

# ESO201A : THERMODYNAMICS

## 2021-22 1st semester

### IIT Kanpur

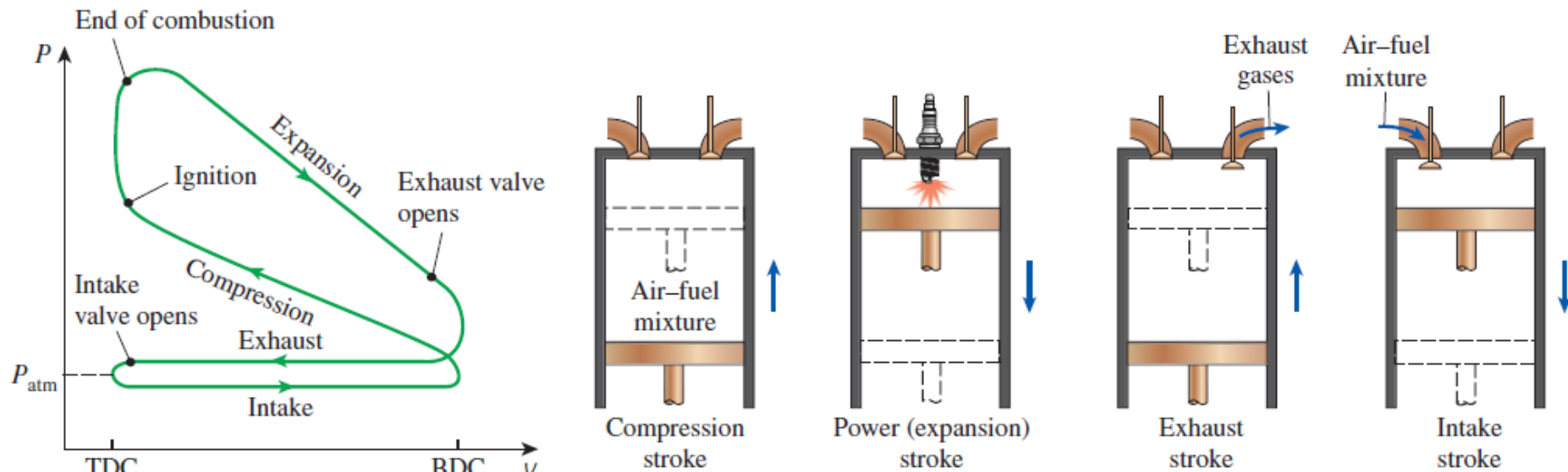
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## Lecture 26

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# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

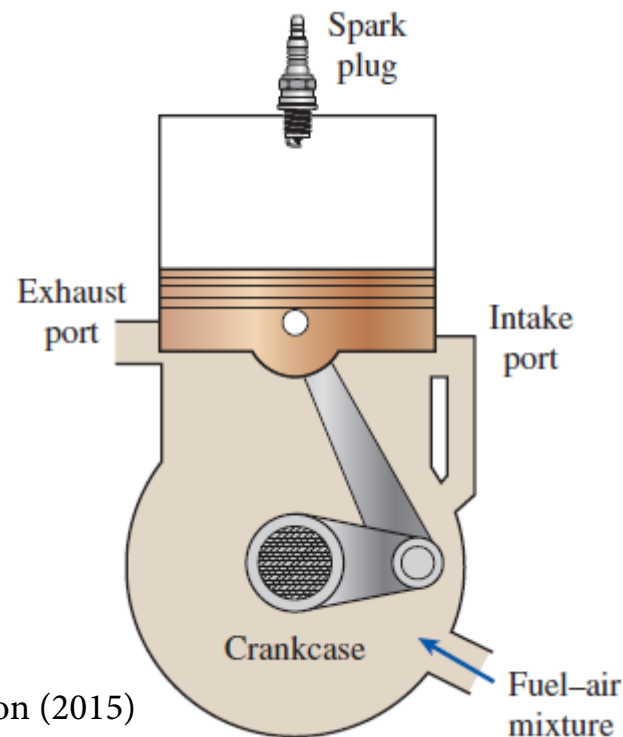
The Otto cycle is the ideal cycle for spark-ignition reciprocating engines. It is named after Nikolaus A. Otto, who built a successful four-stroke engine in 1876 in Germany using the cycle proposed by Frenchman Beau de Rochas in 1862. In most spark-ignition engines, the piston executes four complete strokes (two mechanical cycles) within the cylinder, and the crankshaft completes two revolutions for each thermodynamic cycle. These engines are called **four-stroke** internal combustion engines. A schematic of each stroke as well as a  $P$ - $v$  diagram for an actual four-stroke spark-ignition engine is given in Fig.



(a) Actual four-stroke spark-ignition engine

In **two-stroke engines**, all four functions described above are executed in just two strokes: the power stroke and the compression stroke. In these engines, the crankcase is sealed, and the outward motion of the piston is used to slightly pressurize the air–fuel mixture in the crankcase, as shown in Fig.

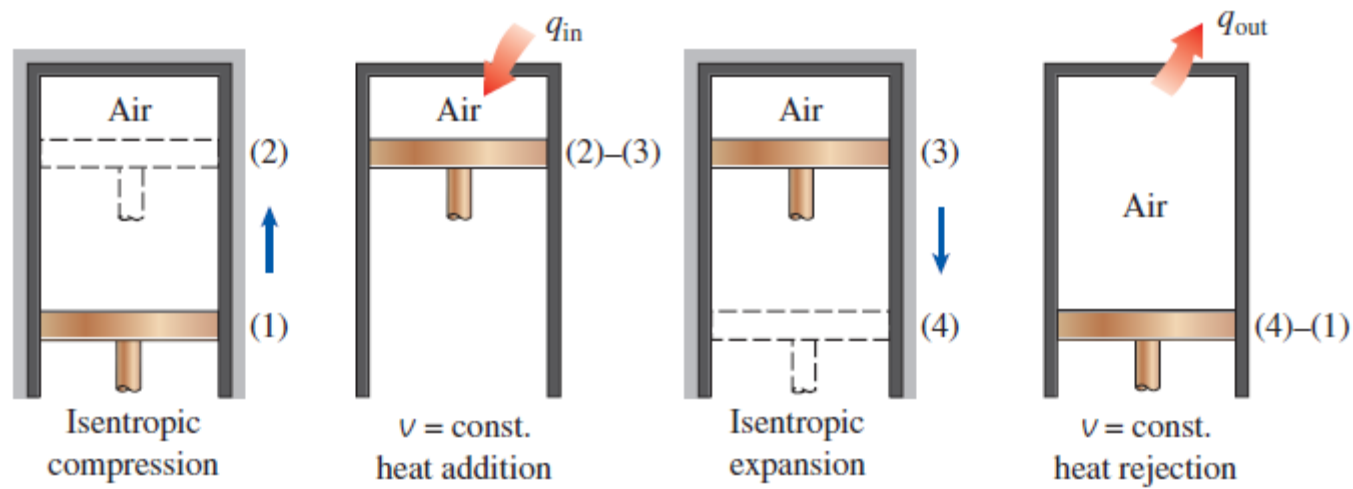
Also, the intake and exhaust valves are replaced by openings in the lower portion of the cylinder wall. During the latter part of the power stroke, the piston uncovers first the exhaust port, allowing the exhaust gases to be partially expelled, and then the intake port, allowing the fresh air–fuel mixture to rush in and drive most of the remaining exhaust gases out of the cylinder. This mixture is then compressed as the piston moves upward during the compression stroke and is subsequently ignited by a spark plug.



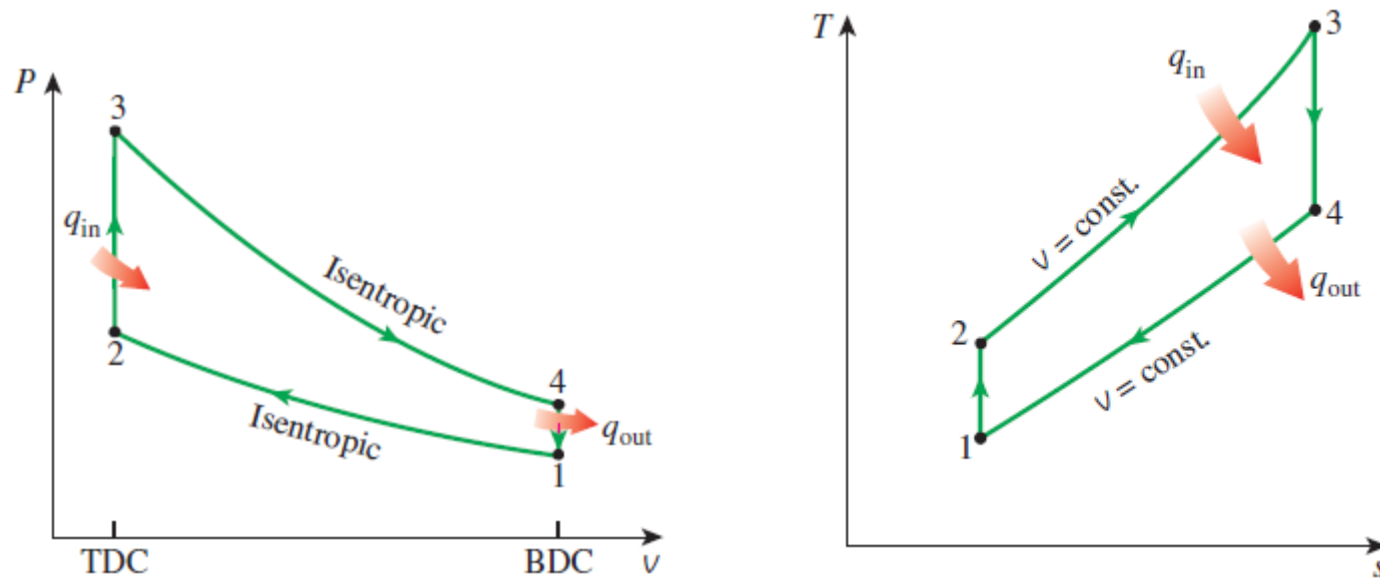
The two-stroke engines are generally less efficient than their four-stroke counterparts because of the incomplete expulsion of the exhaust gases and the partial expulsion of the fresh air–fuel mixture with the exhaust gases. However, they are relatively simple and inexpensive, and they have high power-to-weight and power-to-volume ratios, which make them suitable for applications requiring small size and weight such as for motorcycles, chain saws, and lawn mowers

The thermodynamic analysis of the actual four-stroke or two-stroke cycles described is not a simple task. However, the analysis can be simplified significantly if the air-standard assumptions are utilized. The resulting cycle, which closely resembles the actual operating conditions, is the ideal **Otto cycle**. It consists of four internally reversible processes:

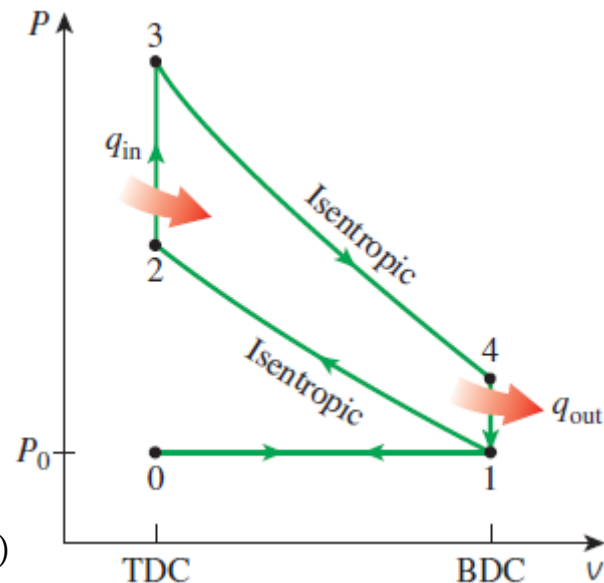
- 1-2 Isentropic compression
- 2-3 Constant-volume heat addition
- 3-4 Isentropic expansion
- 4-1 Constant-volume heat rejection



(b) Ideal Otto cycle



The ideal Otto cycle has one shortcoming. This ideal cycle consists of two strokes equivalent to one mechanical cycle or one crankshaft rotation. The actual engine operation on the other hand, involves four strokes equivalent to two mechanical cycles or two crankshaft rotations. This can be corrected by including intake and exhaust strokes in the ideal Otto cycle, as shown in Fig. . In this modified cycle, air-fuel mixture (approximated as air due to air-standard assumptions) enters the cylinder through the open intake valve at atmospheric pressure  $P_0$  during process 0-1 as the piston moves from TDC to BDC. The intake valve is closed at state 1 and air is compressed isentropically to state 2. Heat is transferred at constant volume (process 2-3); it is expanded isentropically to state 4; and heat is rejected at constant volume (process 4-1). Exhaust gases (again approximated as air) are expelled through the open exhaust valve (process 1-0) as the pressure remains constant at  $P_0$ .





The modified Otto cycle is executed in an open system during the intake and exhaust processes and in a closed system during the remaining four processes. We should point out that the constant-volume heat addition process (2-3) in the ideal Otto cycle replaces the combustion process of the actual engine operation while the constant-volume heat rejection process (4-1) replaces the exhaust blowdown.

The work interactions during the constant-pressure intake (0-1) and constant-pressure exhaust (1-0) processes can be expressed as

$$W_{\text{out},0-1} = P_0(v_1 - v_0)$$

$$W_{\text{in},1-0} = P_0(v_1 - v_0)$$

These two processes cancel each other as the work output during the intake is equal to work input during the exhaust.

Therefore, inclusion of the intake and exhaust processes has no effect on the net work output from the cycle. However, when calculating power output from the cycle during an ideal Otto cycle analysis, we must consider the fact that the ideal Otto cycle has four strokes just like actual four-stroke spark-ignition engine. This is illustrated in the last part of Example 9-2.

The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit-mass basis, as

$$(q_{\text{in}} - q_{\text{out}}) + (w_{\text{in}} - w_{\text{out}}) = \Delta u \quad (\text{kJ/kg})$$

No work is involved during the two heat transfer processes since both take place at constant volume. Therefore, heat transfer to and from the working fluid can be expressed as

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

and

$$q_{\text{out}} = u_4 - u_1 = c_v(T_4 - T_1)$$

Then the thermal efficiency of the ideal Otto cycle under the cold air standard assumptions becomes

$$\eta_{\text{th, Otto}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$



Processes 1-2 and 3-4 are isentropic, and  $v_2 = v_3$  and  $v_4 = v_1$ . Thus,

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3}$$

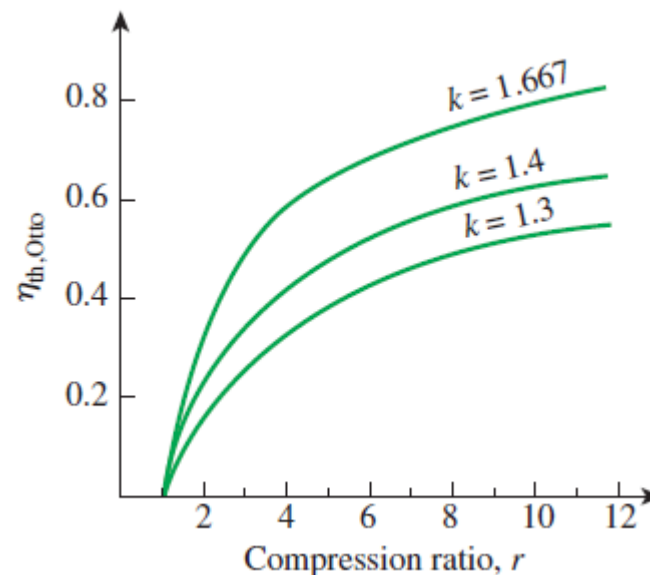
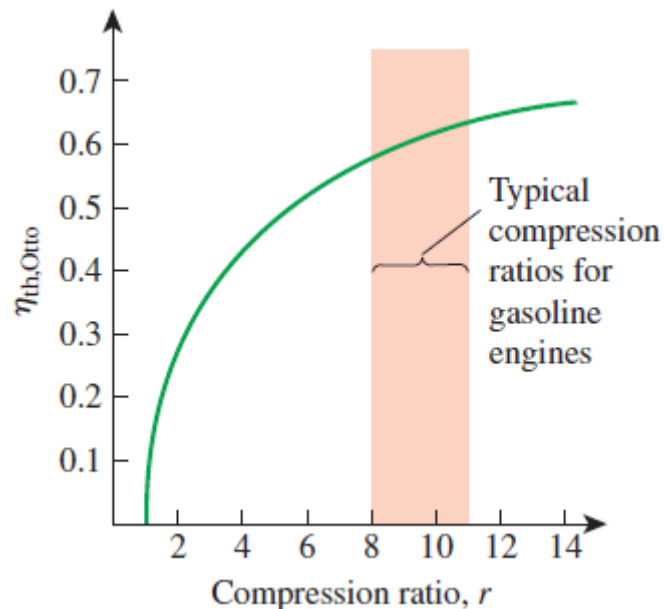
Substituting these equations into the thermal efficiency relation and simplifying give

$$\eta_{\text{th,Otto}} = 1 - \frac{1}{r^{k-1}}$$

where

$$r = \frac{V_{\text{max}}}{V_{\text{min}}} = \frac{V_1}{V_2} = \frac{v_1}{v_2}$$

is the **compression ratio** and  $k$  is the specific heat ratio  $c_p/c_v$ .



## EXAMPLE 9–2 The Ideal Otto Cycle

An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 100 kPa and 17°C, and 800 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Accounting for the variation of specific heats of air with temperature, determine (a) the maximum temperature and pressure that occur during the cycle, (b) the net work output, (c) the thermal efficiency, and (d) the mean effective pressure for the cycle.

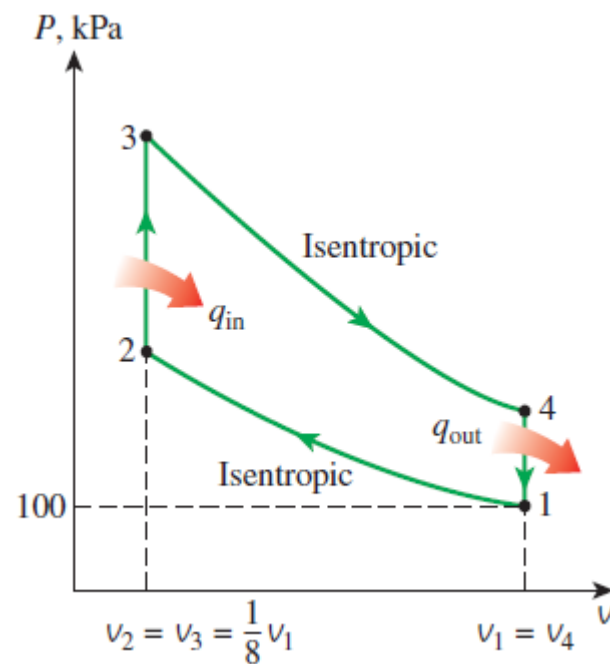
(e) Also, determine the power output from the cycle, in kW, for an engine speed of 4000 rpm (rev/min). Assume that this cycle is operated on an engine that has four cylinders with a total displacement volume of 1.6 L.

**SOLUTION** An ideal Otto cycle is considered. The maximum temperature and pressure, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

**Assumptions** 1 The air-standard assumptions are applicable. 2 Kinetic and potential energy changes are negligible. 3 The variation of specific heats with temperature is to be accounted for.

(a) The maximum temperature and pressure in an Otto cycle occur at the end of the constant-volume heat-addition process (state 3). But first we need to determine the temperature and pressure of air at the end of the isentropic compression process (state 2), using data from Table A–17:

$$T_1 = 290 \text{ K} \rightarrow u_1 = 206.91 \text{ kJ/kg} \\ v_{r1} = 676.1$$



Process 1-2 (isentropic compression of an ideal gas):

$$\frac{v_{r2}}{v_{r1}} = \frac{v_2}{v_1} = \frac{1}{r} \rightarrow v_{r2} = \frac{v_{r1}}{r} = \frac{676.1}{8} = 84.51 \rightarrow T_2 = 652.4 \text{ K}$$

$$u_2 = 475.11 \text{ kJ/kg}$$

$$\begin{aligned} \frac{P_2 v_2}{T_2} &= \frac{P_1 v_1}{T_1} \rightarrow P_2 = P_1 \left( \frac{T_2}{T_1} \right) \left( \frac{v_1}{v_2} \right) \\ &= (100 \text{ kPa}) \left( \frac{652.4 \text{ K}}{290 \text{ K}} \right) (8) = 1799.7 \text{ kPa} \end{aligned}$$

$$q_{\text{in}} = u_3 - u_2$$

$$800 \text{ kJ/kg} = u_3 - 475.11 \text{ kJ/kg}$$

$$u_3 = 1275.11 \text{ kJ/kg} \rightarrow T_3 = \mathbf{1575.1 \text{ K}}$$

$$v_{r3} = 6.108$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \rightarrow P_3 = P_2 \left( \frac{T_3}{T_2} \right) \left( \frac{v_2}{v_3} \right)$$

$$= (1.7997 \text{ MPa}) \left( \frac{1575.1 \text{ K}}{652.4 \text{ K}} \right) (1) = \mathbf{4.345 \text{ MPa}}$$

(b) The net work output for the cycle is determined either by finding the boundary ( $P dV$ ) work involved in each process by integration and adding them or by finding the net heat transfer that is equivalent to the net work done during the cycle. We take the latter approach. However, first we need to find the internal energy of the air at state 4:

Process 3-4 (isentropic expansion of an ideal gas):

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3} = r \rightarrow v_{r4} = r v_{r3} = (8)(6.108) = 48.864 \rightarrow T_4 = 795.6 \text{ K}$$

$$u_4 = 588.74 \text{ kJ/kg}$$

Process 4-1 (constant-volume heat rejection):

$$-q_{\text{out}} = u_1 - u_4 \rightarrow q_{\text{out}} = u_4 - u_1$$

$$q_{\text{out}} = 588.74 - 206.91 = 381.83 \text{ kJ/kg}$$

Thus,

$$w_{\text{net}} = q_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 800 - 381.83 = \mathbf{418.17 \text{ kJ/kg}}$$

(c) The thermal efficiency of the cycle is determined from its definition:

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{418.17 \text{ kJ/kg}}{800 \text{ kJ/kg}} = \mathbf{0.523} \text{ or } \mathbf{52.3\%}$$

Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be (Eq. 9–8)

$$\eta_{\text{th, Otto}} = 1 - \frac{1}{r^{k-1}} = 1 - r^{1-k} = 1 - (8)^{1-1.4} = 0.565 \text{ or } 56.5\%$$

which is considerably different from the value obtained above. Therefore, care should be exercised in utilizing the cold-air-standard assumptions.

(d) The mean effective pressure is determined from its definition,

$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{w_{\text{net}}}{v_1 - v_1/r} = \frac{w_{\text{net}}}{v_1(1 - 1/r)}$$

where

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})}{100 \text{ kPa}} = 0.8323 \text{ m}^3/\text{kg}$$

Thus,

$$\text{MEP} = \frac{418.17 \text{ kJ/kg}}{(0.8323 \text{ m}^3/\text{kg})(1 - \frac{1}{8})} \left( \frac{1 \text{ kPa} \cdot \text{m}^3}{1 \text{ kJ}} \right) = \mathbf{574 \text{ kPa}}$$



(e) The total air mass taken by all four cylinders when they are charged is

$$m = \frac{V_d}{v_1} = \frac{0.0016 \text{ m}^3}{0.8323 \text{ m}^3/\text{kg}} = 0.001922 \text{ kg}$$

The net work produced by the cycle is

$$W_{\text{net}} = mw_{\text{net}} = (0.001922 \text{ kg})(418.17 \text{ kJ/kg}) = 0.8037 \text{ kJ}$$

That is, the net work produced per thermodynamic cycle is 0.8037 kJ/cycle. Noting that there are two revolutions per thermodynamic cycle ( $n_{\text{rev}} = 2 \text{ rev/cycle}$ ) in a four-stroke engine (or in the ideal Otto cycle including intake and exhaust strokes), the power produced by the engine is determined from

$$\dot{W}_{\text{net}} = \frac{W_{\text{net}} \dot{n}}{n_{\text{rev}}} = \frac{(0.8037 \text{ kJ/cycle})(4000 \text{ rev/min}) \left( \frac{1 \text{ min}}{60 \text{ s}} \right)}{2 \text{ rev/cycle}} = \mathbf{26.8 \text{ kW}}$$

**Discussion** If we analyzed a two-stroke engine operating on an ideal Otto cycle with the same values, the power output would be calculated as

$$\dot{W}_{\text{net}} = \frac{W_{\text{net}} \dot{n}}{n_{\text{rev}}} = \frac{(0.8037 \text{ kJ/cycle})(4000 \text{ rev/min}) \left( \frac{1 \text{ min}}{60 \text{ s}} \right)}{1 \text{ rev/cycle}} = 53.6 \text{ kW}$$

Note that there is one revolution in one thermodynamic cycle in two-stroke engines.