere into which the breakdown plasma soon expands.[52, 53]

A breakdown phase always precedes arc and glow discharges: it creates the electrically conductive path between the electrodes. The arc phase voltage is low

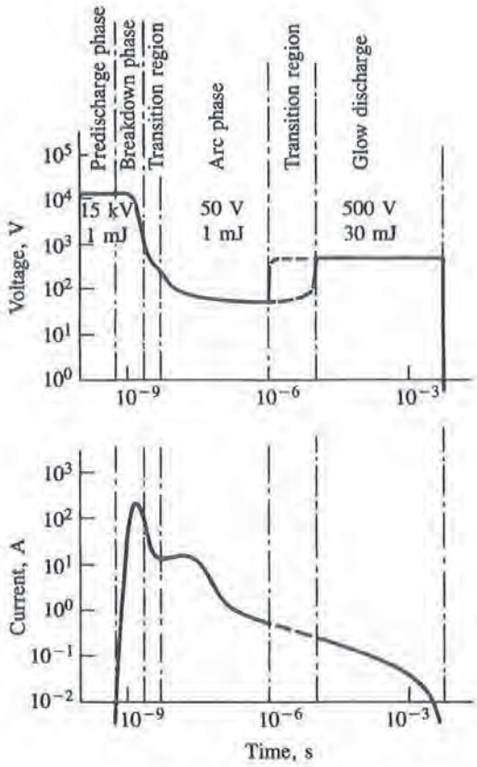


FIGURE 9-39
Schematic of voltage and current variation with time for conventional coil spark-ignition system. Typical values for energy and voltage in the three phases—breakdown, arc, and glow discharge—are given.[52]

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(<100 V), though the current can be as high as the external circuit permits. In contrast to the breakdown phase where the gas in the channel is fully dissociated and ionized, in the arc phase the degree of dissociation may still be high at the center of the discharge, but the degree of ionization is much lower (about 1 percent). Voltage drops at the cathode and anode electrodes are a significant fraction of the arc voltage, and the energy deposited in these electrode sheath regions, which is conducted away by the metal electrodes, is a substantial fraction of the total arc energy (see Table 9.4 below). The arc requires a hot cathode spot, so evaporation of the cathode material occurs. The arc increases in size due primarily to heat conduction and mass diffusion. Due to these energy transfers the gas temperature in the arc is limited to about 6000 K: the temperature and degree of dissociation decrease rapidly with increasing distance from the arc axis. Currents less than 200 mA, a large electrode voltage drop at the cathode (300 to 500 V), a cold cathode, and less than 0.01 percent ionization are typical for the glow discharge. Energy losses are higher than in the arc phase, and peak equilibrium gas temperatures are about 3000 K.[52]

About 0.2 mJ of energy is required to ignite a quiescent stoichiometric fuel-air mixture at normal engine conditions by means of a spark. For substantially leaner and richer mixtures, and where the mixture flows past the electrodes, an order of magnitude greater energy (~ 3 mJ) may be required.[54] Conventional ignition systems deliver 30 to 50 mJ of electrical energy to the spark. Due to the physical characteristics of the discharge modes discussed above, only a fraction of the energy supplied to the spark gap is transmitted to the gas mixture. The energy balance for the breakdown, arc, and glow phases of the discharge is given in Table 9.4. Radiation losses are small throughout. The end of the breakdown phase occurs when a hot cathode spot develops, turning the discharge into an arc; heat losses to the electrodes then become substantial. The breakdown phase reaches the highest power level (~ 1 MW), but the energy supplied is small (0.3 to 1 mJ). The glow discharge has the lowest power level (~ 10 W) but the highest energy (30 to 100 mJ), due to its long discharge time. The arc phase lies between.

The proportions of the electrical energy supplied, which can be transferred to the plasma in these three phases of the discharge, are shown in Fig. 9-40.[53, 55] The different transfer capabilities for breakdown arc and glow discharges arise

TABLE 9.4
Energy distribution for breakdown, arc, and glow discharges†

	Breakdown, %	Arc, %	Glow, %
Radiation loss	<1	5	<1
Heat loss to electrodes	5	45	70
Total losses	6	50	70
Plasma energy	94	50	30

† Typical values, under idealized conditions with small electrodes.
Source: Maly and Vogel.[52]

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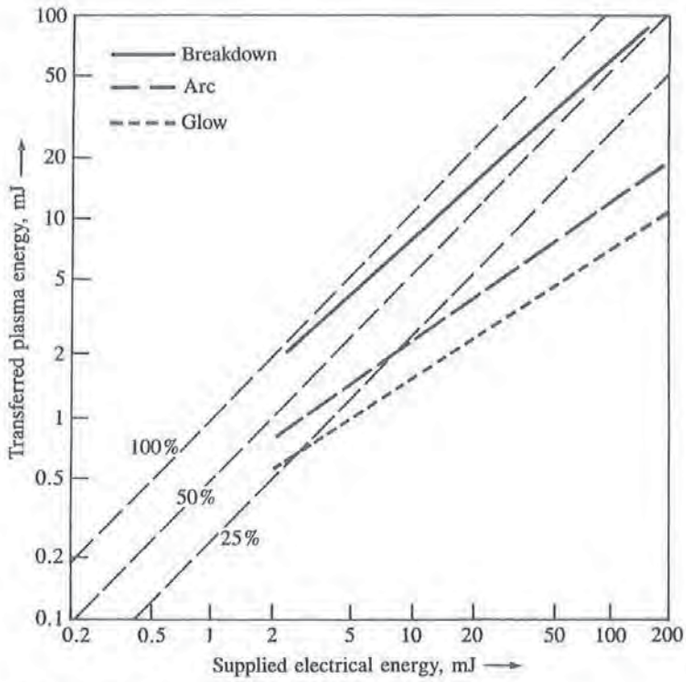


FIGURE 9-40
Energy transferred to the spark kernal as a function of supplied electrical energy for breakdown, arc, and glow discharges.[55]

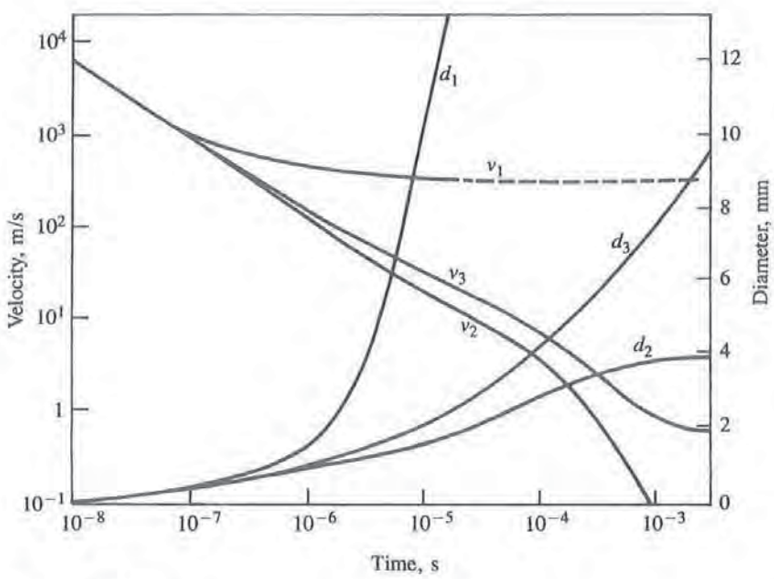


FIGURE 9-41
Diameters (d) and expansion velocities (v) of volumes activated by a capacitive-discharge ignition system: 3 mJ electrical energy, 100 μs duration. Subscripts denote: 1, shock in air at 1 atm; 2, plasma in air at 1 atm; 3, electrical and chemical plasma in stoichiometric methane-air mixture at 1 atm.[52]

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