Gas Power Cycles – Dual Cycle

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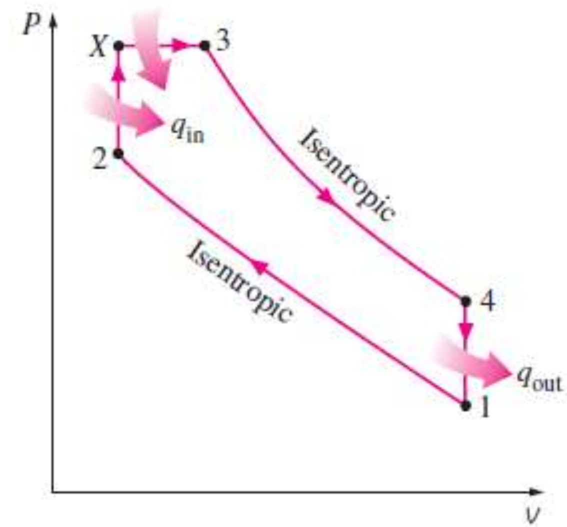
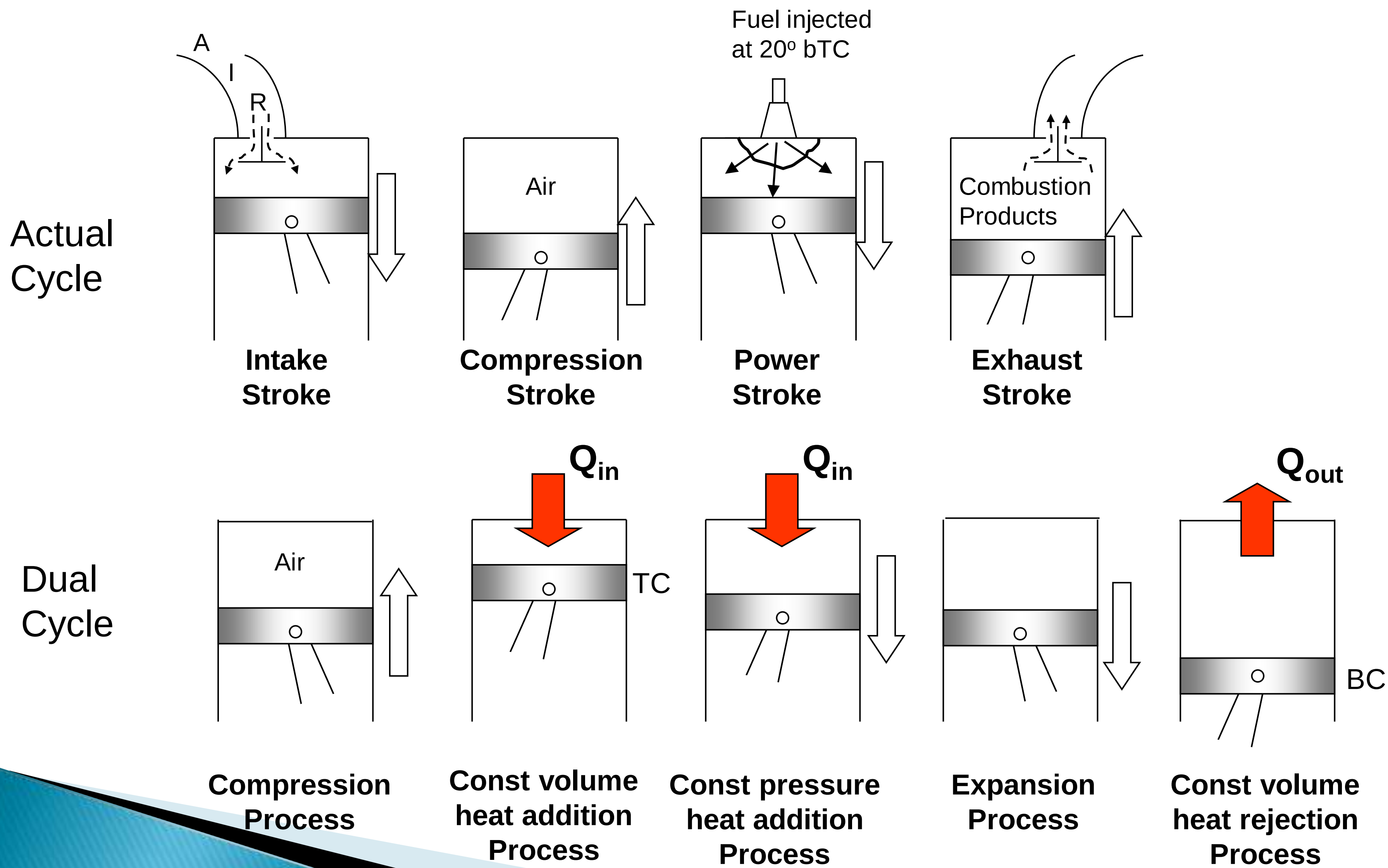
PV and TS diagram for Ideal Dual Cycle

- the combustion process in internal combustion engines as a constant-volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic.
- Probably a better (but slightly more complex) approach would be to model the combustion process in both gasoline and diesel engines as a combination of two heat-transfer processes, one at constant volume and the other at constant pressure. The ideal cycle based on this concept is called the **dual cycle**



Modern CI Engine Cycle vs Dual Cycle



Thermal Efficiency

$$\eta_{\substack{Dual\\cycle}} = 1 - \frac{Q_{out}/m}{Q_{in}/m} = 1 - \frac{u_4 - u_1}{(u_X - u_2) + (h_3 - h_X)}$$

For cold air-standard the above reduces to:

$$\eta_{\substack{Diesel\\const\ c_v}} = 1 - \frac{1}{r^{k-1}}\left[\frac{\alpha r_c^k - 1}{(\alpha - 1) + \alpha k (r_c - 1)}\right]$$

where $r_c = v_3/v_X$ and $\alpha = P_3/P_2$

Note, the Otto cycle (r_c =1) and the Diesel cycle (α=1) are special cases:

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

$$\eta_{\substack{Diesel\\const\ c_V}} = 1 - \frac{1}{r^{k-1}}\left[\frac{1}{k}\cdot\frac{(r_c^k - 1)}{(r_c - 1)}\right]$$

The use of the Dual cycle requires information about either the fractions of constant volume and constant pressure heat addition (common assumption is to equally split the heat addition), or the maximum pressure P_3.

Transformation of r_c and α into more natural variables yields

$$r_c = 1 - \frac{k-1}{\alpha k}\left[\left(\frac{Q_{in}}{P_1 V_1}\right)\frac{1}{r^{k-1}} - \frac{\alpha - 1}{k-1}\right] \qquad \alpha = \frac{1}{r^k}\frac{P_3}{P_1}$$

For the same inlet conditions P_1, V_1 and the same compression ratio:

$$\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$$

For the same inlet conditions P_1, V_1 and the same peak pressure (actual design limitation in engines):

$$\eta_{Diesel} > \eta_{Dual} > \eta_{otto}$$