The First Law of Thermodynamics for Closed Systems Part III

The Air-Standard Diesel Cycle (Compression-Ignition) Engine

The Air Standard Diesel cycle is the ideal cycle for Compression-Ignition (CI) reciprocating engines, first proposed by Rudolph Diesel over 100 years ago. The following link by the Kruse Technology Partnership describes the four-stroke diesel cycle operation including a short history of Rudolf Diesel. The four-stroke diesel engine is usually used in motor vehicle systems, whereas larger marine systems usually use the two-stroke diesel cycle. Once again we have an excellent animation produced by Matt Keveney presenting the operation of the four-stroke diesel cycle.

The actual CI cycle is extremely complex, thus in initial analysis we use an ideal "air-standard" assumption, in which 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 ideal air-standard diesel engine undergoes 4 distinct processes, each one of which can be separately analysed, as shown in the *P-V* diagrams below. Two of the four processes of the cycle are adiabatic processes (adiabatic = no transfer of heat), thus before we can continue we need to develop equations for an ideal gas adiabatic process as follows:

The Adiabatic Process of an Ideal Gas (Q = 0)

The analysis results in the following three general forms representing an adiabatic process:

$$T\,v^{k-1} = \text{const} \qquad T\,P^{(1-k)/k} = \text{const} \qquad P\,v^{k} = \text{const}$$

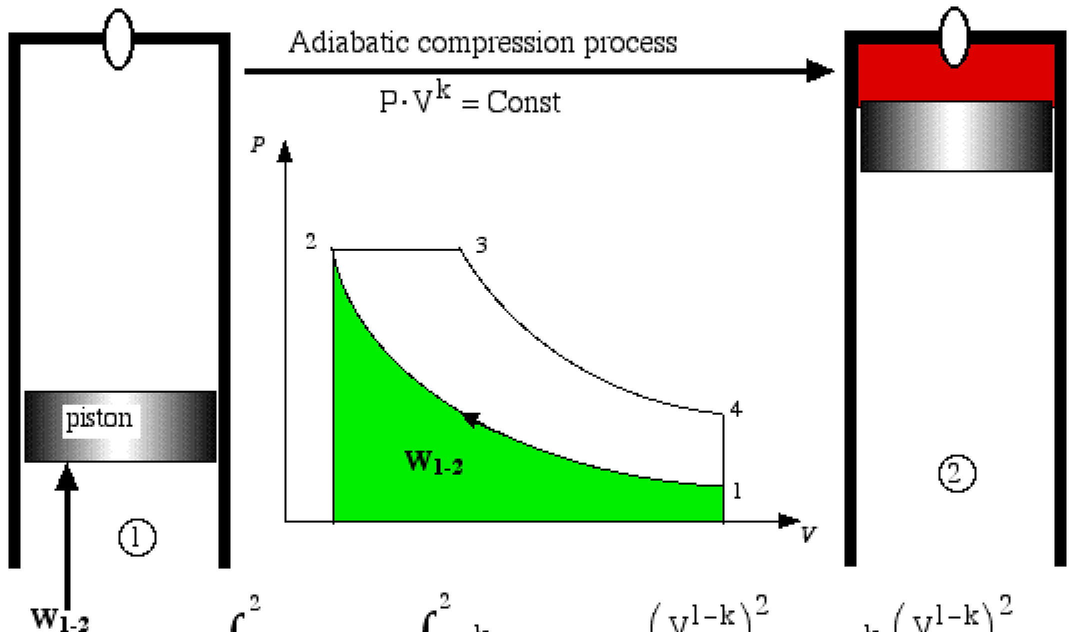
where k is the ratio of heat capacities and has a nominal value of 1.4 at 300K for air.

Process 1-2 is the adiabatic compression process. Thus the temperature of the air increases during the compression process, and with a large compression ratio (usually > 16:1) it will reach the ignition temperature of the injected fuel. Thus given the conditions at state 1 and the compression ratio of the engine, in order to determine the pressure and temperature at state 2 (at the end of the adiabatic compression process) we have:

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^k = r^k \qquad \left[r = \frac{V_1}{V_2} \Rightarrow \text{Compression ratio} \right]$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{V_1}{V_2}\right)^{k-1} = r^{k-1}$$

Work $W_{1\text{-}2}$ required to compress the gas is shown as the area under the *P-V* curve, and is evaluated as follows.



$$W_{1\text{-}2} = \int_1^2 P\,\mathbf{d}V = \text{Const}\int_1^2 V^{-k}\mathbf{d}V = \text{Const}\left(\frac{V^{1-k}}{1-k}\right)_1^2 = P\cdot V^k\left(\frac{V^{1-k}}{1-k}\right)_1^2$$

$$\text{thus:}\quad W_{1\text{-}2} = \left(\frac{P\cdot V}{1-k}\right)_1^2 = \left(\frac{P_2V_2 - P_1V_1}{(1-k)}\right) = \left(\frac{m\cdot R\cdot(T_2 - T_1)}{(1-k)}\right)$$

$$\text{since for an ideal gas:}\quad P\cdot V = m\cdot R\cdot T$$

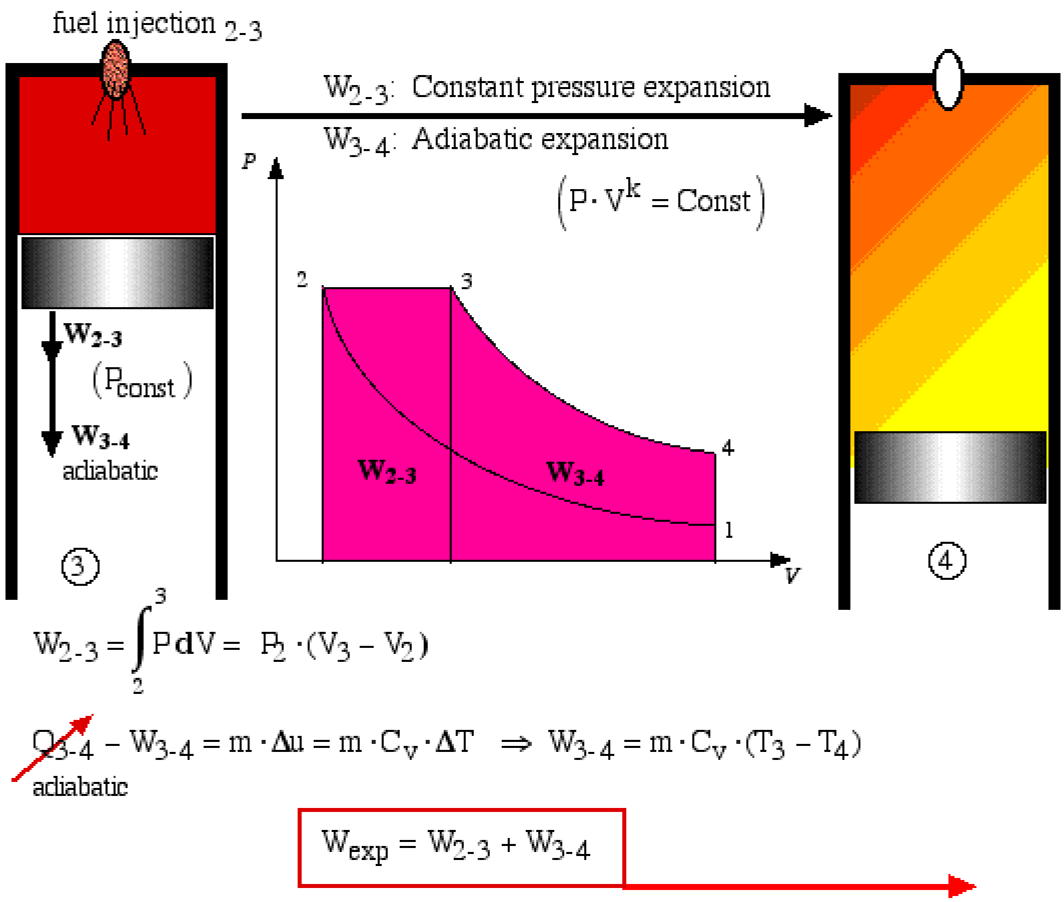
An alternative approach using the energy equation takes advantage of the adiabatic process ($Q_{1\text{-}2} = 0$) results in a much simpler process:

$$\overset{\text{adiabatic}}{\cancel{Q_{1\text{-}2}}} - W_{1\text{-}2} = m\cdot\Delta u = m\cdot C_v\cdot\Delta T \Rightarrow \boxed{W_{1\text{-}2} = m\cdot C_v\cdot(T_1 - T_2)} \longrightarrow$$

(thanks to student Nichole Blackmore for making me aware of this alternative approach)

During process 2-3 the fuel is injected and combusted and this is represented by a constant pressure expansion process. At state 3 ("fuel cutoff") the expansion process continues adiabatically with the temperature decreasing until the expansion is complete.

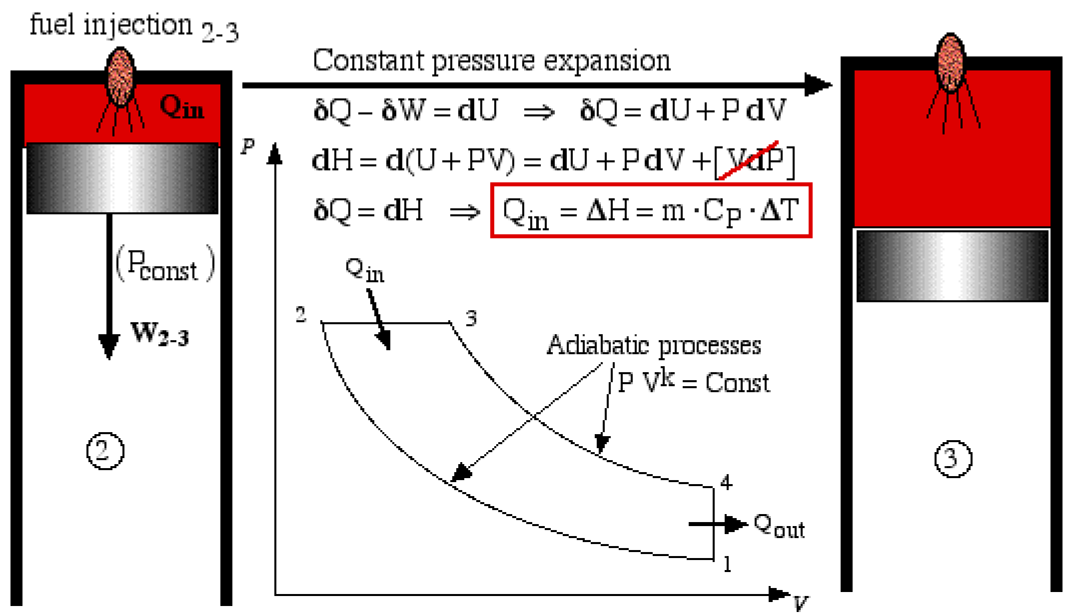
Process 3-4 is thus the adiabatic expansion process. The total expansion work is $W_{exp} = (W_{2-3} + W_{3-4})$ and is shown as the area under the *P-V* diagram and is analysed as follows:



Finally, process 4-1 represents the constant volume heat rejection process. In an actual Diesel engine the gas is simply exhausted from the cylinder and a fresh charge of air is introduced.

The net work W_{net} done over the cycle is given by: $W_{net} = (W_{exp} + W_{1-2})$, where as before the compression work W_{1-2} is negative (work done *on* the system).

In the Air-Standard Diesel cycle engine the heat input Q_{in} occurs by combusting the fuel which is injected in a controlled manner, ideally resulting in a constant pressure expansion process 2-3 as shown below. At maximum volume (bottom dead center) the burnt gasses are simply exhausted and replaced by a fresh charge of air. This is represented by the equivalent constant volume heat rejection process $Q_{out} = -Q_{4\text{-}1}$. Both processes are analyzed as follows:



$$Q_{in} = \Delta H = m \cdot C_p \cdot \Delta T = m \cdot C_p \cdot (T_3 - T_2)$$

$$P \cdot V = m \cdot R \cdot T \Rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} \quad \text{(constant pressure)}$$

$$Q_{in} = m \cdot C_p \cdot T_2 \cdot \left(\frac{V_3}{V_2} - 1\right) = m \cdot C_p \cdot T_2 \cdot (r_c - 1) \quad \text{where: } r_c = \frac{V_3}{V_2} \text{ (cutoff ratio)}$$

Constant volume heat rejection Q_{out}

$$Q_{out} = -Q_{4\text{-}1} = -\Delta U = -m \cdot C_v \cdot \Delta T = m \cdot C_v \cdot (T_4 - T_1)$$

At this stage we can conveniently determine the engine efficiency in terms of the heat flow as follows:

$$Q_{in} = m \cdot C_p \cdot (T_3 - T_2) \quad \text{(constant pressure)}$$

$$Q_{out} = m \cdot C_v \cdot (T_4 - T_1) \quad \text{(constant volume)}$$

Again from the First Law for a cycle:

$$W_{net} = W_{1\text{-}2} + W_{2\text{-}3} + W_{3\text{-}4} = Q_{in} - Q_{out}$$

Thus thermal efficiency: $\eta_{th} = \dfrac{W_{net}}{Q_{in}} = \left(1 - \dfrac{Q_{out}}{Q_{in}}\right)$

Source: http://www.ohio.edu/mechanical/thermo/Intro/Chapt.1_6/Chapter3c.html