Lecture 7
ENGINE CYCLES (Otto Cycle)

INTERNAL COMBUSTION ENGINES

By

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CHAPTER 3
ENGINE CYCLES

3.1 AIR - STANDARD CYCLES

The cycle experienced in the cylinder of an internal combustion engine is very complex. First, air (CI engine) or air mixed with fuel (SI engine) is ingested and mixed with the slight amount of exhaust residual remaining from the previous cycle. This mixture is then compressed and combusted, changing the composition to exhaust products consisting largely of CO_2, H_2O, and N_2 with many other lesser components. Then, after an expansion process, the exhaust valve is opened and this gas mixture is expelled to the surroundings. Thus, **it is an open cycle with changing composition**, a difficult system to analyze. To make the analysis of the engine cycle much more manageable,

The real cycle is approximated with an ideal air-standard cycle which differs from the actual by some simplifying assumptions:

1. Cylinder contains constant amount of air and it is treated as ideal gas.
2. The specific heats and other physical and chemical properties remain unchanged during the cycle.
3. Instead of heat generation by combustion, heat is added from external heat source Q_{in}.
4. The process of heat removal in the exhaust gases is represented by heat transfer from the cycle to external heat sink Q_{out}.
5. There is neither friction nor turbulence; all processes are assumed to be reversible.
6. No heat loss from the working fluid to the surroundings.
7. Cycles can be presented on any diagram of properties.

In air-standard cycles, air is considered an ideal gas such that the following ideal gas relationships can be used:

$$Pv = RT \quad \text{(a)}$$

$$PV = mRT \quad \text{(b)}$$

$$P = \rho RT \quad \text{(c)}$$

$$dh = c_P dT \quad \text{(d)}$$

$$du = c_v dT \quad \text{(e)}$$

$$Pv^k = \text{constant} \qquad \text{isentropic process} \quad \text{(f)}$$

$$Tv^{k-1} = \text{constant} \qquad \text{isentropic process} \quad \text{(g)}$$

$$TP^{(1-k)/k} = \text{constant} \qquad \text{isentropic process} \quad \text{(h)}$$

$$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k) \quad \text{isentropic work in closed system} \quad \text{(i)}$$

$$= R(T_2 - T_1)/(1-k)$$

$$c = \sqrt{kRT} \qquad \text{speed of sound} \quad \text{(j)}$$

(3-1)

where:
P = gas pressure in cylinder
V = volume in cylinder
v = specific volume of gas
R = gas constant of air
T = temperature
m = mass of gas in cylinder
ρ = density
h = specific enthalpy
u = specific internal energy
c_p, c_v = specific heats
$k = c_p/c_v$
w = specific work
c = speed of sound

In addition to these, the following variables are used in this chapter for cycle analysis:

AF = air–fuel ratio
$\dot{m}$ = mass flow rate
q = heat transfer per unit mass for one cycle
$\dot{q}$ = heat transfer rate per unit mass
Q = heat transfer for one cycle
$\dot{Q}$ = heat transfer rate
Q_{HV} = heating value of fuel

r_c = compression ratio
W = work for one cycle
$\dot{W}$ = power
η_c = combustion efficiency

Subscripts used include:

a = air
f = fuel
ex = exhaust

For thermodynamic analysis the specific heats of air can be treated as functions of temperature. A constant specific heat analysis will be used. Because of the high temperatures and large temperature range experienced during an engine cycle, the specific heats and ratio of specific heats k do vary by a fair amount. At the low-temperature end of a cycle during intake and start of compression, a value of k = 1.4 is correct. However, at the end of combustion the temperature has risen such that k = 1.3 would be more accurate. A constant average value between these extremes is found to give better results than a standard condition (25°C) value, as is often used in elementary thermodynamics textbooks. **When analyzing what occurs within engines <u>during the operating cycle and exhaust flow</u>, here will use the following air property values:**

$$c_p = 1.108 \text{ kJ/kg-K} = 0.265 \text{ BTU/lbm-°R}$$
$$c_v = 0.821 \text{ kJ/kg-K} = 0.196 \text{ BTU/lbm-°R}$$
$$k = c_P/c_v = 1.108/0.821 = 1.35$$
$$R = c_P - c_v = 0.287 \text{ kJ/kg-K}$$

Air flow before it enters an engine is usually closer to standard temperature, and for these conditions a value of k = 1.4 is correct. This would include processes such as inlet flow in superchargers, turbochargers, and carburetors, and air flow through the engine radiator. For these conditions, the following air property values are used:

$$c_p = 1.005 \text{ kJ/kg-K} = 0.240 \text{ BTU/lbm-°R}$$
$$c_v = 0.718 \text{ kJ/kg-K} = 0.172 \text{ BTU/lbm-°R}$$
$$k = c_p/c_v = 1.005/0.718 = 1.40$$
$$R = c_p - c_v = 0.287 \text{ kJ/kg-K}$$

3.1.1 OTTO CYCLE

The cycle of a four-stroke, SI, naturally aspirated engine at WOT. This is the cycle of most automobile engines and other four-stroke SI engines. For analysis, this cycle is approximated by the air-standard cycle shown in **Fig. 3-l.** This ideal air-standard cycle is called an **Otto cycle,** named after one of the early developers of this type of engine.

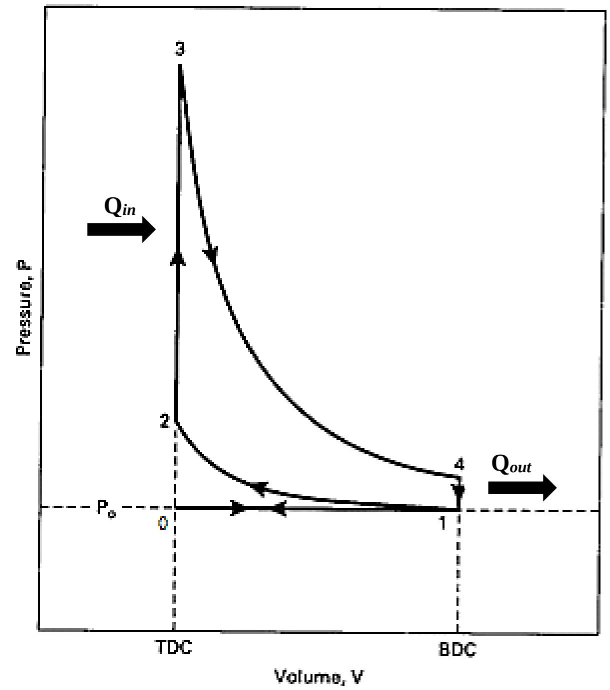


Figure 3-1 Ideal air-standard Otto cycle, 0-1-2-3-4-1-0 which approximates the four-stroke cycle of an SI engine on P-V coordinates.

Thermodynamic Analysis of Air-Standard Otto Cycle

Process 0-1—constant-pressure intake of air at P_o.
Intake valve open and exhaust valve closed:

$$P_1 = P_o \tag{3-2}$$

$$w_{o-1} = P_o(v_1 - v_o) \tag{3-3}$$

Process 1-2—isentropic compression stroke.
All valves closed:

$$T_2 = T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \tag{3-4}$$

$$P_2 = P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k \tag{3-5}$$

$$q_{1-2} = 0 \tag{3-6}$$

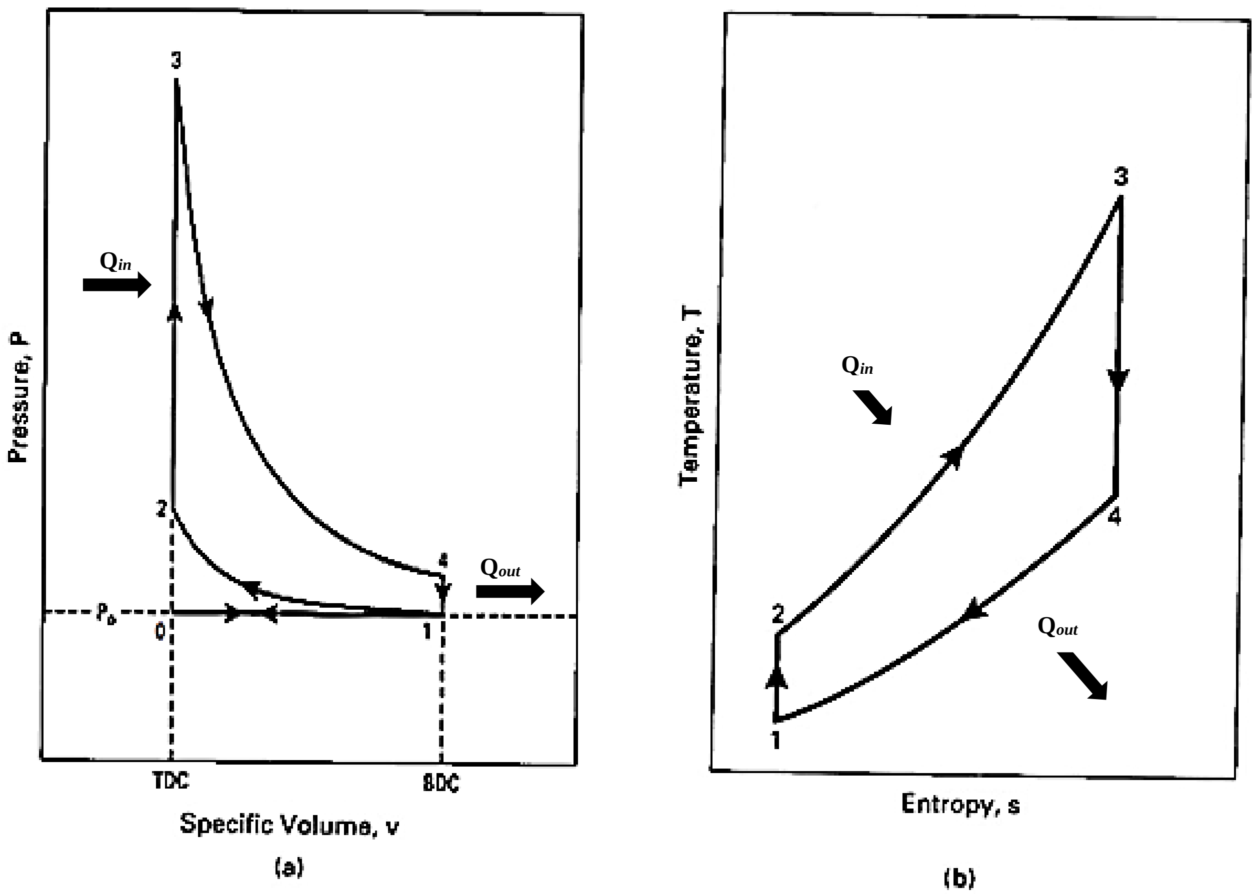
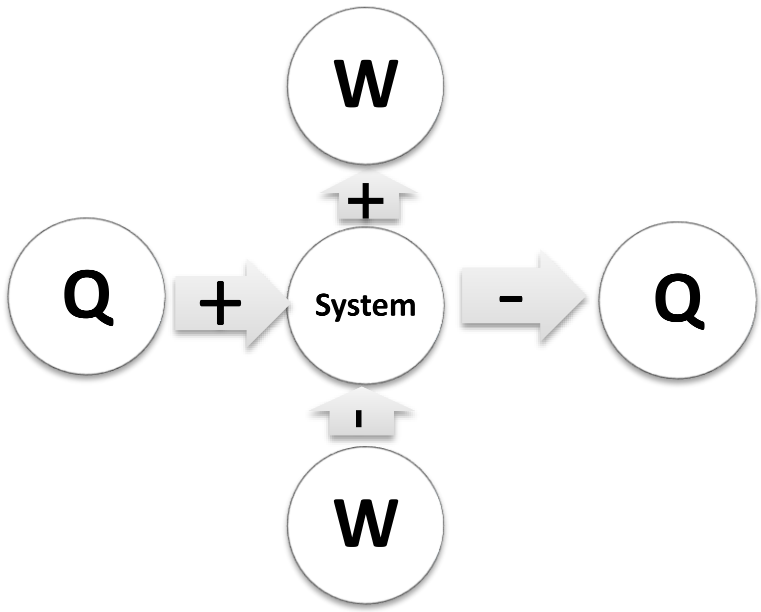


Figure 3-2 Otto cycle, 0-1-2-3-4-1-0 , on (a) pressure-specific volume coordinates, and (b) temperature-entropy coordinates.

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1-k) = R(T_2 - T_1)/(1-k) \quad (3\text{-}7)$$
$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

Process 2-3—constant-volume heat input (combustion).
All valves closed:

$$v_3 = v_2 = v_{TDC} \quad (3\text{-}8)$$
$$w_{2-3} = 0 \quad (3\text{-}9)$$
$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_c = m_m c_v (T_3 - T_2) \quad (3\text{-}10)$$
$$= (m_a + m_f) c_v (T_3 - T_2)$$
$$Q_{HV} \eta_c = (AF + 1) c_v (T_3 - T_2) \quad (3\text{-}11)$$
$$q_{2-3} = q_{in} = c_v (T_3 - T_2) = (u_3 - u_2) \quad (3\text{-}12)$$
$$T_3 = T_{max} \quad (3\text{-}13)$$
$$P_3 = P_{max} \quad (3\text{-}14)$$



Process 3-4—isentropic power or expansion stroke.
All valves closed:

$$q_{3-4} = 0 \tag{3-15}$$

$$T_4 = T_3(v_3/v_4)^{k-1} = T_3(V_3/V_4)^{k-1} = T_3(1/r_c)^{k-1} \tag{3-16}$$

$$P_4 = P_3(v_3/v_4)^k = P_3(V_3/V_4)^k = P_3(1/r_c)^k \tag{3-17}$$

(+) $$w_{3-4} = (P_4v_4 - P_3v_3)/(1-k) = R(T_4 - T_3)/(1-k) \tag{3-18}$$

$$= (u_3 - u_4) = c_v(T_3 - T_4)$$

Process 4-1 —constant-volume heat rejection (exhaust blowdown).
Exhaust valve open and intake valve closed:

$$v_4 = v_1 = v_{BDC} \tag{3-19}$$

$$w_{4-1} = 0 \tag{3-20}$$

(-) $$Q_{4-1} = Q_{out} = m_m c_v(T_4 - T_1) \tag{3-21}$$

$$q_{4-1} = q_{out} = c_v(T_4 - T_1) = (u_4 - u_1) \tag{3-22}$$

Process 1-0—constant-pressure exhaust stroke at P_o.
Exhaust valve open and intake valve closed:

$$P_1 = P_o \tag{3-23}$$

(-) $$w_{1-o} = P_o(v_o - v_1) \tag{3-24}$$

The thermal efficiency of Otto cycle:

$$(\eta_t)_{Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \tag{3.25}$$

$$= 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2}$$

$$= 1 - \frac{(T_4 - T_1)}{(T_3 - T_2}$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} \quad \rightarrow T_2 = T_1(r_c)^{k-1} \tag{3.26}$$

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{k-1} = \left(\frac{v_1}{v_2}\right)^{k-1} \quad \rightarrow \ T_3 = T_4(r_c)^{k-1} \tag{3.27}$$

By substitution equations (3.26) and (3.27) in eq.(3.25) we get:

$$(\eta_t)_{Otto} = 1 - \frac{1}{(r_c)^{k-1}} \tag{3.28}$$

$$(r_c)^{k-1} = \frac{T_2}{T_1}$$

$$(\eta_t)_{Otto} = 1 - \frac{T_1}{T_2} \tag{3.29}$$

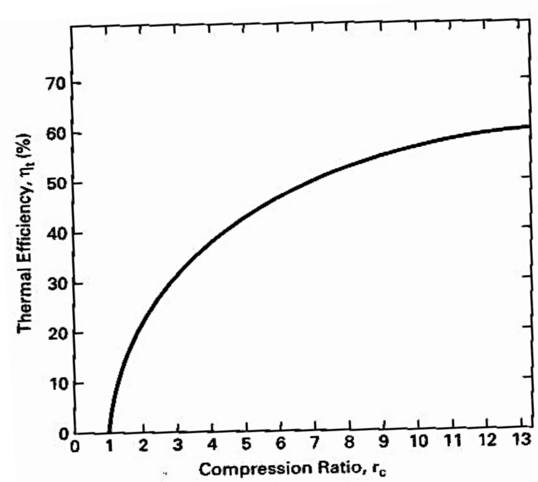


Figure 3-3 Indicated thermal efficiency as a function of compression ratio for SI engines operating at WOT on air-standard Otto cycle ($k = 1.35$).

EXAMPLE PROBLEM 3-1

A four-cylinder, 2.5-liter, SI automobile engine operates at WOT on a four-stroke air-standard Otto cycle at 3000 RPM. The engine has a compression ratio of 8.6:1, a mechanical efficiency of 86%, and a stroke-to-bore ratio $S/B = 1.025$. Fuel is isooctane with AF = 15, a heating value of 44,300 kJ/kg, and combustion efficiency $\eta_c = 100\%$. At the start of the compression stroke, conditions in the cylinder combustion chamber are 100 kPa and 60°C. It can be assumed that there is a 4% exhaust residual left over from the previous cycle.

Do a complete thermodynamic analysis of this engine.

For one cylinder:

Displacement volume:

$$V_d = 2.5 \text{ liter}/4 = \underline{0.625 \text{ L} = 0.000625 \text{ m}^3}$$

Using Eq. (2-12) to find clearance volume:

$$r_c = V_1/V_2 = (V_c + V_d)/V_c = 8.6 = (V_c + 0.000625)/V_c$$

$$\underline{V_c = 0.0000822 \text{ m}^3 = 0.0822 \text{ L} = 82.2 \text{ cm}^3}$$

Using Eq. (2-8) to find bore and stroke:

$$V_d = (\pi/4)B^2S = (\pi/4)B^2(1.025B) = 0.000625 \text{ m}^3$$

$$\underline{B = 0.0919 \text{ m} = 9.19 \text{ cm}}$$

$$S = 1.025B = \underline{0.0942 \text{ m} = 9.42 \text{ cm}}$$

State 1:

$\underline{T_1 = 60°\text{C} = 333 \text{ K}}$ given in problem statement

$\underline{P_1 = 100 \text{ kPa}}$ given

$$V_1 = Vd + V_c = 0.000625 + 0.0000822 = \underline{0.000707 \text{ m}^3}$$

Mass of gas mixture in cylinder can be calculated at State 1. The mass within the cylinder will then remain the same for the entire cycle.

$$m_m = P_1V_1/RT_1 = (100 \text{ kPa})(0.000707 \text{ m}^3)/(0.287 \text{ kJ/kg-K})(333 \text{ K})$$

$$= 0.000740 \text{ kg}$$

State 2: The compression stroke 1-2 is isentropic. Using Eqs. (3-4) and (3-5) to find pressure and temperature:

$$P_2 = P_1(r_c)^k = (100\text{ kPa})(8.6)^{1.35} = \underline{1826\text{ kPa}}$$

$$T_2 = T_1(r_c)^{k-1} = (333\text{ K})(8.6)^{0.35} = \underline{707\text{ K} = 434°\text{C}}$$

$$V_2 = mRT_2/P_2 = (0.000740\text{ kg})(0.287\text{ kJ/kg-K})(707\text{ K})/(1826\text{ kPa})$$

$$= \underline{0.0000822\text{ m}^3 = V_c}$$

This is the clearance volume of one cylinder, which agrees with the above. Another way of getting this value is to use Eq. (2-12):

$$V_2 = V_1/r_c = 0.000707\text{ m}^3/8.6 = 0.0000822\text{ m}^3$$

The mass of gas mixture m_m in the cylinder is made up of air m_a, fuel m_f, and exhaust residual m_{ex}:

mass of air	$m_a = (15/16)(0.96)(0.000740)$	$= 0.000666$ kg
mass of fuel	$m_f = (1/16)(0.96)(0.000740)$	$= 0.000044$ kg
mass of exhaust	$m_{ex} = (0.04)(0.000740)$	$= 0.000030$ kg
Total		$m_m = 0.000740$ kg

State 3: Using Eq. (3-10) for the heat added during one cycle:

$$Q_{in} = m_f Q_{HV} \eta_c = m_m c_v (T_3 - T_2)$$

$$= (0.000044\text{ kg})(44{,}300\text{ kJ/kg})(1.00)$$

$$= (0.000740\text{ kg})(0.821\text{ kJ/kg-K})(T_3 - 707\text{ K})$$

Solving this for T_3:

$$\underline{T_3 = 3915\text{ K} = 3642°\text{C} = T_{max}}$$

$$V_3 = V_2 = \underline{0.0000822\text{ m}^3}$$

For constant volume:

$$P_3 = P_2(T_3/T_2) = (1826\text{ kPa})(3915/707) = \underline{10{,}111\text{ kPa} = P_{max}}$$

State 4: Power stroke 3-4 is isentropic. Using Eq. (3-16) and (3-17) to find temperature and pressure:

$$T_4 = T_3(1/r_c)^{k-1} = (3915\ \text{K})(1/8.6)^{0.35} = \underline{1844\ \text{K} = 1571°\text{C}}$$

$$P_4 = P_3(1/r_c)^{k} = (10{,}111\ \text{kPa})(1/8.6)^{1.35} = \underline{554\ \text{kPa}}$$

$$V_4 = mRT_4/P_4 = (0.000740\ \text{kg})(0.287\ \text{kJ/kg-K})(1844\ \text{K})/(554\ \text{kPa})$$

$$\underline{= 0.000707\ \text{m}^3 = V_1}$$

This agrees with the value of V_1 found earlier.

Work produced in isentropic power stroke for one cylinder during one cycle:

$$W_{3-4} = mR(T_4 - T_3)/(1-k)$$

$$= (0.000740\ \text{kg})(0.287\ \text{kJ/kg-K})(1844 - 3915)\text{K}/(1 - 1.35)$$

$$= 1.257\ \text{kJ}$$

Work absorbed during isentropic compression stroke for one cylinder during one cycle:

$$W_{1-2} = mR(T_2 - T_1)/(1-k)$$

$$= (0.000740\ \text{kg})(0.287\ \text{kJ/kg-K})(707 - 333)\text{K}/(1 - 1.35)$$

$$= -0.227\ \text{kJ}$$

Work of the intake stroke is canceled by work of the exhaust stroke.

Net indicated work for one cylinder during one cycle is:

$$W_{net} = W_{1-2} + W_{3-4} = (+1.257) + (-0.227) = \underline{+1.030\ \text{kJ}}$$

Using Eq. (3-10) to find heat added for one cylinder during one cycle:

$$Q_{in} = m_f Q_{HV}\,\eta_c = (0.000044\ \text{kg})(44{,}300\ \text{kJ/kg})(1.00) = \underline{1.949\ \text{kJ}}$$

Indicated thermal efficiency:

$$\eta_t = W_{net}/Q_{in} = 1.030/1.949 = \underline{0.529 = 52.9\%}$$

or using Eqs. (3-29) and (3-31):

$$\eta_t = 1 - (T_1/T_2) = 1 - (1/r_c)^{k-1}$$

$$= 1 - (333/707) = 1 - (1/8.6)^{0.35} = 0.529$$

Equation (2-29) is used to find indicated mean effective pressure:

$$\text{imep} = W_{net}/(V_1 - V_2) = (1.030\ \text{kJ})/(0.000707 - 0.0000822)\text{m}^3 = \underline{1649\ \text{kPa}}$$

Indicated power at 3000 RPM is obtained using Eq. (2-42):

$$\dot{W}_i = WN/n$$

$$= [(1.030\ \text{kJ/cyl-cycle})(3000/60\ \text{rev/sec})/(2\ \text{rev/cycle})](4\ \text{cyl})$$

$$\underline{= 103\ \text{kW} = 138\ \text{hp}}$$

Equation (2-2) is used to find mean piston speed:

$$\overline{U}_p = 2SN = (2\ \text{strokes/rev})(0.0942\ \text{m/stroke})(3000/60\ \text{rev/sec})$$

$$\underline{= 9.42\ \text{m/sec}}$$

Equation (2-27) gives net brake work for one cylinder during one cycle:

$$W_b = \eta_m W_i = (0.86)(1.030\ \text{kJ}) \underline{= 0.886\ \text{kJ}}$$

Brake power at 3000 RPM:

$$\dot{W}_b = (3000/60\ \text{rev/sec})(0.5\ \text{cycle/rev})(0.886\ \text{kJ/cyl-cycle})(4\ \text{cyl})$$

$$\underline{= 88.6\ \text{kW} = 119\ \text{hp}}$$

or:

$$\dot{W}_b = \eta_m \dot{W}_i = (0.86)(103\ \text{kW}) = 88.6\ \text{kW}$$

Torque is calculated using Eq. (2-43):

$$\tau = \dot{W}_b/2\pi N = (88.6\ \text{kJ/sec})/(2\pi\ \text{radians/rev})(3000/60\ \text{rev/sec})$$

$$\underline{= 0.282\ \text{kN-m} = 282\ \text{N-m}}$$

Friction power lost using Eq. (2-49):

$$\dot{W}_f = \dot{W}_i - \dot{W}_b = 103 - 88.6 \underline{= 14.4\ \text{kW} = 19.3\ \text{hp}}$$

Equation (2-37c) is used to find brake mean effective pressure:

$$\text{bmep} = \eta_m(\text{imep}) = (0.86)(1649\ \text{kPa}) = \underline{1418\ \text{kPa}}$$

This allows another way of finding torque using Eq. (2-41), which gives consistent results:

$$\tau = (\text{bmep})V_d/4\pi = (1418\ \text{kPa})(0.0025\ \text{m}^3)/4\pi = 0.282\ \text{kN-m}$$

Brake specific power using Eq. (2-51):

$$\text{BSP} = \dot{W}_b/A_p = (88.6\ \text{kW})/\{[(\pi/4)(9.19\ \text{cm})^2](4\ \text{cyl})\} = \underline{0.334\ \text{kW/cm}^2}$$

Output per displacement using Eq. (2-52):

$$\text{OPD} = \dot{W}_b/V_d = (88.6\ \text{kW})/(2.5\ \text{L}) = \underline{35.4\ \text{kW/L}}$$

Equation (2-58) is used to find brake specific fuel consumption:

$$\begin{aligned}\text{bsfc} &= \dot{m}_f/\dot{W}_b \\ &= (0.000044\ \text{kg/cyl-cycle})(50\ \text{rev/sec})(0.5\ \text{cycle/rev})(4\ \text{cyl})/(88.6\ \text{kW}) \\ &= \underline{0.000050\ \text{kg/sec/kW} = 180\ \text{gm/kW-hr}}\end{aligned}$$

Equation (2-69) is used to find volumetric efficiency using one cylinder and standard air density:

$$\begin{aligned}\eta_v &= m_a/\rho_a V_d = (0.000666\ \text{kg})/(1.181\ \text{kg/m}^3)(0.000625\ \text{m}^3) \\ &= \underline{0.902 = 90.2\%}\end{aligned}$$