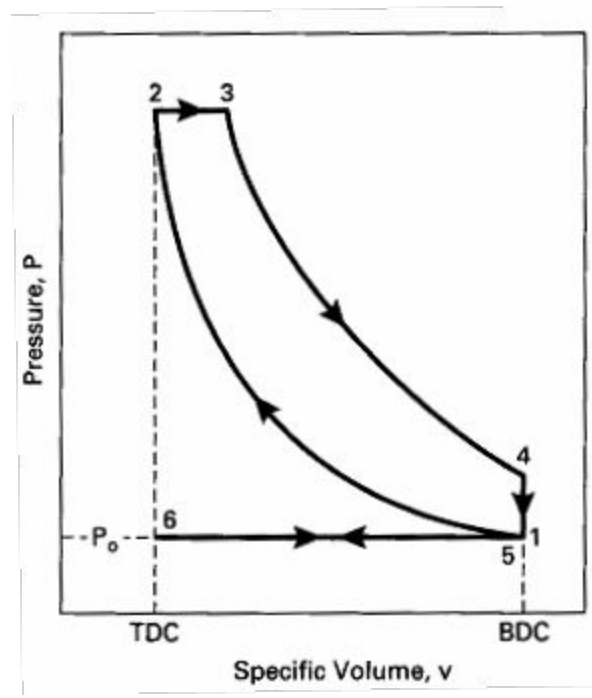
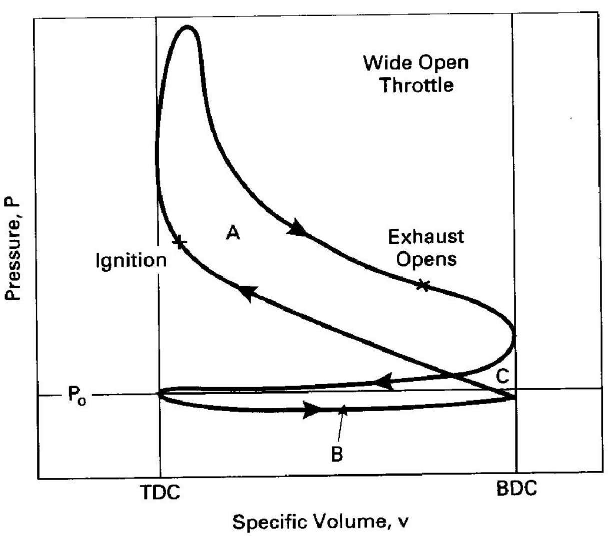
Internal Combustion Engine

Thermodynamic analysis

Engine Cycles





Aleksey Terentyev

Basic terms and definitions

- An **ideal gas** is a theoretical is a theoretical gas is a theoretical gas composed of a set of randomly moving, non-interacting point particles is a theoretical gas composed of a set of randomly moving, non-interacting point particles. The ideal gas concept is useful because it obeys the ideal gas law is a theoretical gas composed of a set of randomly moving, non-interacting point particles. The ideal gas concept is useful because it obeys the ideal gas law, a simplified equation of state is a theoretical gas composed of a set of randomly moving, non-interacting point particles. The ideal gas concept is useful because it obeys the ideal gas law, a simplified equation of state, and is amenable to analysis under statistical mechanics.
- **Enthalpy** is a measure of the total energy is a measure of the total energy of a thermodynamic system is a measure of the total energy of a thermodynamic system. It includes the internal energy is a measure of the total energy of a thermodynamic system. It includes the internal energy, which is the energy required to create a system, and the amount of energy required to make room for it by displacing its environment is a measure of the total energy of a thermodynamic system. It includes the internal energy, which is the energy required to create a system, and the amount of energy required to make room for it by displacing its environment and establishing its volume is a measure of the total energy of a thermodynamic system. It includes the internal energy, which is the energy required to create a system, and the amount of energy required to make room for it by displacing its environment and establishing its volume and pressure.
- An **isentropic process** or **isoentropic process** is one in which, for purposes of engineering analysis and calculation, one may assume that the process takes place from initiation to completion without an increase or decrease in the entropy of the system, i.e., the entropy of the system remains constant. It can be proven that any reversible is one in which, for purposes of engineering analysis and

Basic terms and definitions

- specific volume of gas - удельный объем газа;
- specific enthalpy - удельная энтальпия;
- specific internal energy - удельная внутренняя энергия;
  specific heats – теплоемкости;
- specific work – удельная работа;
- mass flow rate - едельный массовый расход;
- heat transfer rate for unit mass - теплопроизводительность для единицы массы;
- heat transfer rate - скорость теплопередачи;
- WOT (Wide-Open Throttle) – полностью открытый дроссель;
- **Bottom-Dead-Center** (BDC);
- **Top-Dead-Center** (TDC);
- When an occurrence in a cycle happens before TDC, it is often abbreviated **bTDC or bTe;**
- When the occurrence happens after TDC, it will be abbreviated **aTDC or aTe;**
- During an engine cycle things can happen before bottom-dead-center, **bBDC or bBC**, and after bottom-dead-center, **aBDC or aBe**;
- crevice flow – щелевой поток;
- blowby – прорыв газов;

ENGINE CYCLES

This chapter studies the basic cycles used in reciprocating internal combustion engines, both four stroke and two stroke.

The most common four-stroke SI and CI cycles are analyzed in detail using air-standard analysis.

WORK

Work is the output of any heat engine, and in a reciprocating IC engine this work is generated by the gases in the combustion chamber of the cylinder. Work is the result of a force acting through a distance. Force due to gas pressure on the moving piston generates the work in an IC engine cycle.

$$W = \int F\,dx = \int P A_p\,dx \tag{2-18}$$

where: P = pressure in combustion chamber
A_P = area against which the pressure acts (i.e., the piston face)
x = distance the piston moves

and:

$$A_p\,dx = dV \tag{2-19}$$

dV is the differential volume displaced by the piston, so work done can be written:

$$W = \int P\,dV \tag{2-20}$$

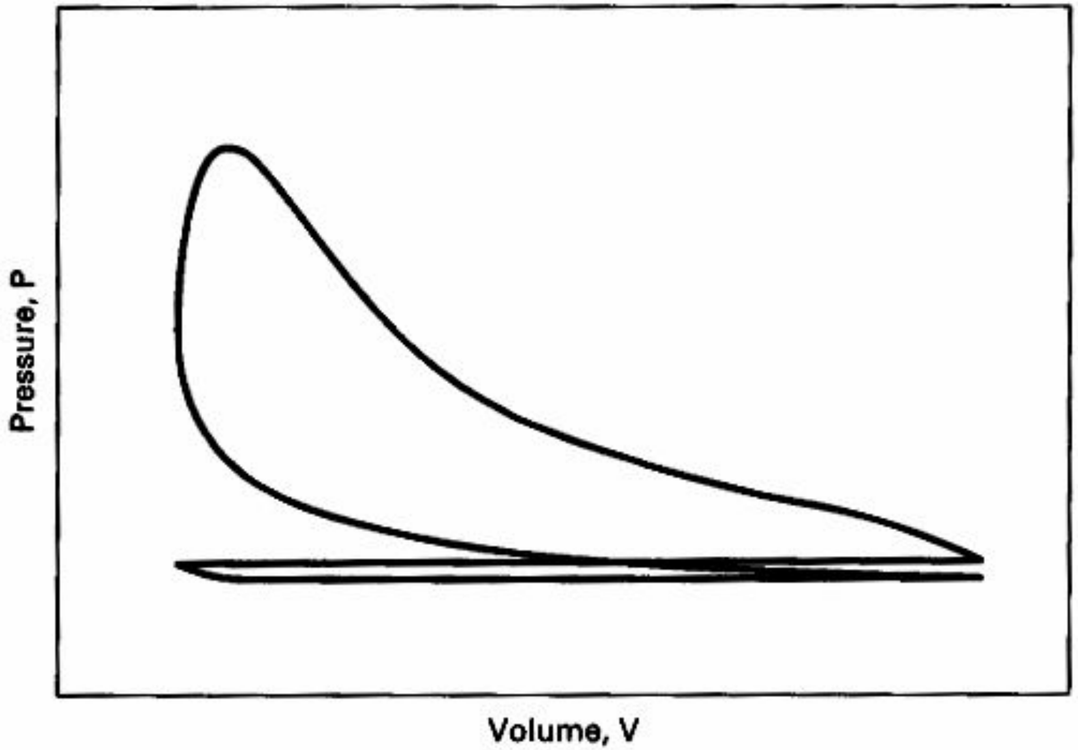


Figure 1 Indicator diagram for a typical four stroke cycle SI engine. An indicator diagram plots cylinder pressure as a function of combustion chamber volume over a 720° cycle

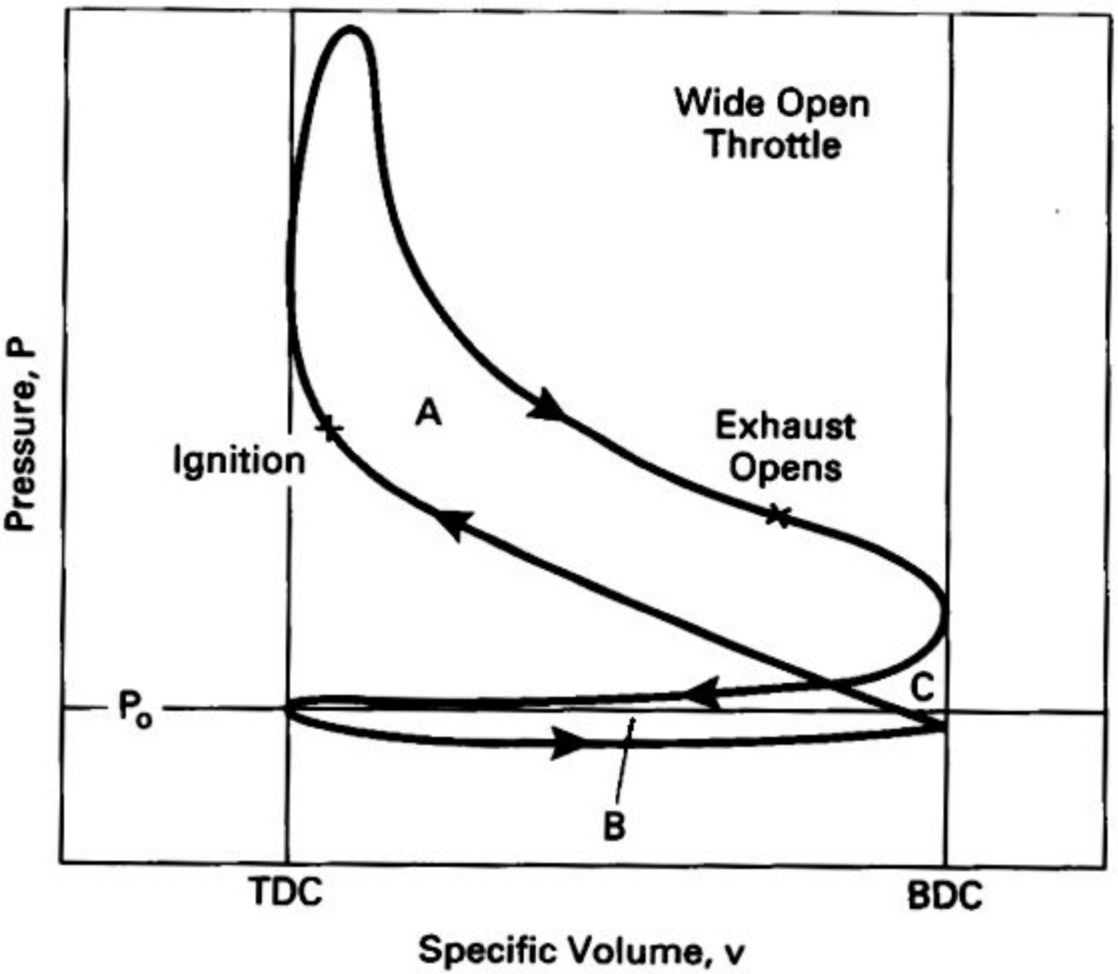
Figure 1 , which plots the engine cycle on P–V coordinates, is often called an indicator diagram. Early indicator diagrams were generated by mechanical plotters linked directly to the engine. Modern P–V indicator diagrams are generated on an oscilloscope using a pressure transducer mounted in the combustion chamber and an electronic position sensor mounted on the piston or crankshaft.

Because engines are often multicylinder, it is convenient to analyze engine cycles per unit mass of gas m within the cylinder. To do so, volume V is replaced with specific volume v and work is replaced with specific work:

$$w = W/m \qquad v = V/m \tag{2-21}$$

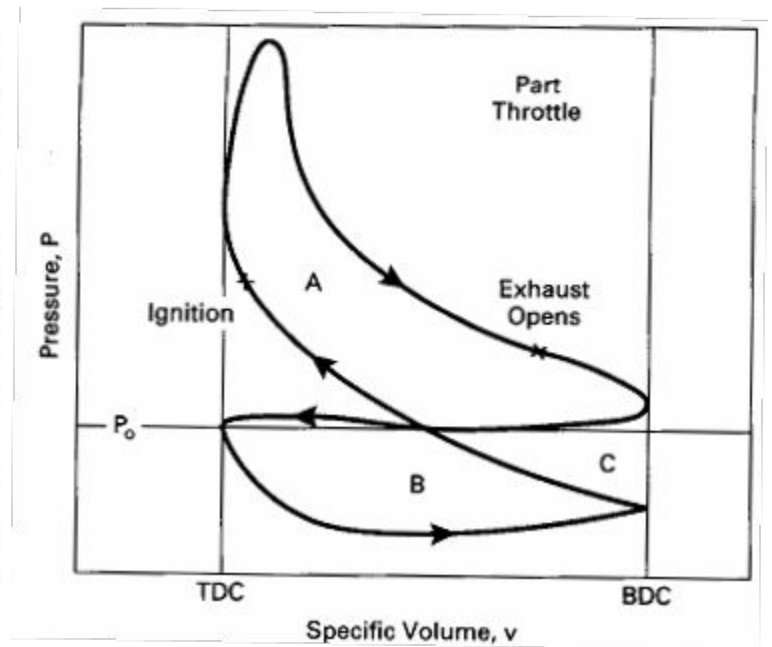
$$w = \int P\,dv \tag{2-22}$$

Specific work w is equal to the area under the process lines on the P–v coordinates of Fig. 2 .



(a)

Figure 2 Four-stroke cycle of typical SI engine plotted on P-v coordinates at (a) wide open throttle, and (b) part throttle. The upper loop consists of the compression stroke and power stroke and the area represents gross indicated work. The lower loop represents negative work of the intake stroke and exhaust stroke. This is called indicated pump work.



(b)

If P represents the pressure inside the cylinder combustion chamber, then equation (2-22) and the areas shown in Fig. 2 give the work inside the combustion chamber. This is called **indicated work.** Work delivered by the crankshaft is less than indicated work due to mechanical friction and parasitic loads of the engine. Parasitic loads include the oil pump, supercharger, air conditioner compressor, alternator, etc. Actual work available at the crankshaft is called **brake work** ω_b. Units of specific work will be kJ/kg:

$$\omega_b = \omega_i - \omega_t \qquad (2\text{-}23)$$

where: ω_i = indicated specific work generated inside combustion chamber;
ω_t = specific work lost due to friction and parasitic loads.

The upper loop of the engine cycle in Fig. 2 consists of the compression and power strokes where output work is generated and is called the gross **indicated work** (areas A and C in Fig. 2). The lower loop, which includes the intake and exhaust stroke, is called **pump work** and absorbs work from the engine (areas B and C).

Net indicated work is:

$$w_{\text{net}} = w_{\text{gross}} + w_{\text{pump}} \qquad (2\text{-}24)$$

Pump work w_{pump} is negative for engines without superchargers:

$$w_{\text{net}} = (\text{Area A}) - (\text{Area B}) \qquad (2\text{-}25)$$

Engines with superchargers or turbochargers can have intake pressure greater than exhaust pressure, giving a positive pump work (Fig. 3). When this occurs:

$$w_{\text{net}} = (\text{Area A}) + (\text{Area B}) \qquad (2\text{-}26)$$

Superchargers increase net indicated work but add to the friction work of the engine since they are driven by the crankshaft.

The ratio of brake work at the crankshaft to indicated work in the combustion chamber defines the **mechanical efficiency** of an engine:

$$\eta_m = w_b / w_i = W_b / W_i \qquad (2\text{-}27)$$

Mechanical efficiency will be on the order of 75% to 95%, at high speed for modern automobile engines operating at wide-open throttle.

It then decreases with decreasing engine speed to zero at idle conditions, when no work is taken off the crankshaft.

Care should be taken when using the terms "gross work" and "net work". **In** some older literature and textbooks, net work (or net power) meant the output of an engine with all components, while gross work (or gross power) meant the output of the engine with fan and exhaust system removed.

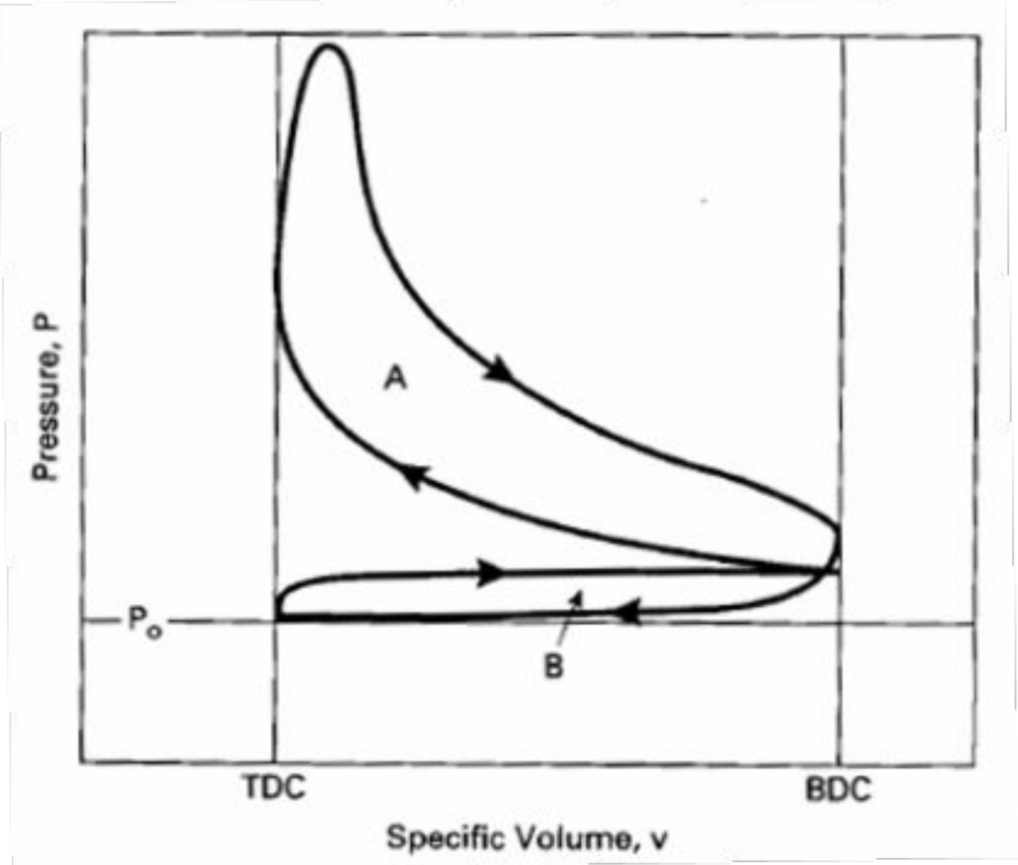


Figure 3 Four-stroke cycle of a SI engine equipped with a supercharger or turbocharger, plotted on P-v coordinates. For this cycle intake pressure is greater than exhaust pressure and the pump work loop represents positive work.

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 residue remaining from the previous cycle. This mixture is then compressed and combusted, changing the composition to exhaust products consisting largely of CO_2, H_20, 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 the following:

AIR-STANDARD CYCLES

1. The gas mixture in the cylinder is treated as air for the entire cycle, and property values of air are used in the analysis. This is a good approximation during the first half of the cycle, when most of the gas in the cylinder is air with only up to about 7% fuel vapor. Even in the second half of the cycle, when the gas compo-sition is mostly CO_2, H_20, and N_2, using air properties does not create large errors in the analysis. Air will be treated as an ideal gas with constant specific heats.

2. The real open cycle is changed into a closed cycle by assuming that the gases being exhausted are fed back into the intake system. This works with ideal air-standard cycles, as both intake gases and exhaust gases are air. Closing the cycle simplifies the analysis.

3. The combustion process is replaced with a heat addition term Q_{in} of equal energy value. Air alone cannot combust.

4. 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.

5. Actual engine processes are approximated with ideal processes.

(a) The almost-constant-pressure intake and exhaust strokes are assumed to be constant pressure. At WOT (Wide-Open Throttle), the intake stroke is assumed to be at a pressure Po of one atmosphere. At partially closed throttle or when supercharged, inlet pressure will be some constant value other than one atmosphere. The exhaust stroke pressure is assumed constant at one atmosphere.

AIR-STANDARD CYCLES

(b) Compression strokes and expansion strokes are approximated by isentropic processes. To be truly isentropic would require these strokes to be reversible and adiabatic. There is some friction between the piston and cylinder walls but, because the surfaces are highly polished and lubricated, this friction is kept to a minimum and the processes are close to frictionless and reversible. If this were not true, automobile engines would wear out long before the 150-200 thousand miles which they now last if properly maintained. There is also fluid friction because of the gas motion within the cylinders during these strokes. This too is minimal. Heat transfer for anyone stroke will be negligibly small due to the very short time involved for that single process. Thus, an almost reversible and almost adiabatic process can quite accurately be approximated with an isentropic process.

(c) 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).

(d) Exhaust blowdown is approximated by a constant-volume process.

(e) All processes are considered reversible.

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

$$Pv = RT \tag{a}$$

$$PV = mRT \tag{b}$$

$$P = \rho RT \tag{c}$$

$$dh = c_P dT \tag{d}$$

$$du = c_v dT \tag{e}$$

$$Pv^k = \text{constant} \qquad \text{isentropic process} \tag{f}$$

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

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

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1-k) \qquad \text{isentropic work in closed system} \tag{i}$$

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

$$c = \sqrt{kRT} \qquad \text{speed of sound} \tag{j}$$

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

m = mixture of all gases

For thermodynamic analysis the specific heats of air can be treated as functions of temperature, which they are, or they can be treated as constants, which simplifies calculations at a slight loss of accuracy.

In this chapter, 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 (see Table A-I). 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 during the operating cycle and exhaust flow, this book uses the following air property values:

$$c_p = 1.108 \text{ kJ/kg-K}$$

$$c_v = 0.821 \text{ kJ/kg-K}$$

$$k = c_P/c_v = 1.108/0.821 = 1.35$$

$$R = c_P - c_v = 0.287 \text{ kJ/kg-K}$$

TABLE A-1 THERMODYNAMIC PROPERTIES OF AIR

Temperature K	Temperature °C	c_p (kJ/kg-K)	c_v (kJ/kg-K)	$k = c_p/c_v$	Gas Constant $R = c_p - c_v$ (kJ/kg-K)
273	0	1.004	0.717	1.40	0.287
298	25	1.005	0.718	1.40	0.287
300	27	1.005	0.718	1.40	0.287
500	227	1.029	0.742	1.39	0.287
850	577	1.108	0.821	1.35	0.287
1000	727	1.140	0.853	1.34	0.287
1500	1227	1.210	0.923	1.31	0.287
2000	1727	1.249	0.962	1.30	0.287
2500	2227	1.274	0.987	1.29	0.287
3000	2727	1.291	1.004	1.29	0.287

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:

$$c_p = 1.005 \text{ kJ/kg-K}$$
$$c_v = 0.718 \text{ kJ/kg-K}$$
$$k = c_p/c_v = 1.005/0.718 = 1.40$$
$$R = c_p - c_v = 0.287 \text{ kJ/kg-K}$$

OTTO CYCLE

The cycle of a four-stroke, SI, naturally aspirated engine at WOT is shown in Fig. 4. 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. 5. This ideal air-standard cycle is called an Otto cycle, named after one of the early developers of this type of engine.

The intake stroke of the Otto cycle starts with the piston at TDC and is a constant-pressure process at an inlet pressure of one atmosphere (process 6-1 in Fig. 5).

This is a good approximation to the inlet process of a real engine at WOT, which will actually be at a pressure slightly less than atmospheric due to pressure losses in the inlet air flow. The temperature of the air during the inlet stroke is increased as the air passes through the hot intake manifold. The temperature at point 1 will generally be on the order of 25° to 35°C hotter than the surrounding air temperature.

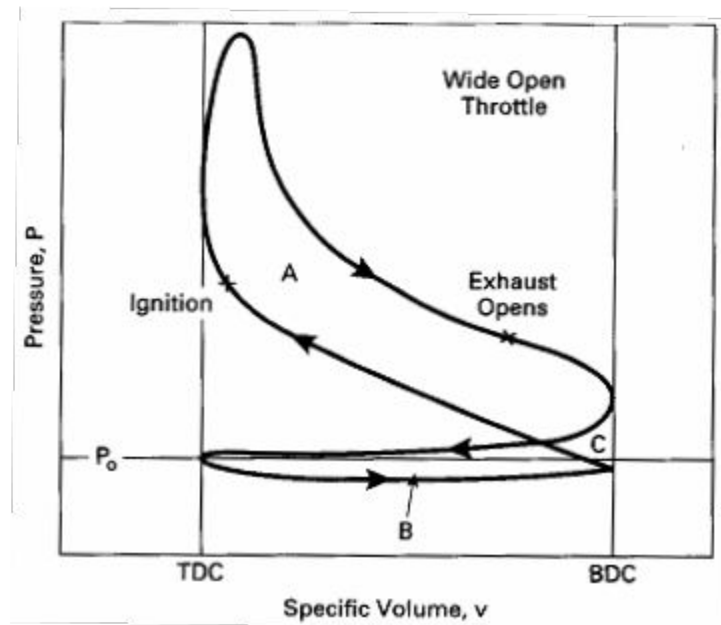


Figure 4 Indicator diagram for a typical four stroke cycle SI engine. An indicator diagram plots cylinder pressure as a function of combustion chamber volume over a 720° cycle. The diagram is generated on an oscilloscope using a pressure transducer mounted in the combustion chamber and a position sensor mounted on the piston or crankshaft.

The second stroke of the cycle is the compression stroke, which in the Otto cycle is an isentropic compression from BDC to TDC (process 1-2). This is a good approximation to compression in a real engine, except for the very beginning and the very end of the stroke. In a real engine, the beginning of the stroke is affected by the intake valve not being fully closed until slightly after BDC. The end of compression is affected by the firing of the spark plug before TDC. Not only is there an increase in pressure during the compression stroke, but the temperature within the cylinder is increased substantially due to compressive heating.

The compression stroke is followed by a constant-volume heat input process 2-3 at TDC.

This replaces the combustion process of the real engine cycle, which occurs at close to constant-volume conditions. In a real engine combustion is started slightly bTDC, reaches its maximum speed near TDC, and is terminated a little aTDC. During combustion or heat input, a large amount of energy is added to the air within the cylinder. This energy raises the temperature of the air to very high values, giving peak cycle temperature at point 3. This increase in temperature during a closed constant-volume process results in a large pressure rise also. Thus, peak cycle pressure is also reached at point 3.

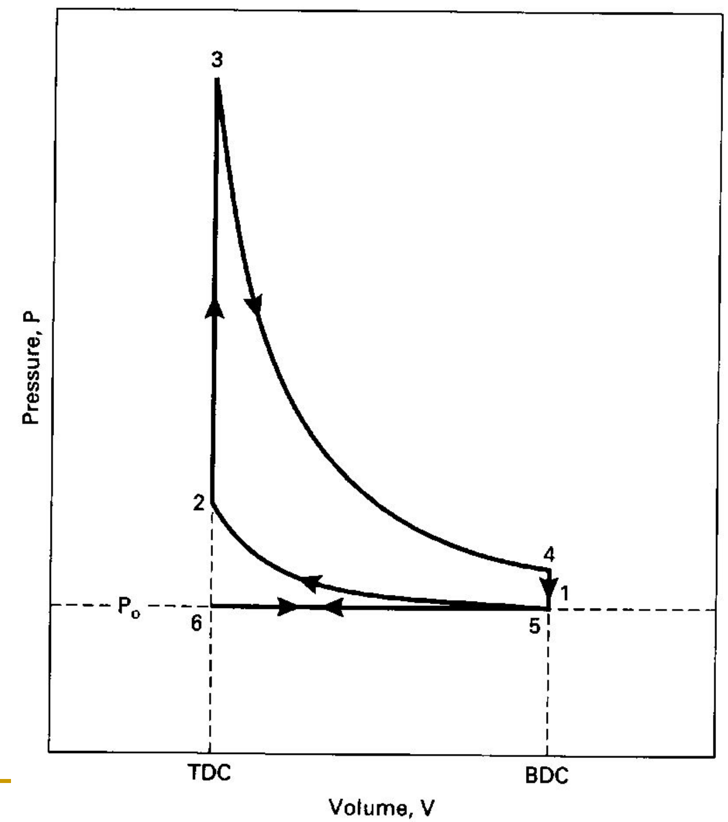


Figure 5 Ideal air-standard Otto cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an SI engine on P-V coordinates.

The very high pressure and enthalpy values within the system at TDC generate the power stroke (or expansion stroke) which follows combustion (process 3-4).

High pressure on the piston face forces the piston back towards BDC and produces the work and power output of the engine.

The power stroke of the real engine cycle is approximated with an isentropic process in the Otto cycle. This is a good approximation, subject to the same arguments as the compression stroke on being frictionless and adiabatic. In a real engine, the beginning of the power stroke is affected by the last part of the combustion process.

The end of the power stroke is affected by the exhaust valve being opened before BDC. During the power stroke, values of both the temperature and pressure within the cylinder decrease as volume increases from TDC to BDC.

Near the end of the power stroke of a real engine cycle, the exhaust valve is opened and the cylinder experiences exhaust blowdown. A large amount of exhaust gas is expelled from the cylinder, reducing the pressure to that of the exhaust manifold. The exhaust valve is opened bBDC to allow for the finite time of blowdown to occur. It is desirable for blowdown to be complete by BDC so that there is no high pressure in the cylinder to resist the piston in the following exhaust stroke.

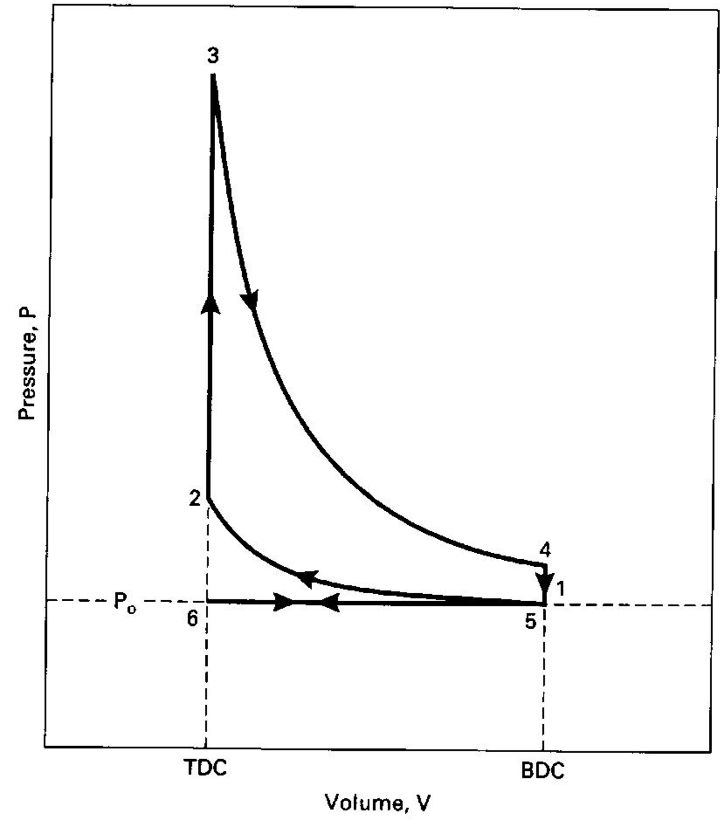


Figure 5 Ideal air-standard Otto cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an SI engine on P-V coordinates.

Blowdown in a real engine is therefore almost, but not quite, constant volume. A large quantity of enthalpy is carried away with the exhaust gases, limiting the thermal efficiency of the engine. The Otto cycle replaces the exhaust blowdown open-system process of the real cycle with a constant-volume pressure reduction, 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.

The last stroke of the four-stroke cycle now occurs as the piston travels from BDC to TDC. Process 5-6 is the exhaust stroke that occurs at a constant pressure of one atmosphere due to the open exhaust valve. This is a good approximation to the real exhaust stroke, which occurs at a pressure slightly higher than the surrounding pressure due to the small pressure drop across the exhaust valve and in the exhaust system.

At the end of the exhaust stroke the engine has experienced two revolutions, the piston is again at TDC, the exhaust valve closes, the intake valve opens, and a new cycle begins.

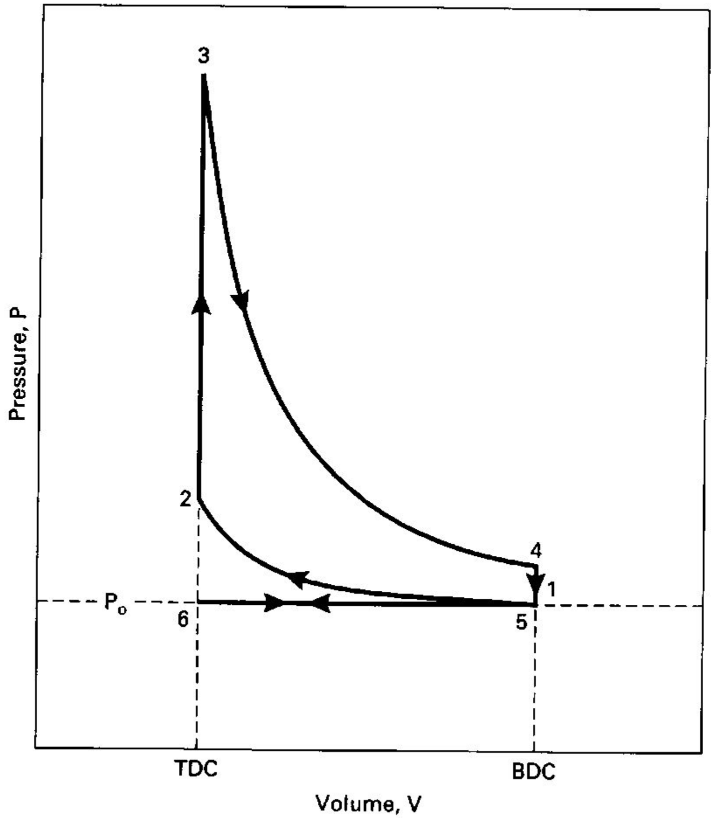


Figure 5 Ideal air-standard Otto cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an SI engine on P-V coordinates.

When analyzing an Otto cycle, it is more convenient to work with specific properties by dividing by the mass within the cylinder. Figure 6 shows the Otto cycle in P-v and T-s coordinates. It is not uncommon to find the Otto cycle shown with processes 6-1 and 5-6 left off the figure. The reasoning to justify this is that these two processes cancel each other thermodynamically and are not needed in analyzing the 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:

$$P_1 = P_6 = P_o \tag{3-2}$$

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

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

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

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

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

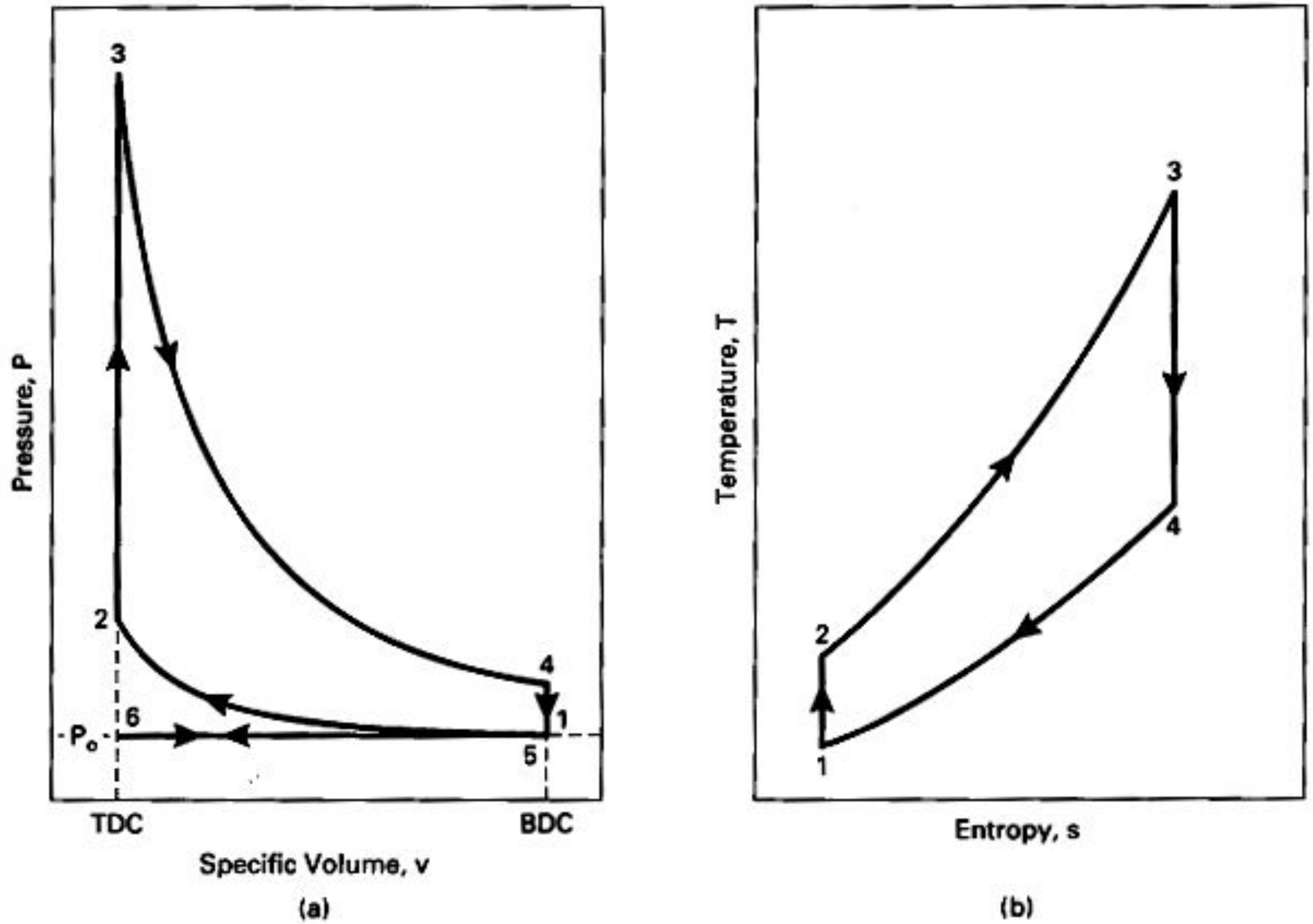


Figure 6 Otto cycle, 6-1-2-3-4-5-6, on (a) pressure-specific volume coordinates, and (b) temperature-entropy coordinates.

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

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

$$v_3 = v_2 = v_{\mathrm{TDC}} \tag{3-8}$$

$$w_{2-3} = 0 \tag{3-9}$$

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

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

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

$$T_3 = T_{\max} \tag{3-13}$$

$$P_3 = P_{\max} \tag{3-14}$$

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

$$v_5 = v_4 = v_1 = v_{BDC} \tag{3-19}$$

$$w_{4-5} = 0 \tag{3-20}$$

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

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

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

$$P_5 = P_6 = P_o \tag{3-23}$$

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

Thermal efficiency of Otto cycle:

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

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 $v_2 = v_3$:

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

Rearranging the temperature terms gives:

$$T_4/T_1 = T_3/T_2 \tag{3-27}$$

Equation (3-25) can be rearranged to:

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

Using Eq. (3-27) gives:

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

Combining this with Eq. (3-4):

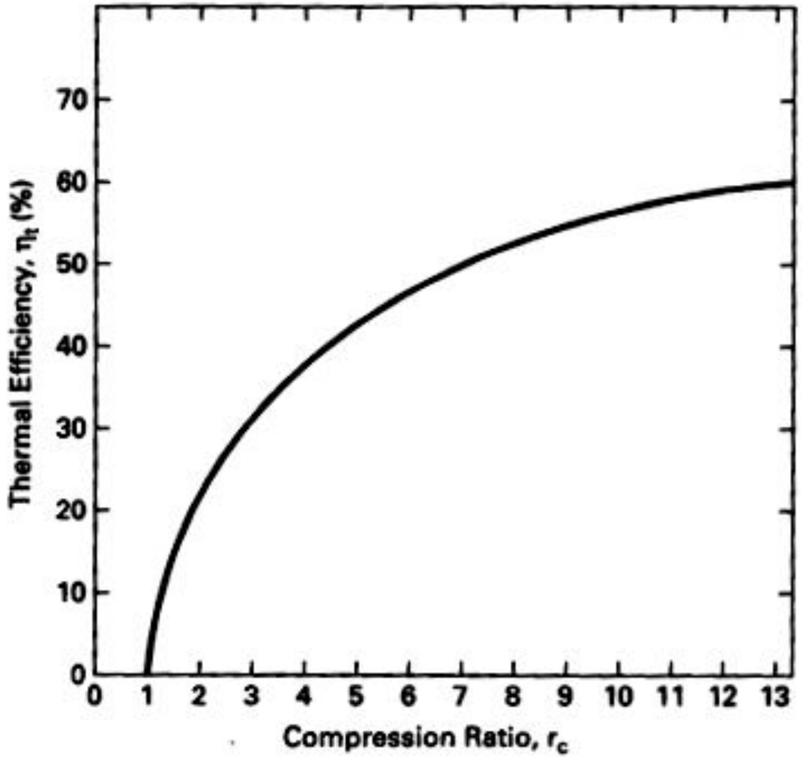


Figure 7 Indicated thermal efficiency as a function of compression ratio for SI engines operating at WOT on air-standard Otto cycle ($k = 1.35$).

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

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

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

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. 7 . This efficiency is the **indicated thermal efficiency**, as the heat transfer values are those to and from the air within the combustion chamber.

REAL AIR-FUEL ENGINE CYCLES

The actual cycle experienced by an internal combustion engine is not, in the true sense, a thermodynamic cycle. An ideal air-standard thermodynamic cycle occurs on a closed system of constant composition. This is not what actually happens in an IC engine, and for this reason air-standard analysis gives, at best, only approximations to actual conditions and outputs. Major differences include:

1. Real engines operate on an open cycle with changing composition. Not only does the inlet gas composition differ from what exits, but often the mass flow rate is not the same. Those engines which add fuel into the cylinders after air induction is complete (CI engines and some SI engines) change the amount of mass in the gas composition part way through the cycle. There is a greater gaseous mass exiting the engine in the exhaust than what entered in the induction process. This can be on the order of several percent. Other engines carry liquid fuel droplets with the inlet air which are idealized as part of the gaseous mass in air-standard analysis. During combustion, total mass remains about the same but molar quantity changes. Finally, there is a loss of mass during the cycle due to crevice flow and blowby past the pistons. Most of crevice flow is a temporary loss of mass from the cylinder, but because it is greatest at the start of the power stroke, some output work is lost during expansion. Blowby can decrease the amount of mass in the cylinders by as much as 1% during compression and combustion.

2. Air-standard analysis treats the fluid flow through the entire engine as air and approximates air as an ideal gas. 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. During combustion the composition is then changed to a gas mixture of mostly CO_2, H_20, and N_2, with lesser amounts of CO and hydrocarbon vapor. In CI engines there will also be solid carbon particles in the combustion products gas mixture. Approximating exhaust products as air simplifies analysis but introduces some error.

Even if all fluid in an engine cycle were air, some error would be introduced by assuming it to be an ideal gas with constant specific heats in air-standard analysis. At the low pressures of inlet and exhaust, air can accurately be treated as an ideal gas, but at the higher pressures during combustion, air will deviate from ideal gas behavior.

A more serious error is introduced by assuming constant specific heats for the analysis. Specific heats of a gas have a fairly strong dependency on temperature and can vary as much as 30% in the temperature range of an engine (for air, c_p = 1.004 kJ/kg-K at 300 K and c_p = 1.292 kJ/kg-K at 3000 K.

3. There are heat losses during the cycle of a real engine which are neglected in air-standard analysis. Heat loss during combustion lowers actual peak temperature and pressure from what is predicted. The actual power stroke, therefore, starts at a lower pressure, and work output during expansion is decreased. Heat transfer continues during expansion, and this lowers the temperature and pressure below the ideal isentropic process towards the end of the power stroke. The result of heat transfer is a lower indicated thermal efficiency than predicted by air-standard analysis. Heat transfer is also present during compression, which deviates the process from isentropic. However, this is less than during the expansion stroke due to the lower temperatures at this time.

4. Combustion requires a short but finite time to occur, and heat addition is not instantaneous at TDC, as approximated in an Otto cycle. A fast but finite flame speed is desirable in an engine. This results in a finite rate of pressure rise in the cylinders, a steady force increase on the piston face, and a smooth engine cycle. A supersonic detonation would give almost instantaneous heat addition to a cycle, but would result in a.rough cycle and quick engine destruction. Because of the finite time required, combustion is started before TDC and ends after TDC, not at constant volume as in air-standard analysis. By starting combustion bTDC, cylinder pressure increases late in the compression stroke, requiring greater negative work in that stroke. Because combustion is not completed until aTDC, some power is lost at the start of the expansion stroke (see Fig. 1). Another loss in the combustion process of an actual engine occurs because combustion efficiency is less than 100%. This happens because of less than perfect mixing, local variations in temperature and air-fuel due to turbulence, flame quenching, etc. SI engines will generally have a combustion efficiency of about 95%, while CI engines are generally about 98% efficient.

5. The blowdown process requires a finite real time and a finite cycle time, and does not occur at constant volume as in air-standard analysis. For this reason, the exhaust valve must open *40°* to *60°* bBDC, and output work at the latter end of expansion is lost.

6. In an actual engine, the intake valve is not closed until after bottom-dead center at the end of the intake stroke. Because of the flow restriction of the valve, air is still entering the cylinder at BDC, and volumetric efficiency would be lower if the valve closed here. Because of this, however, actual compression does not start at BDC but only after the inlet valve closes. With ignition then occurring before top dead-center, temperature and pressure rise before combustion is less than predicted by air-standard analysis.

7. Engine valves require a finite time to actuate. Ideally, valves would open and close instantaneously, but this is not possible when using a camshaft. Cam profiles must allow for smooth interaction with the cam follower, and this results in fast but finite valve actuation. To assure that the intake valve is fully open at the start of the induction stroke, it must start to open before TDC. Likewise, the exhaust valve must remain fully open until the end of the exhaust stroke, with final closure occurring after TDC. The resulting valve overlap period causes a deviation from the ideal cycle.

Because of these differences which real air-fuel cycles have from the ideal cycles, results from air-standard analysis will have errors and will deviate from actual conditions. Interestingly, however, the errors are not great, and property values of temperature and pressure are very representative of actual engine values, depending on the geometry and operating conditions of the real engine. By changing operating variables such as inlet temperature and/or pressure, compression ratio, peak temperature, etc., in Otto cycle analysis, good approximations can be obtained for output changes that will Occur in a real engine as these variables are changed. Good approximation of power output, thermal efficiency, and mep can be expected.

Indicated thermal efficiency of a real four-stroke SI engine is always somewhat less than what air-standard Otto cycle analysis predicts. This is caused by the heat losses, friction, ignition timing, valve timing, finite time of combustion and blowdown, and deviation from ideal gas behavior of the real engine. Reference shows that over a large range of operating variables the indicated thermal efficiency of an actual SI four-stroke cycle engine can be approximated by:

(T/t)actual = 0.85 *(T/t)OTTO* (3-32)

This will be correct to within a few percent for large ranges of air-fuel equivalence ratio, ignition timing, engine speed, compression ratio, inlet pressure, exhaust pressure, and valve timing.

DIESEL CYCLE

Early CI engines injected fuel into the combustion chamber very late in the compression stroke, resulting in the indicator diagram shown in Fig. 8. Due to ignition delay and the finite time required to inject the fuel, combustion lasted into the expansion stroke.

This kept the pressure at peak levels well past TDC. This combustion process is best approximated as a constant-pressure heat input in an air-standard cycle, resulting in the Diesel cycle shown in Fig. 9. The rest of the cycle is similar to the air-standard Otto cycle. The diesel cycle is sometimes called a Constant Pressure cycle.

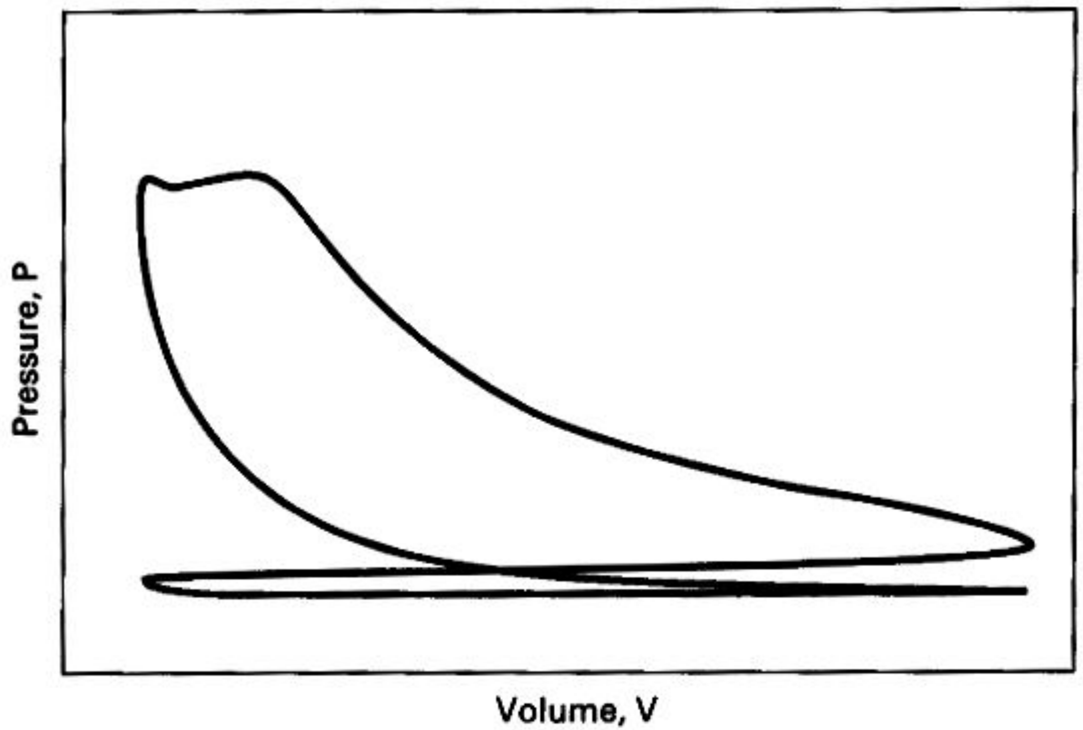


Figure 8 Indicator diagram of a historic CI engine operating on an early four-stroke cycle.

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:

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

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

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

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

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1-k) = R(T_2 - T_1)/(1-k) \tag{3-56}$$

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

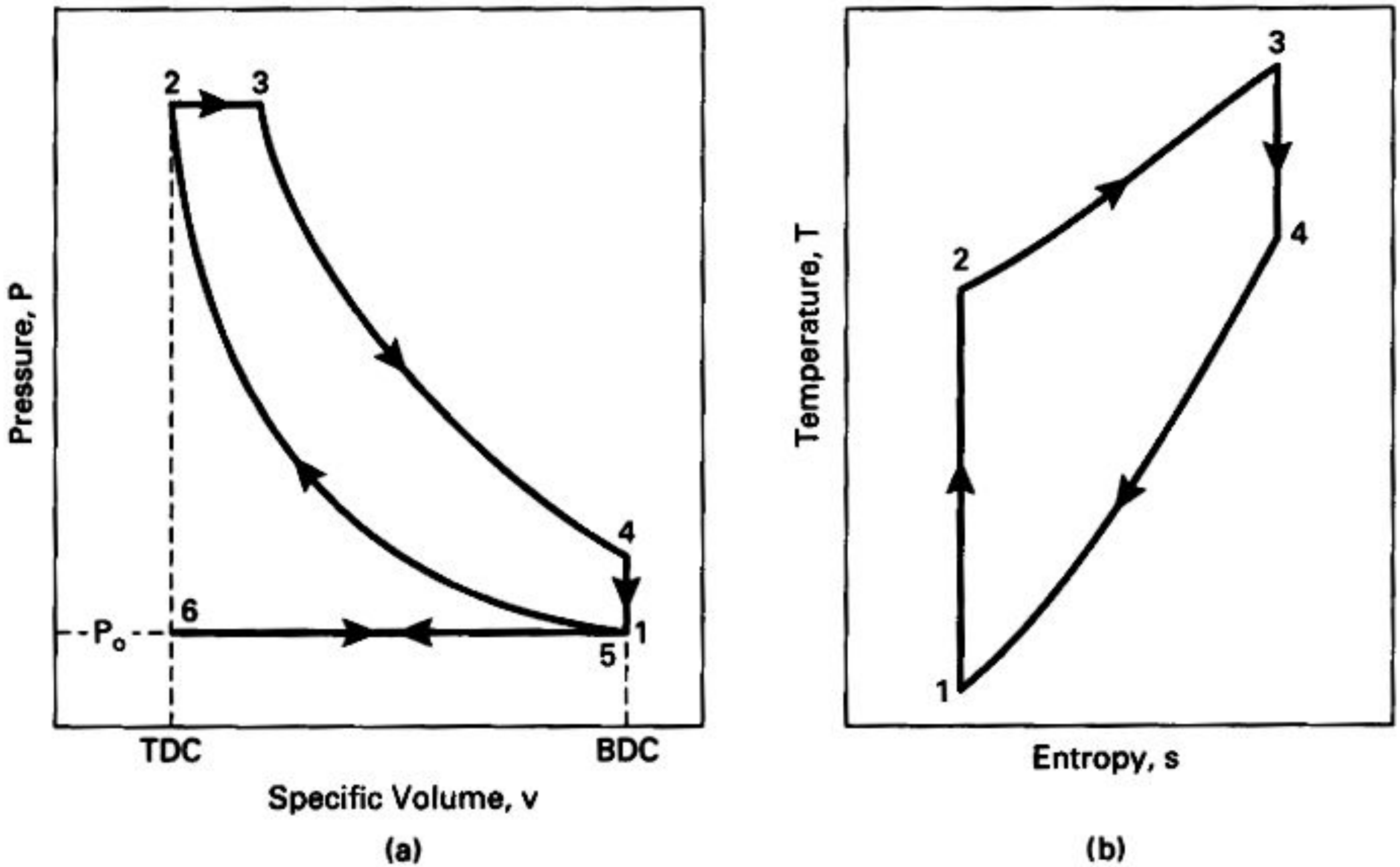


Figure 9 Air-standard diesel cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an early CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

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:

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

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

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

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

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

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

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

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

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

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

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:

$$\begin{aligned}(\eta_t)_{\text{DIESEL}} &= |w_{net}|/|q_{in}| = 1 - (|q_{out}|/|q_{in}|) \quad (3\text{-}72)\\ &= 1 - [c_v(T_4 - T_1)/c_p(T_3 - T_2)]\\ &= 1 - (T_4 - T_1)/[k(T_3 - T_2)]\end{aligned}$$

With rearrangement, this can be shown to equal:

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

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

With rearrangement, this can be shown to equal:

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

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

If reptesentative numbers are introduced into Eq. (3-73), it is found that the value of the term in brackets is greater than one. When this equation is compared with Eq. (3-31), it can be seen that for a given compression ratio the thermal efficiency of the Otto cycle would be greater than the thermal efficiency of the Diesel cycle. Constant-volume combustion at TDC is more efficient than constant-pressure combustion. However, it must be remembered that CI engines operate with much higher compression ratios than SI engines (12 to 24 versus 8 to 11) and thus have higher thermal efficiencies.

TWO-STROKE CYCLE

Two-Stroke SI Engine Cycle

An air-standard approximation to a typical two-stroke SI engine cycle is shown in Fig. 10.

Process 1-2—isentropic power or expansion stroke.
All ports (or valves) closed:

$$T_2 = T_1(V_1/V_2)^{k-1} \tag{3-95}$$

$$P_2 = P_1(V_1/V_2)^k \tag{3-96}$$

$$q_{1-2} = 0 \tag{3-97}$$

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1-k) = R(T_2 - T_1)/(1-k) \tag{3-98}$$

Process 2-3—exhaust blowdown.
Exhaust port open and intake port closed:

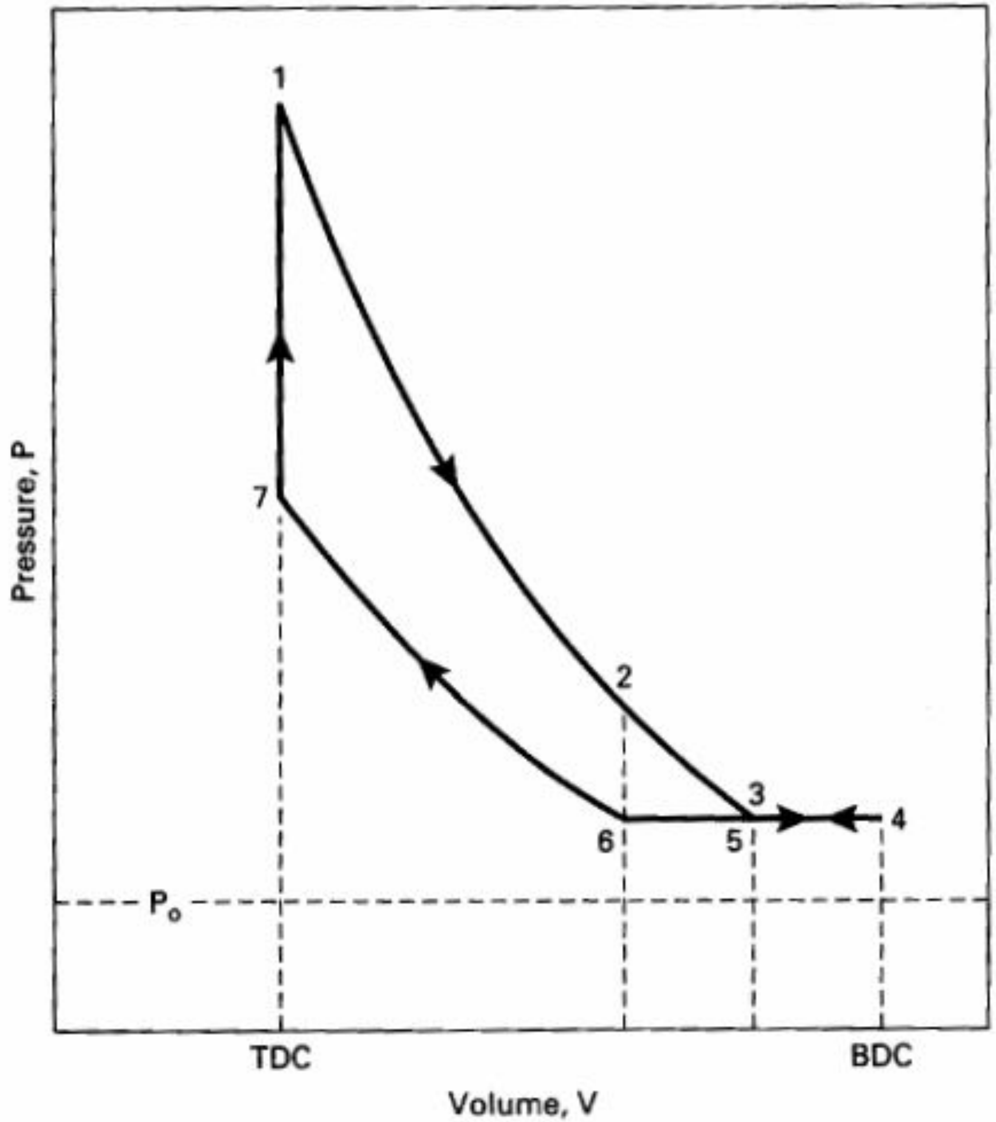


Figure 10 Air-standard approximation for a two-stroke cycle SI engine, 1-2-3-4-5-6-7-1.

Process 3-4-5-intake, and exhaust scavenging.

Exhaust port open and intake port open:

Intake air entering at an absolute pressure on the order of 140-180 kPa fills and scavenges the cylinder. Scavenging is a process in which the air pushes out most of the remaining exhaust residual from the previous cycle through the open exhaust port into the exhaust system, which is at about one atmosphere pressure. The piston uncovers the intake port at point 3, reaches BDC at point 4, reverses direction, and again closes the intake port at point 5. In some engines fuel is mixed with the incoming air. In other engines the fuel is injected later, after the exhaust port is closed.

Process 5-6- exhaust scavenging.

Exhaust port open and intake port closed:

Exhaust scavenging continues until the exhaust port is closed at point 6.

Process 6-7—isentropic compression.

All ports closed:

$$T_7 = T_6(V_6/V_7)^{k-1} \tag{3-99}$$

$$P_7 = P_6(V_6/V_7)^k \tag{3-100}$$

$$q_{6-7} = 0 \tag{3-101}$$

$$w_{6-7} = (P_7 v_7 - P_6 v_6)/(1-k) = R(T_7 - T_6)/(1-k) \tag{3-102}$$

In some engines, fuel is added very early in the compression process. The spark plug is fired near the end of process 6-7.

Process 7-1—constant-volume heat input (combustion).

All ports closed:

$$V_7 = V_1 = V_{\mathrm{TDC}} \tag{3-103}$$

$$W_{7-1} = 0 \tag{3-104}$$

$$Q_{7-1} = Q_{\mathrm{in}} = m_f Q_{\mathrm{HV}} \eta_c = m_m c_v(T_1 - T_7) \tag{3-105}$$

$$T_1 = T_{\max} \tag{3-106}$$

$$P_1 = P_{\max} = P_7(T_1/T_7) \tag{3-107}$$

TWO-STROKE CI ENGINE CYCLE

Many compression ignition engines-especially large ones-operate on two-stroke cycles. These cycles can be approximated by the air-standard cycle shown in Fig. 11. This cycle is the same as the two-stroke SI cycle except for the fuel input and combustion process. Instead of adding fuel to the intake air or early in the compression process, fuel is added with injectors late in the compression process, the same as with four-stroke cycle CI engines. Heat input or combustion can be approximated by a two-step (dual) process .

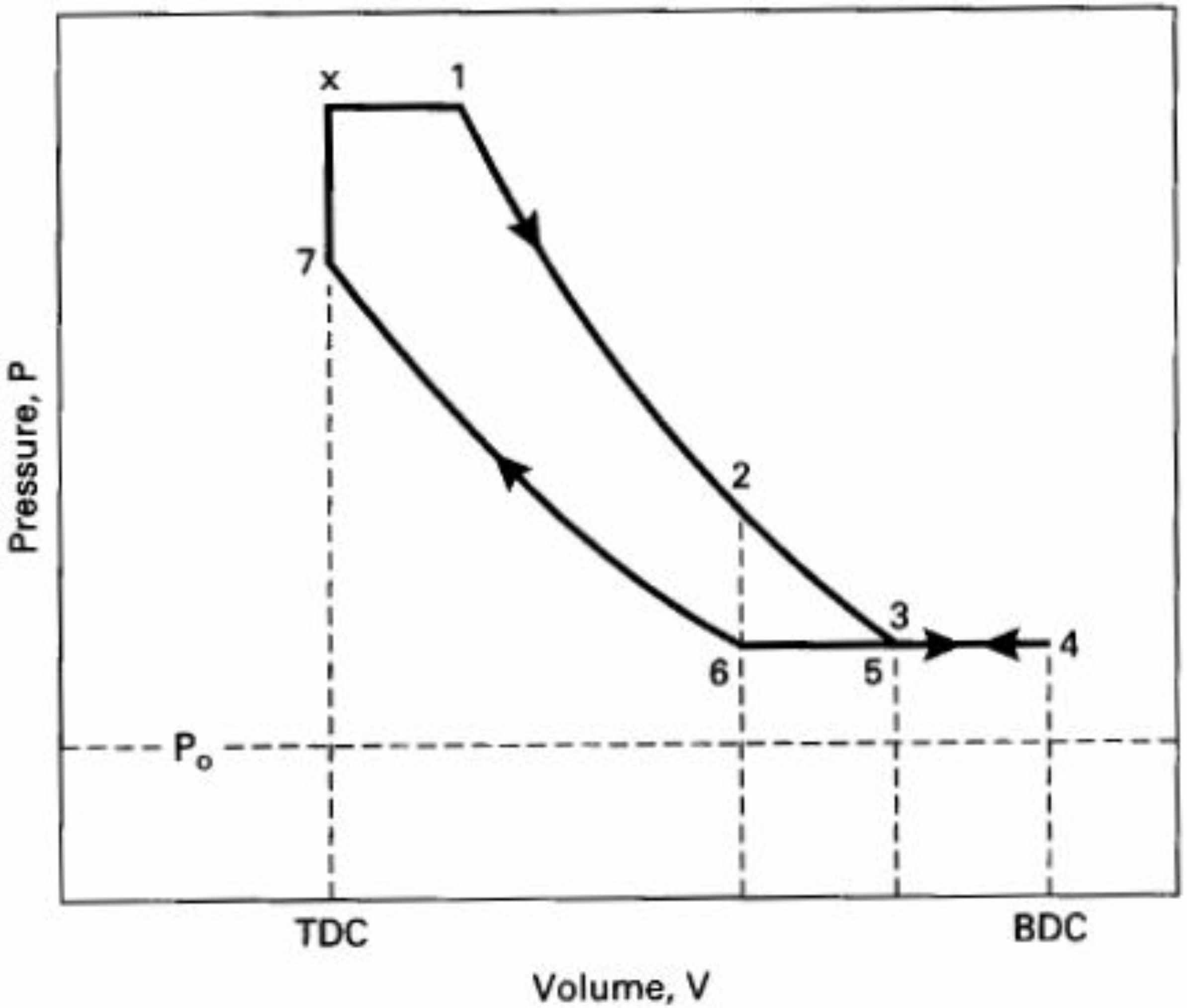


Figure 11 Air-standard approximation for a two-stroke cycle CI engine, 1-2-3-4-5-6-7-*x*-1.

Process 7-x—constant-volume heat input (first part of combustion).
All ports closed:

$$V_7 = V_x = V_{\mathrm{TDC}} \tag{3-108}$$

$$W_{7-x} = 0 \tag{3-109}$$

$$Q_{7-x} = m_m c_v (T_x - T_7) \tag{3-110}$$

$$P_x = P_{\max} = P_7 (T_x / T_7) \tag{3-111}$$

Process x-1—constant-pressure heat input (second part of combustion).
All ports closed:

$$P_1 = P_x = P_{\max} \tag{3-112}$$

$$W_{x-1} = P_1 (V_1 - V_x) \tag{3-113}$$

$$Q_{x-1} = m_m c_p (T_1 - T_x) \tag{3-114}$$

$$T_1 = T_{\max} \tag{3-115}$$

SUMMARY

This chapter reviewed the basic cycles used in internal combustion engines. Although many engine cycles have been developed, for over a century most automobile engines have operated on the basic SI four-stroke cycle developed in the 1870s by Otto and others. This can be approximated and analyzed using the ideal air-standard Otto cycle. Many small SI engines operate on a two-stroke cycle, sometimes (erroneously) called a two-stroke Otto cycle. Early four-stroke CI engines operated on a cycle that can be approximated by the air-standard Diesel cycle. This cycle was improved in modern CI engines of the type used in automobiles and trucks.

Most small CI engines and very large CI engines operate on a two-stroke cycle. At present, most automobile engines operate on the four-stroke Otto cycle, but major research and development is resulting in two additional cycles for modem vehicles. Several companies have done major development work to try to create an automobile engine that would operate on an SI two-stroke cycle. Throughout history, two-stroke cycle automobile engines have periodically appeared with varying success. These offer greater power per unit weight, but none would pass modem emission standards. Recent development has concentrated on producing an engine that would satisfy pollution laws. The major technological change is the input of fuel by injection directly into the combustion chamber after exhaust and air intake are completed. If this development work is successful, there will be automobiles on the market with two-stroke cycle engines.