

The First Law of Thermodynamics for Closed Systems Part IV

The Air-Standard Otto Cycle (Spark-Ignition) Engine

The Air Standard Otto cycle is the ideal cycle for Spark-Ignition (SI) internal combustion engines, first proposed by Nikolaus Otto over 130 years ago, and which is currently used most motor vehicles. The following link by the Kruse Technology Partnership presents a description of the four-stroke Otto cycle operation including a short history of Nikolaus Otto. Once again we have excellent animations produced by Matt Keveney presenting both the four-stroke and the two-stroke spark-ignition internal combustion engine operation

The analysis of the Otto cycle is very similar to that of the Diesel cycle which we analysed in the previous section. We will use the ideal "air-standard" assumption in our analysis. Thus the working fluid is a fixed mass of air undergoing the complete cycle which is treated throughout as an ideal gas. All processes are ideal, combustion is replaced by heat addition to the air, and exhaust is replaced by a heat rejection process which restores the air to the initial state.

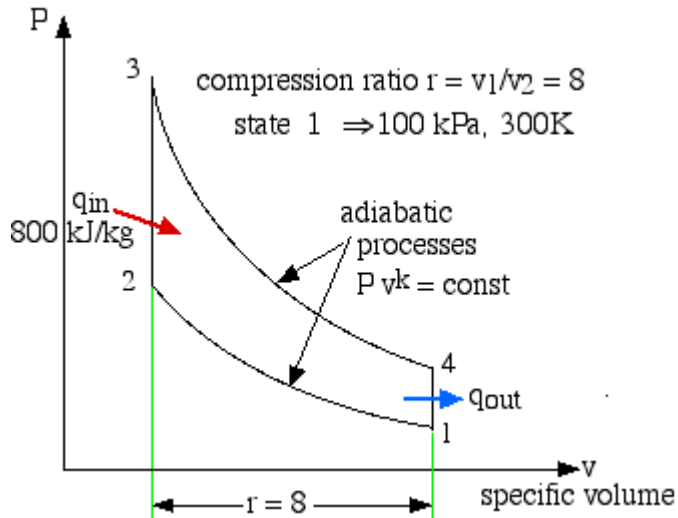
The most significant difference between the ideal Otto cycle and the ideal Diesel cycle is the method of igniting the fuel-air mixture. Recall that in the ideal Diesel cycle the extremely high compression ratio (around 18:1) allows the air to reach the ignition temperature of the fuel. The fuel is then injected such that the ignition process occurs at a constant pressure. In the ideal Otto cycle the fuel-air mixture is introduced during the induction stroke and compressed to a much lower compression ratio (around 8:1) and is then ignited by a spark. The combustion results in a sudden jump in pressure while the volume remains essentially constant. The continuation of the cycle including the expansion and exhaust processes are essentially identical to that of the ideal Diesel cycle. We find it convenient to develop the analysis approach of the ideal Otto cycle through the following solved problem:

Solved Problem - An ideal air-standard Otto cycle engine has a compression ratio of 8. At the beginning of the compression process, the working fluid is at 100 kPa, 27°C (300 K), and 800 kJ/kg heat is supplied during the constant volume heat addition process. Neatly sketch the pressure-volume [P - v] diagram for this cycle, and using the specific heat values for air at a typical average cycle temperature of 900K determine:

- a) the temperature and pressure of the air at the end of each process
- b) the net work output/cycle [kJ/kg], and
- c) the thermal efficiency [η_{th}] of this engine cycle.

Solution Approach:

The first step is to draw the P - v diagram of the complete cycle, including all the relevant information. We notice that neither volume nor mass have been provided, hence the diagram and solution will be in terms of specific quantities.



We assume that the fuel-air mixture is represented by pure air. The relevant equations of state, internal energy and adiabatic process for air follow:

$$P v = R T, \quad R = 0.287 \text{ [kJ/kg K]}$$

$$\Delta u = C_v \Delta T$$

$$\left. \begin{array}{l} \text{adiabatic} \Rightarrow P v^k = \text{const} \\ \Rightarrow T v^{k-1} = \text{const} \end{array} \right\} \begin{array}{l} \text{use values of } C_v, k \text{ at} \\ \text{a typical average cycle} \\ \text{temperature} \end{array}$$

We recall from the previous section that the nominal specific heat capacity values used for air at 300K are $C_v = 0.717 \text{ kJ/kg.K}$, and $k = 1.4$. However they are all functions of temperature, and with the extremely high temperature range experienced in internal combustion engines one can obtain significant errors. In this problem we use a typical average cycle temperature of 900K taken from the table of **Specific Heat Capacities of Air**.

$$C_{v,900K} = 0.834 \left[\frac{\text{kJ}}{\text{kg K}} \right], \quad k_{900K} = 1.344$$

We now go through all four processes in order to determine the temperature and pressure at the end of each process, as well as the work done and heat transferred during each process.

① → ② - Adiabatic compression

$$T v^{k-1} = \text{const} \Rightarrow T_2 = T_1 \left(\frac{v_1}{v_2} \right)^{k-1} = T_1 r^{k-1}$$

$$\Rightarrow T_2 = 300(\text{K}) \cdot 8^{0.344} = 613 \text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \Rightarrow P_2 = P_1 \left(\frac{v_1}{v_2} \cdot \frac{T_2}{T_1} \right) = 100[\text{kPa}] \left(8 \cdot \frac{613}{300} \right)$$

$$P_2 = 1635 \text{ kPa}$$

adiabatic

$$q - w_{12} = \Delta u = C_v \Delta T = C_v (T_2 - T_1)$$

$$\Rightarrow w_{12} = C_v (T_1 - T_2) = 0.834 \left[\frac{\text{kJ}}{\text{kg K}} \right] (300 - 613) \text{K}$$

$$w_{12} = -261 [\text{kJ/kg}]$$

② → ③ Constant Volume Combustion

Heat in during combustion: $q_{23} = 800 [\text{kJ/kg}]$

$$q_{23} - w = \Delta u = C_v \Delta T = C_v (T_3 - T_2)$$

$$\Rightarrow 800 [\text{kJ/kg}] = 0.834 \left[\frac{\text{kJ}}{\text{kg K}} \right] (T_3 - 613) \text{K}$$

$$T_3 = 1572 \text{K}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \Rightarrow P_3 = P_2 \left(\frac{T_3}{T_2} \right) = 1635 [\text{kPa}] \left(\frac{1572}{613} \right)$$

$$P_3 = 4193 \text{ kPa}$$

③ → ④ - Adiabatic expansion

$$T v^{k-1} = \text{const} \Rightarrow T_4 = T_3 \left(\frac{v_3}{v_4} \right)^{k-1} = T_3 \left(\frac{1}{r} \right)^{k-1}$$

$$\Rightarrow T_4 = 1572 \left(\frac{1}{8} \right)^{0.344} = 769 \text{K}$$

$$\frac{P_4 v_4}{T_4} = \frac{P_3 v_3}{T_3} \Rightarrow P_4 = P_3 \left(\frac{v_3}{v_4} \cdot \frac{T_4}{T_3} \right) = 4193 [\text{kPa}] \left(\frac{1}{8} \cdot \frac{769}{1572} \right)$$

$$P_4 = 256 \text{ kPa}$$

adiabatic

$$q - w_{34} = \Delta u = C_v \Delta T = C_v (T_4 - T_3)$$

$$\Rightarrow w_{34} = C_v (T_3 - T_4) = 0.834 \left[\frac{\text{kJ}}{\text{kg K}} \right] (1572 - 769) \text{K}$$

$$w_{34} = 670 \text{ [kJ/kg]}$$

Note that the pressure P_4 (as well as P_2 above) could also be evaluated from the adiabatic process equation. We do so below as a validity check, however we find it more convenient to use the ideal gas equation of state wherever possible. Either method is satisfactory.

$$P v^k = \text{const} \Rightarrow P_4 = P_3 \left(\frac{v_3}{v_4} \right)^k = P_3 \left(\frac{1}{r} \right)^k$$

$$P_4 = 4193 [\text{kPa}] \left(\frac{1}{8} \right)^{1.344} = 256 \text{ kPa}$$

We continue with the final process to determine the heat rejected:

④ → ① - Constant Volume Exhaust

Heat rejected during exhaust process:

$$q_{41} - w = \Delta u = C_v \Delta T = C_v (T_1 - T_4)$$

$$q_{41} = 0.834 \left[\frac{\text{kJ}}{\text{kg K}} \right] (300 - 769) \text{K} = -391 \text{ kJ/kg}$$

Notice that we have applied the energy equation to all four processes allowing us two alternative means of evaluating the "net work output per cycle" and the thermal efficiency, as follows:

$$w_{\text{net}} = w_{12} + w_{34} = (-261 + 670) = 409 \text{ kJ/kg} \rightarrow$$

As a check, over a complete cycle: $w_{\text{net}} = q_{\text{net}}$

$$w_{\text{net}} = q_{\text{net}} = q_{23} + q_{41} = (800 - 391) = 409 \text{ kJ/kg}$$

Thermal Efficiency:

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{w_{\text{net}}}{q_{23}} = \frac{409}{800} \Rightarrow \eta_{\text{th}} = 51\% \rightarrow$$

Note that using constant specific heat values over the cycle we can determine the thermal efficiency directly from the ratio of specific heat capacities k with the following formula:

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{r^{k-1}}$$

where r is the compression ratio

Source: http://www.ohio.edu/mechanical/thermo/Intro/Chapt.1_6/Chapter3d.html