Internal combustion engines

Chapter three

Engine cycle

By

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Engine cycle

3.1 AIR-STANDARD CYCLES

❖The gases inside combustion chamber is a mixture of air + fuel vapor + exhaust gasses from the previous cycle.
❖The gases inside the combustion chamber is assumed as air to simplify the analysis. hence, these gases are assumed ideal gas through the entire process.
❖ this assumption is valid because:-

- the percentage of exhaust gases in 2nd half of the cycle and fuel vapor in the 1st half of the cycle are relatively small in.
- Closing the cycle simplifies the analysis: both exhaust and intake gases contain large amount of air
- The combustion process is replaced with a heat addition term Q_{in}
- The open exhaust process, which carries a large amount of enthalpy out of the system, is replaced with a closed system heat rejection process Q_{out} of equal energy value.
- Actual engine processes are approximated with ideal processes.

Engine cycle

Assumption for analysis

❖The almost-constant-pressure intake and exhaust strokes are assumed to be constant pressure.
❖Compression strokes and expansion strokes are approximated by isentropic processes (reversible and adiabatic)
❖The combustion process is idealized by a constant-volume process (SI cycle), a constant-pressure process (CI cycle), or a combination of both (CI Dual cycle).
❖Exhaust blow-down is approximated by a constant-volume process.
❖All processes are considered reversible

Engine cycle

According to the previous assumption, the following air-relations can be used

$$Pv = RT \quad \text{(a)}$$
$$PV = mRT \quad \text{(b)}$$
$$P = \rho RT \quad \text{(c)}$$
$$dh = c_p dT \quad \text{(d)}$$
$$du = c_v dT \quad \text{(e)}$$
$$Pv^k = \text{constant} \quad \text{isentropic process} \quad \text{(f)}$$
$$Tv^{k-1} = \text{constant} \quad \text{isentropic process} \quad \text{(g)}$$
$$TP^{(1-k)/k} = \text{constant} \quad \text{isentropic process} \quad \text{(h)}$$
$$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k) \quad \text{isentropic work in closed system} \quad \text{(i)}$$
$$= R(T_2 - T_1)/(1-k)$$
$$c = \sqrt{kRT} \quad \text{speed of sound} \quad \text{(j)}$$

(3-1)

Engine cycle

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:

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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

Engine cycle

Standard values of air properties

$$c_p = 1.108 \text{ kJ/kg-K} = 0.265 \text{ BTU/lbm-°R}$$
$$c_v = 0.821 \text{ kJ/kg-K} = 0.196 \text{ BTU/lbm-°R}$$
$$k = c_P/c_v = 1.108/0.821 = 1.35$$
$$R = c_P - c_v = 0.287 \text{ kJ/kg-K}$$
$$= 0.069 \text{ BTU/lbm-°R} = 53.33 \text{ ft-lbf/lbm-°R}$$

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:

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

Engine cycle

3-2 OTTO CYCLE

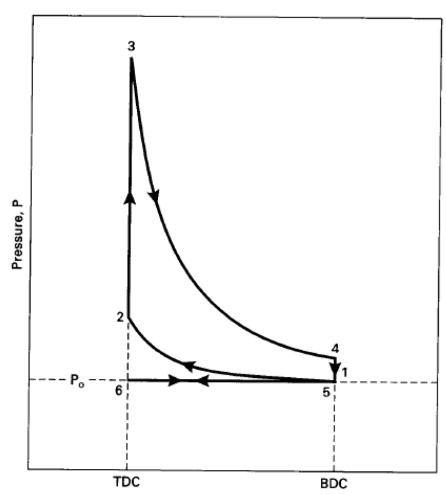
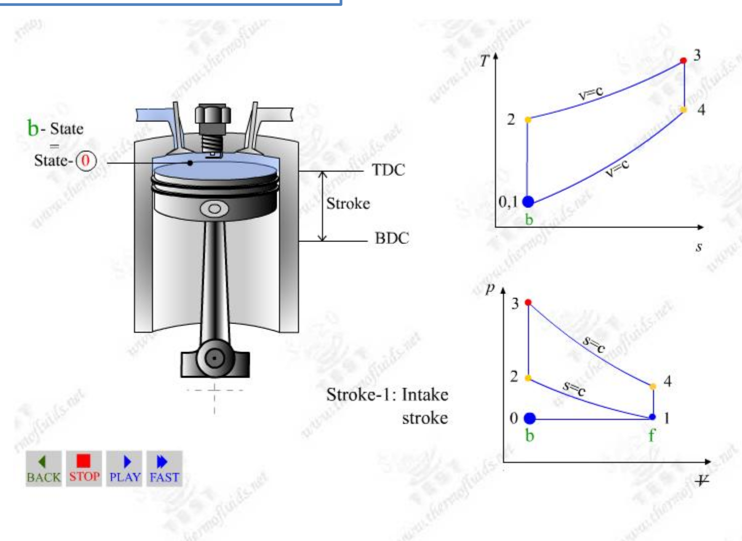


Figure 3-1 Ideal air-standard Otto cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an SI engine on P-V coordinates.

Engine cycle

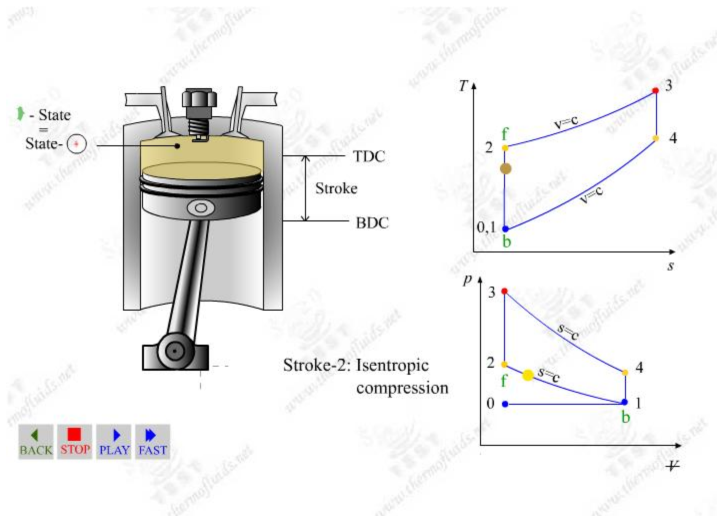
Engine cycle

The intake stroke of the Otto cycle



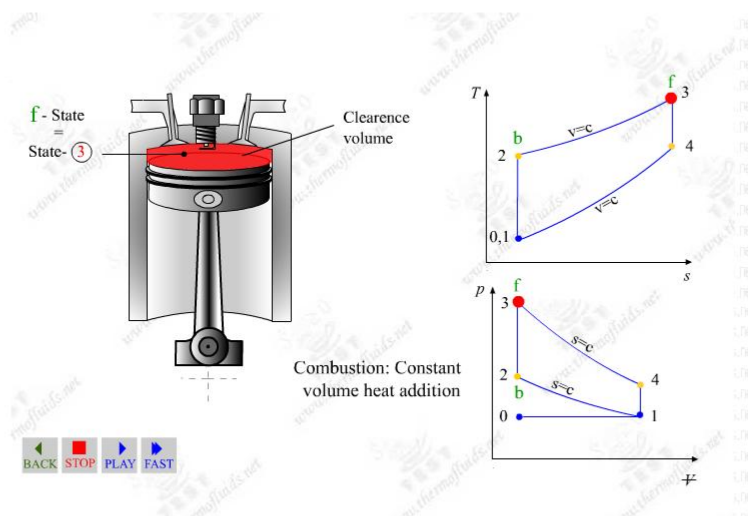
Engine cycle

the compression stroke



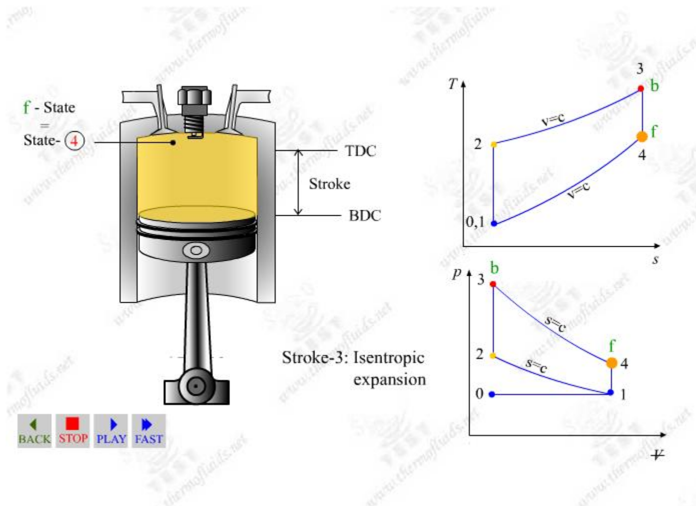
Engine cycle

The combustion process



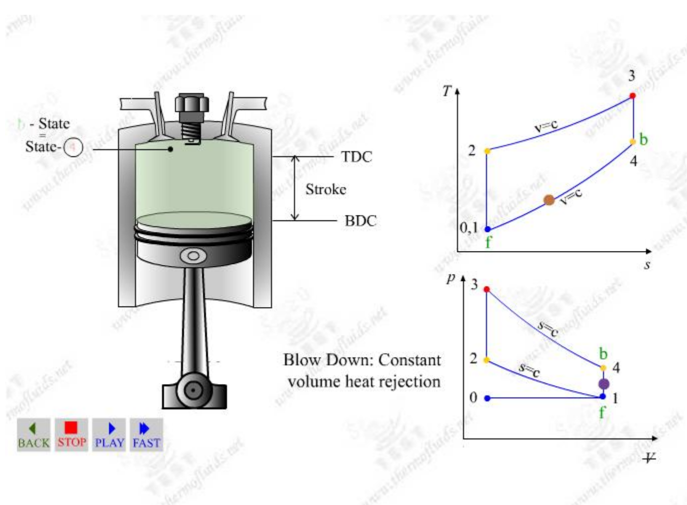
Engine cycle

the power stroke



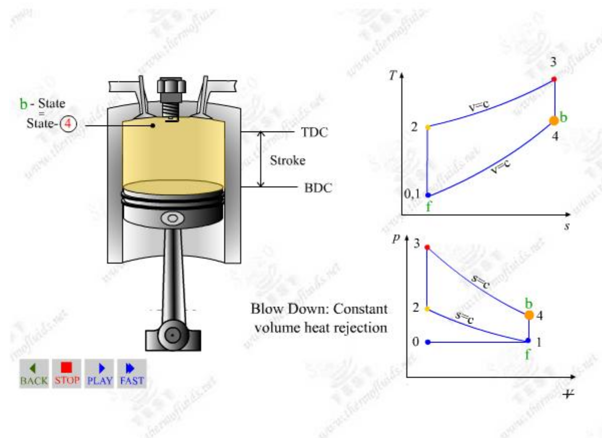
Engine cycle

Blow – down process



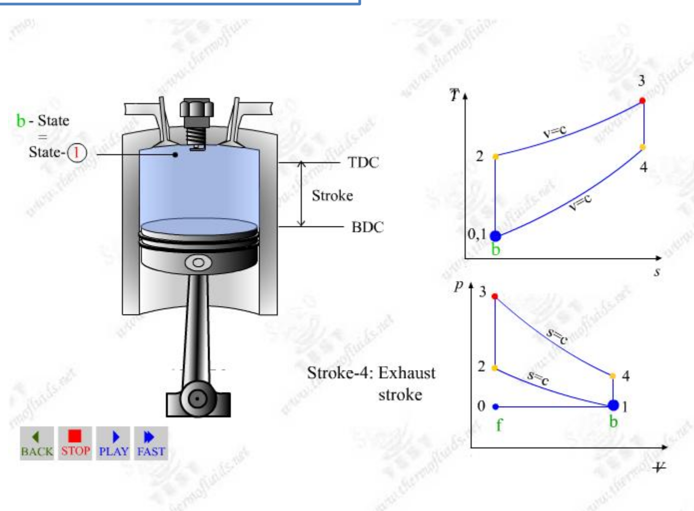
Engine cycle

closed-system process 4-5. Enthalpy loss during this process is replaced with heat rejection in the engine analysis. Pressure within the cylinder at the end of exhaust blowdown has been reduced to about one atmosphere, and the temperature has been substantially reduced by expansion cooling.



Engine cycle

the exhaust stroke



Engine cycle

Thermodynamic Analysis of Air-Standard Otto Cycle

Process 6-1—constant-pressure intake of air at P_o.
Intake valve open and exhaust valve closed:

$$P_1 = P_6 = P_o \quad (3\text{-}2)$$

$$w_{6-1} = P_o(v_1 - v_6) \quad (3\text{-}3)$$

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

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

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

$$q_{1-2} = 0 \quad (3\text{-}6)$$

Engine cycle

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

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

$$v_3 = v_2 = v_{\text{TDC}} \quad (3\text{-}8)$$

$$w_{2-3} = 0 \quad (3\text{-}9)$$

$$Q_{2-3} = Q_{\text{in}} = m_f Q_{\text{HV}} \eta_c = m_m c_v(T_3 - T_2) \quad (3\text{-}10)$$
$$= (m_a + m_f)c_v(T_3 - T_2)$$

$$Q_{\text{HV}} \eta_c = (\text{AF} + 1)c_v(T_3 - T_2) \quad (3\text{-}11)$$

$$q_{2-3} = q_{\text{in}} = c_v(T_3 - T_2) = (u_3 - u_2) \quad (3\text{-}12)$$

$$T_3 = T_{\max} \quad (3\text{-}13)$$

$$P_3 = P_{\max} \quad (3\text{-}14)$$

Engine cycle

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

$$q_{3-4} = 0 \quad (3\text{-}15)$$

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

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

$$w_{3-4} = (P_4v_4 - P_3v_3)/(1-k) = R(T_4 - T_3)/(1-k) \quad (3\text{-}18)$$
$$= (u_3 - u_4) = c_v(T_3 - T_4)$$

Process 4-5—constant-volume heat rejection (exhaust blowdown).
Exhaust valve open and intake valve closed:

$$v_5 = v_4 = v_1 = v_{BDC} \quad (3\text{-}19)$$

$$w_{4-5} = 0 \quad (3\text{-}20)$$

$$Q_{4-5} = Q_{out} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4) \quad (3\text{-}21)$$

$$q_{4-5} = q_{out} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4) \quad (3\text{-}22)$$

Engine cycle

Process 5-6—constant-pressure exhaust stroke at P_o.
Exhaust valve open and intake valve closed:

$$P_5 = P_6 = P_o \quad (3\text{-}23)$$

$$w_{5-6} = P_o(v_6 - v_5) = P_o(v_6 - v_1) \quad (3\text{-}24)$$

Thermal efficiency of Otto cycle:

$$(\eta_t)_{OTTO} = |w_{net}|/|q_{in}| = 1 - (|q_{out}|/|q_{in}|) \quad (3\text{-}25)$$
$$= 1 - [c_v(T_4 - T_1)/c_v(T_3 - T_2)]$$
$$= 1 - [(T_4 - T_1)/(T_3 - T_2)]$$

Only cycle temperatures need to be known to determine thermal efficiency. This can be simplified further by applying ideal gas relationships for the isentropic compression and expansion strokes and recognizing that $v_1 = v_4$ and $v_2 = v_3$:

$$(T_2/T_1) = (v_1/v_2)^{k-1} = (v_4/v_3)^{k-1} = (T_3/T_4) \quad (3\text{-}26)$$

Rearranging the temperature terms gives:

$$T_4/T_1 = T_3/T_2 \quad (3\text{-}27)$$

Equation (3-25) can be rearranged to:

$$(\eta_t)_{OTTO} = 1 - (T_1/T_2)\{[(T_4/T_1) - 1]/[(T_3/T_2) - 1]\} \quad (3\text{-}28)$$

Using Eq. (3-27) gives:

$$(\eta_t)_{OTTO} = 1 - (T_1/T_2) \quad (3\text{-}29)$$

Combining this with Eq. (3-4):

Engine cycle

$$(\eta_t)_{OTTO} = 1 - [1/(v_1/v_2)^{k-1}] \qquad (3\text{-}30)$$

With $v_1/v_2 = r_c$, the compression ratio:

$$(\eta_t)_{OTTO} = 1 - (1/r_c)^{k-1} \qquad (3\text{-}31)$$

Only the compression ratio is needed to determine the thermal efficiency of the Otto cycle at WOT. As the compression ratio goes up, the thermal efficiency goes up as seen in Fig. 3-3. This efficiency is the **indicated thermal efficiency**, as the heat transfer values are those to and from the air within the combustion chamber.

Engine cycle

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:

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

Engine cycle

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.

$$m_m = P_1V_1/RT_1 = (100\ \text{kPa})(0.000707\ \text{m}^3)/(0.287\ \text{kJ/kg-K})(333\ \text{K})$$
$$= 0.000740\ \text{kg}$$

Engine cycle

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

Engine cycle

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

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

$$= (0.000044 \text{ kg})(44{,}300 \text{ kJ/kg})(1.00)$$
$$= (0.000740 \text{ kg})(0.821 \text{ kJ/kg-K})(T_3 - 707 \text{ K})$$

Engine cycle

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:

$$T_4 = T_3(1/r_c)^{k-1} = (3915 \text{ K})(1/8.6)^{0.35} \underline{= 1844 \text{ K} = 1571°\text{C}}$$
$$P_4 = P_3(1/r_c)^k = (10{,}111 \text{ kPa})(1/8.6)^{1.35} \underline{= 554 \text{ kPa}}$$
$$V_4 = mRT_4/P_4 = (0.000740 \text{ kg})(0.287 \text{ kJ/kg-K})(1844 \text{ K})/(554 \text{ kPa})$$
$$\underline{= 0.000707 \text{ m}^3 = V_1}$$

Engine cycle

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.

Engine cycle

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):

$$\eta_t = 1 - (T_1/T_2) = 1 - (1/r_c)^{k-1}$$
$$= 1 - (333/707) = 1 - (1/8.6)^{0.35} = 0.529$$

Engine cycle

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:

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

Engine cycle

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):

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

Engine cycle

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:

Engine cycle

$$\text{bsfc} = \dot{m}_f/\dot{W}_b$$

$$= (0.000044\ \text{kg/cyl-cycle})(50\ \text{rev/sec})(0.5\ \text{cycle/rev})(4\ \text{cyl})/(88.6\ \text{kW})$$

$$= \underline{0.000050\ \text{kg/sec/kW} = 180\ \text{gm/kW-hr}}$$

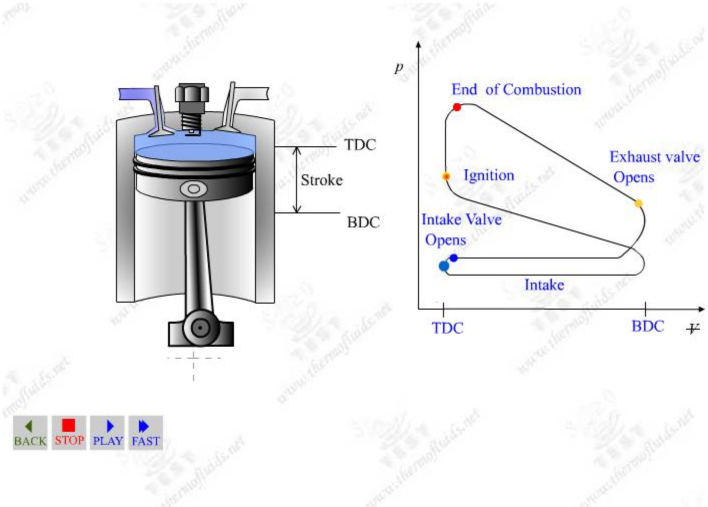
Equation (2-69) is used to find volumetric efficiency using one cylinder and standard air density:

$$\eta_v = m_a/\rho_a V_d = (0.000666\ \text{kg})/(1.181\ \text{kg/m}^3)(0.000625\ \text{m}^3)$$

$$= \underline{0.902 = 90.2\%}$$

Engine cycle

3-3 REAL AIR-FUEL ENGINE CYCLES



Engine cycle

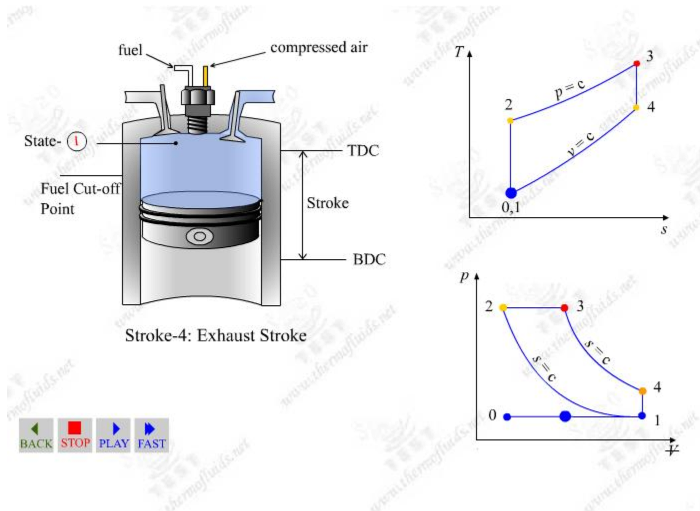
In real cycle

1. Real engines operate on an open cycle with changing composition
2. In a real engine inlet flow may be all air, or it may be air mixed with up to 7% fuel, either gaseous or as liquid droplets, or both
3. There are heat losses during the cycle of a real engine which are neglected in air-standard analysis
4. Combustion requires a short but finite time to occur, and heat addition is not instantaneous at TDC, as approximated in an Otto cycle
5. The blow-down process requires a finite real time and a finite cycle time, and does not occur at constant volume as in air-standard analysis
6. In an actual engine, the intake valve is not closed until after bottom-dead-center at the end of the intake stroke
7. Engine valves require a finite time to actuate

$(\eta t)_{actual} = 0.85\ (\eta t)_{OTTO}$

Engine cycle

3-6 DIESEL CYCLE



Engine cycle

3-6 DIESEL CYCLE

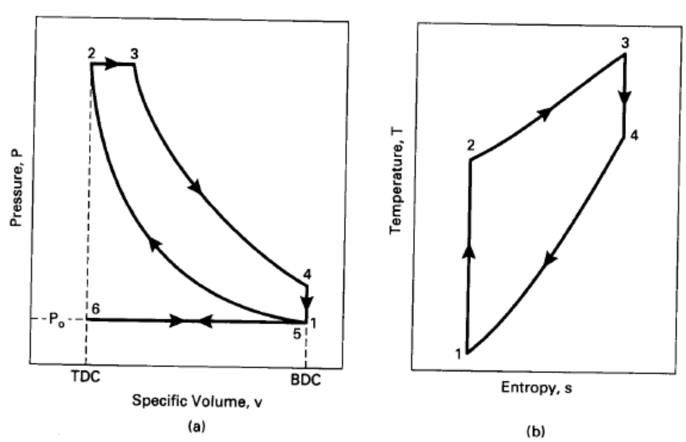


Figure 3-8 Air-standard diesel cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an early CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

Engine cycle

3-6 DIESEL CYCLE

Thermodynamic Analysis of Air-Standard Diesel Cycle

Process 6-1—constant-pressure intake of air at P_o.
Intake valve open and exhaust valve closed:

$$w_{6-1} = P_o(v_1 - v_6) \tag{3-51}$$

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

$$T_2 = T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \tag{3-52}$$

$$P_2 = P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k \tag{3-53}$$

$$V_2 = V_{\text{TDC}} \tag{3-54}$$

$$q_{1-2} = 0 \tag{3-55}$$

$$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k) = R(T_2 - T_1)/(1-k) \tag{3-56}$$

$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

Engine cycle

3-6 DIESEL CYCLE

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

$$Q_{2-3} = Q_{\text{in}} = m_f Q_{\text{HV}} \eta_c = m_m c_p(T_3 - T_2) = (m_a + m_f)c_p(T_3 - T_2) \tag{3-57}$$

$$Q_{\text{HV}} \eta_c = (\text{AF} + 1)c_p(T_3 - T_2) \tag{3-58}$$

$$q_{2-3} = q_{\text{in}} = c_p(T_3 - T_2) = (h_3 - h_2) \tag{3-59}$$

$$w_{2-3} = q_{2-3} - (u_3 - u_2) = P_2(v_3 - v_2) \tag{3-60}$$

$$T_3 = T_{\text{max}} \tag{3-61}$$

Cutoff ratio is defined as the change in volume that occurs during combustion, given as a ratio:

$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2 \tag{3-62}$$

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

Engine cycle

3-6 DIESEL CYCLE

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

$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_c = m_m c_p (T_3 - T_2) = (m_a + m_f) c_p (T_3 - T_2) \quad (3\text{-}57)$$

$$Q_{HV} \eta_c = (AF + 1) c_p (T_3 - T_2) \quad (3\text{-}58)$$

$$q_{2-3} = q_{in} = c_p (T_3 - T_2) = (h_3 - h_2) \quad (3\text{-}59)$$

$$w_{2-3} = q_{2-3} - (u_3 - u_2) = P_2 (v_3 - v_2) \quad (3\text{-}60)$$

$$T_3 = T_{max} \quad (3\text{-}61)$$

Cutoff ratio is defined as the change in volume that occurs during combustion, given as a ratio:

$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2 \quad (3\text{-}62)$$

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

Engine cycle

3-6 DIESEL CYCLE

$$q_{3-4} = 0 \quad (3\text{-}63)$$

$$T_4 = T_3 (v_3/v_4)^{k-1} = T_3 (V_3/V_4)^{k-1} \quad (3\text{-}64)$$

$$P_4 = P_3 (v_3/v_4)^{k} = P_3 (V_3/V_4)^{k} \quad (3\text{-}65)$$

$$w_{3-4} = (P_4 v_4 - P_3 v_3)/(1 - k) = R(T_4 - T_3)/(1 - k) \quad (3\text{-}66)$$
$$= (u_3 - u_4) = c_v (T_3 - T_4)$$

Process 4-5—constant-volume heat rejection (exhaust blowdown).
Exhaust valve open and intake valve closed:

$$v_5 = v_4 = v_1 = v_{BDC} \quad (3\text{-}67)$$

$$w_{4-5} = 0 \quad (3\text{-}68)$$

Engine cycle

3-6 DIESEL CYCLE

$$Q_{4-5} = Q_{out} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4) \quad (3\text{-}69)$$

$$q_{4-5} = q_{out} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4) \quad (3\text{-}70)$$

Process 5-6—constant-pressure exhaust stroke at P_o.
Exhaust valve open and intake valve closed:

$$w_{5-6} = P_o(v_6 - v_5) = P_o(v_6 - v_1) \quad (3\text{-}71)$$

Thermal efficiency of diesel cycle:

$$(\eta_t)_{DIESEL} = |w_{net}|/|q_{in}| = 1 - (|q_{out}|/|q_{in}|) \quad (3\text{-}72)$$

$$= 1 - [c_v(T_4 - T_1)/c_p(T_3 - T_2)]$$

$$= 1 - (T_4 - T_1)/[k(T_3 - T_2)]$$

With rearrangement, this can be shown to equal:

$$(\eta_t)_{DIESEL} = 1 - (1/r_c)^{k-1}[(\beta^k - 1)/\{k(\beta - 1)\}] \quad (3\text{-}73)$$

where: r_c = compression ratio
$k = c_p/c_v$
β = cutoff ratio

Engine cycle

EXAMPLE PROBLEM 3-4

A small truck has a four-cylinder, four-liter CI engine that operates on the air-standard Dual cycle (Fig. 3-10) using light diesel fuel at an air-fuel ratio of 18. The compression ratio of the engine is 16:1 and the cylinder bore diameter is 10.0 cm. At the start of the compression stroke, conditions in the cylinders are 60°C and 100 KPa with a 2% exhaust residual. It can be assumed that half of the heat input from combustion is added at constant volume and half at constant pressure. Calculate:

1. **temperature and pressure at each state of the cycle**
2. indicated thermal efficiency
3. exhaust temperature
4. air temperature in intake manifold
5. engine volumetric efficiency

For one cylinder:

$$V_d = (4\ \text{L})/4 = 1\ \text{L} = 0.001\ \text{m}^3 = 1000\ \text{cm}^3$$

Engine cycle

Using Eq. (2-12):

$$r_c = V_{BDC}/V_{TDC} = (V_d + V_c)/V_c = 16 = (1000 + V_c)/V_c$$

$$V_c = 66.7 \text{ cm}^3 = 0.0667 \text{ L} = 0.0000667 \text{ m}^3$$

Using Eq. (2-8):

$$V_d = (\pi/4)B^2S = 0.001 \text{ m}^3 = (\pi/4)(0.10 \text{ m})^2 S$$

$$S = 0.127 \text{ m} = 12.7 \text{ cm}$$

State 1:

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

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

$V_1 = V_{BDC} = V_d + V_c = 0.001 + 0.0000667 \underline{= 0.0010667 \text{ m}^3}$

Mass of gas in one cylinder at start of compression:

$$m_m = P_1V_1/RT_1 = (100 \text{ kPa})(0.0010667 \text{ m}^3)/(0.287 \text{ kJ/kg-K})(333 \text{ K})$$
$$= 0.00112 \text{ kg}$$

Mass of fuel injected per cylinder per cycle:

$$m_f = (0.00112)(0.98)(1/19) = 0.0000578 \text{ kg}$$

Engine cycle

State 2: Equations (3-52) and (3-53) give temperature and pressure after compression:

$$T_2 = T_1(r_c)^{k-1} = (333 \text{ K})(16)^{0.35} \underline{= 879 \text{ K} = 606°\text{C}}$$

$$P_2 = P_1(r_c)^k = (100 \text{ kPa})(16)^{1.35} \underline{= 4222 \text{ kPa}}$$

$$V_2 = mRT_2/P_2 = (0.00112 \text{ kg})(0.287 \text{ kJ/kg-K})(879 \text{ K})/(4222 \text{ kPa})$$
$$\underline{= 0.000067 \text{ m}^3 = V_c}$$

or using Eq. (2-12):

$$V_2 = V_1/r_c = (0.0010667)/(16) = 0.0000667 \text{ m}^3$$

Engine cycle

State x: Heating value of light diesel fuel is obtained from Table A-2 in the Appendix:

$$Q_{in} = m_f Q_{HV} = (0.0000578 \text{ kg})(42{,}500 \text{ kJ/kg}) = 2.46 \text{ kJ}$$

If half of Q_{in} occurs at constant volume, then using Eq. (3-76):

$$Q_{2-x} = 1.23 \text{ kJ} = m_m c_v (T_x - T_2)$$

$$= (0.00112 \text{ kg})(0.821 \text{ kJ/kg-K})(T_x - 879 \text{ K})$$

$$\underline{T_x = 2217 \text{ K} = 1944°\text{C}}$$

$$\underline{V_x = V_2 = 0.0000667 \text{ m}^3}$$

$$P_x = mRT_x/V_x$$

$$= (0.00112 \text{ kg})(0.287 \text{ kJ/kg-K})(2217 \text{ K})/(0.0000667 \text{ m}^3)$$

$$\underline{= 10{,}650 \text{ kPa} = P_{max}}$$

or:

$$P_x = P_2(T_x/T_2) = (4222 \text{ kPa})(2217/879) = 10{,}650 \text{ kPa}$$

Engine cycle

State 3:

$$\underline{P_3 = P_x = 10{,}650 \text{ kPa} = P_{max}}$$

Equation (3-81) gives:

$$Q_{x-3} = 1.23 \text{ kJ} = m_m c_p (T_3 - T_x)$$

$$= (0.00112 \text{ kg})(1.108 \text{ kJ/kg-K})(T_3 - 2217 \text{ K})$$

$$\underline{T_3 = 3208 \text{ K} = 2935°\text{C} = T_{max}}$$

$$V_3 = mRT_3/P_3 = (0.00112 \text{ kg})(0.287 \text{ kJ/kg-K})(3208 \text{ K})/(10{,}650 \text{ kPa})$$

$$\underline{= 0.000097 \text{ m}^3}$$

Engine cycle

State 4:

$$V_4 = V_1 = 0.0010667 \text{ m}^3$$

Equations (3-64) and (3-65) give temperature and pressure after expansion:

$$T_4 = T_3(V_3/V_4)^{k-1} = (3208 \text{ K})(0.000097/0.0010667)^{0.35}$$
$$= 1386 \text{ K} = 1113°\text{C}$$

$$P_4 = P_3(V_3/V_4)^k = (10{,}650 \text{ kPa})(0.000097/0.0010667)^{1.35} = 418 \text{ kPa}$$

Work out for process x-3 for one cylinder for one cycle using Eq. (3-83):

$$W_{x-3} = P(V_3 - V_x) = (10{,}650 \text{ kPa})(0.000097 - 0.0000667)\text{m}^3 = 0.323 \text{ kJ}$$

Work out for process 3-4 using Eq. (3-66):

$$W_{3-4} = mR(T_4 - T_3)/(1 - k)$$
$$= (0.00112 \text{ kg})(0.287 \text{ kJ/kg-K})(1386 - 3208)\text{K}/(1 - 1.35)$$
$$= 1.673 \text{ kJ}$$

Work in for process 1-2 using Eq. (3-56):

$$W_{1-2} = mR(T_2 - T_1)/(1 - k)$$
$$= (0.00112 \text{ kg})(0.287 \text{ kJ/kg-K})(879 - 333)\text{K}/(1 - 1.35)$$
$$= -0.501 \text{ kJ}$$

$$W_{\text{net}} = (+0.323) + (+1.673) + (-0.501) = +1.495 \text{ kJ}$$

Engine cycle

2) Equation (3-88) gives indicated thermal efficiency:

$$(\eta_t)_{\text{DUAL}} = |W_{\text{net}}|/|Q_{\text{in}}| = (1.495 \text{ kJ})/(2.46 \text{ kJ}) = 0.607 = 60.7\%$$

Pressure ratio:

$$\alpha = P_x/P_2 = 10{,}650/4222 = 2.52$$

Cutoff ratio:

$$\beta = V_3/V_x = 0.000097/0.0000667 = 1.45$$

Using Eq. (3-89) to find thermal efficiency:

$$(\eta_t)_{\text{DUAL}} = 1 - (1/r_c)^{k-1}[\{\alpha\beta^k - 1\}/\{k\alpha(\beta - 1) + \alpha - 1\}]$$
$$= 1 - (1/16)^{0.35}[\{(2.52)(1.45)^{1.35} - 1\}/\{(1.35)(2.52)(1.45 - 1) + 2.52 - 1\}]$$
$$= 0.607$$

3) Assuming exhaust pressure is the same as intake pressure and using Eq. (3-37) for exhaust temperature:

$$T_{\text{ex}} = T_4(P_{\text{ex}}/P_4)^{(k-1)/k} = (1386 \text{ K})(100/418)^{(1.35-1)/1.35} = 957 \text{ K} = 684°\text{C}$$

Engine cycle

Equation (3-46) gives exhaust residual:

$$x_r = (1/r_c)(T_4/T_{ex})(P_{ex}/P_4)$$

$$= (1/16)(1386/957)(100/418) = \underline{0.022 = 2.2\%}$$

4) Using Eq. (3-50) to find air temperature entering the cylinder:

$$(T_m)_1 = x_r T_{ex} + (1 - x_r) T_a$$

$$(333\text{ K}) = (0.022)(957\text{ K}) + (1 - 0.022) T_a$$

$$\underline{T_a = 319\text{ K} = 46°\text{C}}$$

5) Mass of air entering one cylinder during intake:

$$m_a = (0.00112\text{ kg})(0.98) = 0.00110\text{ kg}$$

Volumetric efficiency is found using Eq. (2-69):

$$\eta_v = m_a/\rho_a V_d = (0.00110\text{ kg})/(1.181\text{ kg/m}^3)(0.001\text{ m}^3)$$

$$= \underline{0.931 = 93.1\%}$$

Engine cycle

3-8 COMPARISON OF OTTO, DIESEL, AND DUAL CYCLES

Figure 3-11 compares Otto, Diesel, and Dual cycles with the same inlet conditions and the same compression ratios.

***(ηt)*OTTO > *(ηt)*DUAL > *(ηt)*DIESEL**

in Fig. 3-12. A more realistic way to compare these three cycles would be to have the same peak pressure-