Lecture 19

Topic 4
Power & Refrigeration Cycles

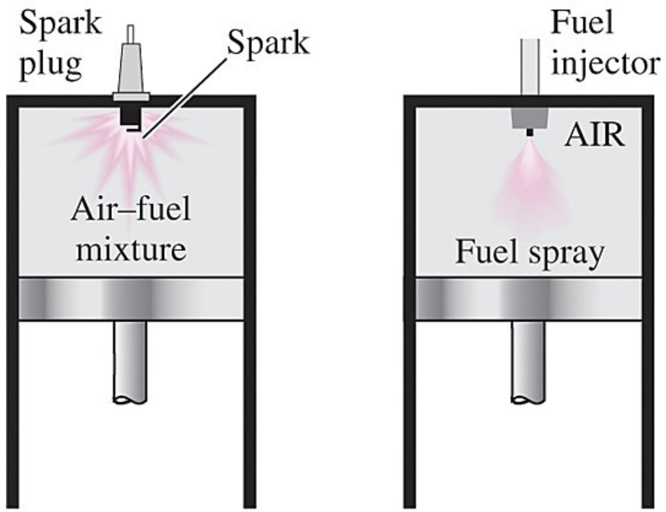
Topic

– 4.4 Diesel Cycle

Reading:
Ch 10: 10.7 + 10.9 Borgnakke & Sonntag Ed. 8
Ch 9: 9-6 Cengel and Boles Ed. 7

4.4.1 Otto vs. Diesel Cycle

- Diesel cycle: approximation for compression-ignition reciprocating engines.
- Otto vs. Diesel
  - Otto:
    - Fuel + air are compressed
    - Spark ignition initiates heat release
  - Diesel
    - Air is compressed
    - Fuel injected directly into cylinder
    - Air compressed to temp. above auto-ignition
    - Ignition occurs as auto-ignition



Gasoline engine   Diesel engine

- Compression ratio (r)
  - Otto (7-11)
  - Diesel (12-24)
- Diesel compresses gas to higher T, P to achieve auto-ignition.

4.4.1 Otto vs. Diesel Cycle

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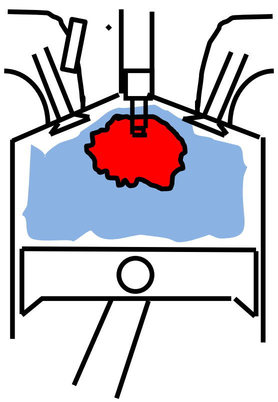
Otto Cycle

- Spark-ignition (gasoline)
- Heat release via flame propagation



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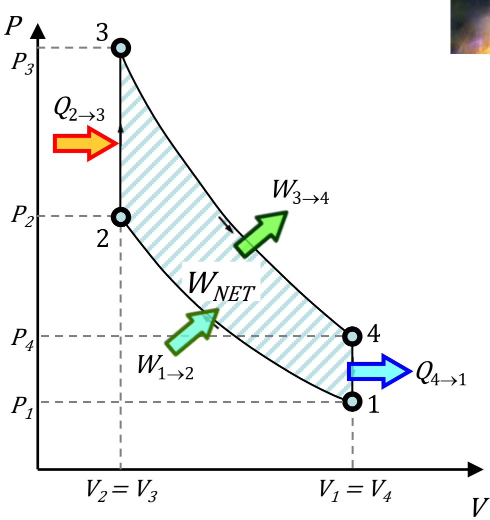
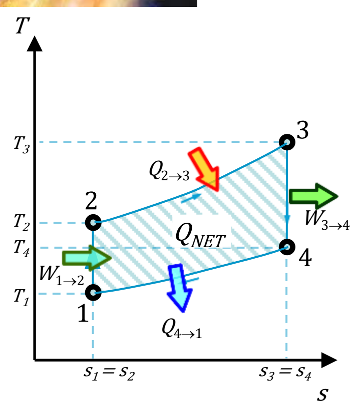
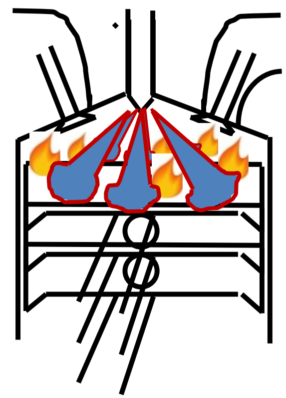
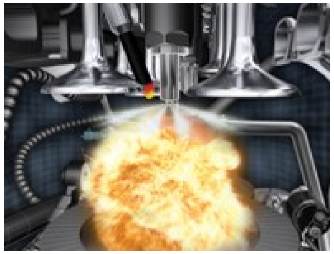




Diesel Cycle

- Heat release via auto-ignition, fuel injection





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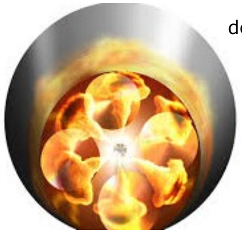
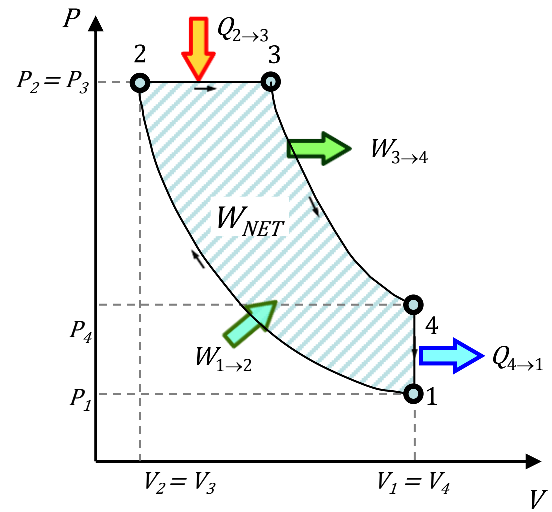
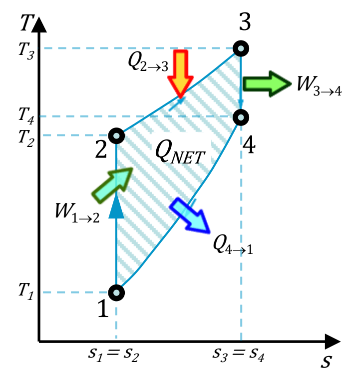


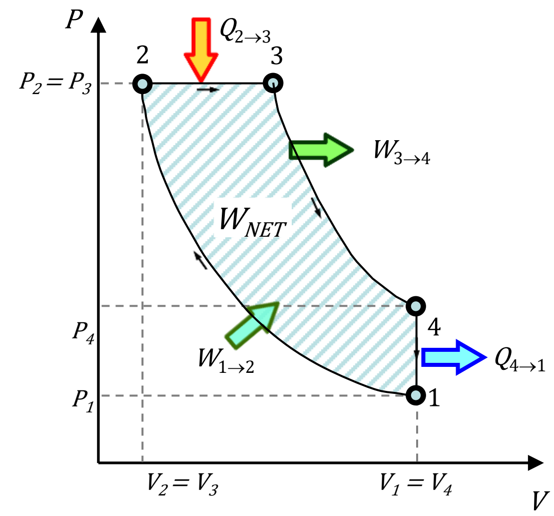




4.4.2 Diesel Cycle

The IDEAL air-standard Diesel cycle

Process	**Description**
1-2	Isentropic compression
2-3	Constant pressure heat addition
3-4	Isentropic expansion
4-1	Constant volume heat rejection

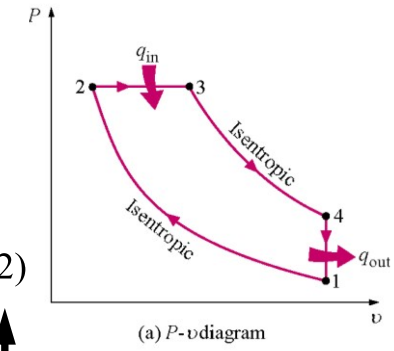


Process 2-3: only process where Diesel & Otto differ

- Otto: constant volume heat addition
- Diesel: constant pressure heat addition
  - Q_H added during the first part of power stroke (i.e. over a longer time interval)

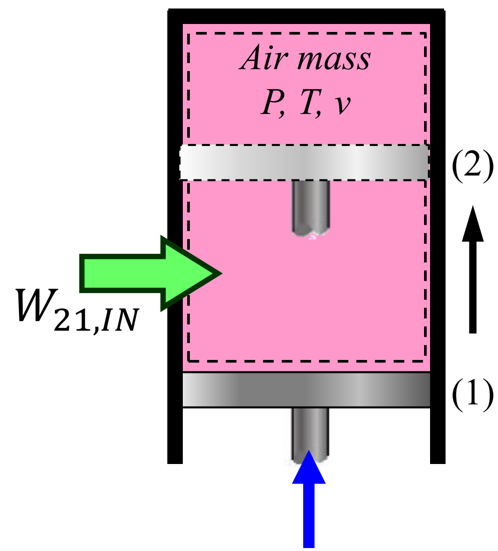
4.4.3 Ideal Diesel Cycle

- Air is treated as an ideal gas



Process 1-2: isentropic compression

- Adiabatic, reversible ($s_2 = s_1$)
- 1st Law: $\Delta U = Q - W$
  - $W_{21,IN} = m(u_2 - u_1)$
  - $W_{21,IN} = mC_V(T_2 - T_1)$



- 2nd Law analysis
  - $s_2 - s_1 = \int \frac{\delta d}{T} + s_{gen} \rightarrow s_2 - s_1 = 0$
  - Ideal gas: (lecture 12)
    - $s_2 - s_1 = \int_1^2 C_V \frac{dT}{T} + R ln\left(\frac{v_2}{v_1}\right)$ OR $s_2 - s_1 = \int_1^2 C_P \frac{dT}{T} - R ln\left(\frac{P_2}{P_1}\right)$
  - If constant specific heat, T, P, v relationship
    - $C_V ln\left(\frac{T_2}{T_1}\right) = R ln\left(\frac{v_1}{v_2}\right) \rightarrow \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$
    - $C_P ln\left(\frac{T_2}{T_1}\right) = R ln\left(\frac{P_2}{P_1}\right) \rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{k-1/k}$
  - If variable specific heat, T, P, v relationship
    - $s^o_{T2} = s^o_{T1} + R ln\left(\frac{P_2}{P_1}\right)$

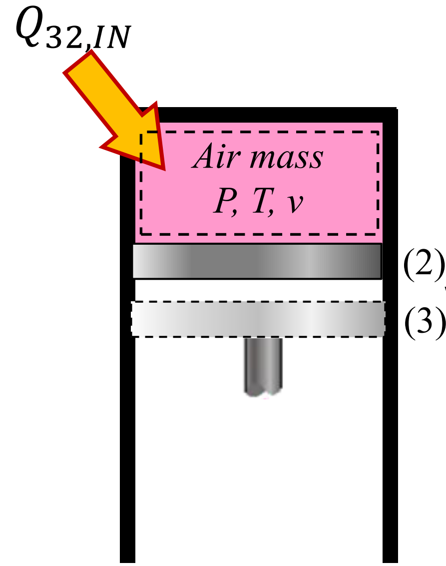
- Isentropic relations: $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{V_1}{V_2}\right)^{k-1}$ & $\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{k}$

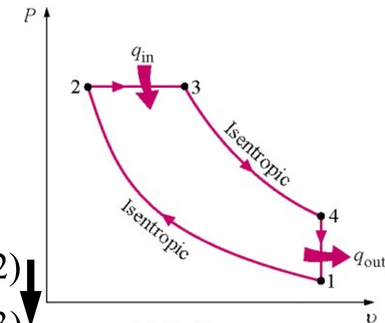
4.4.3 Ideal Diesel Cycle

- Air is treated as an <u>ideal gas</u>

Process 2-3: constant pressure heat addition

- Heat addition → $P_3 = P_2$, $T_3 > T_2$.
- 1st Law: $\Delta U = Q - W$
  - Boundary work: $W_{32} = P_{2,3}(V_3 - V_2)$
  - $Q_{32,IN} = m(u_3 - u_2) + W_{32}$
  - $Q_{32,IN} = m(u_3 - u_2) + P_{2,3}(V_3 - V_2)$
  - $Q_{32,IN} = m\left[\underbrace{(u_3 + P_3 v_3)}_{h_3} - \underbrace{(u_2 + P_2 v_2)}_{h_2}\right]$
  - $Q_{32,IN} = m(h_3 - h_2)$
  - $Q_{32,IN} = mC_P(T_3 - T_2)$





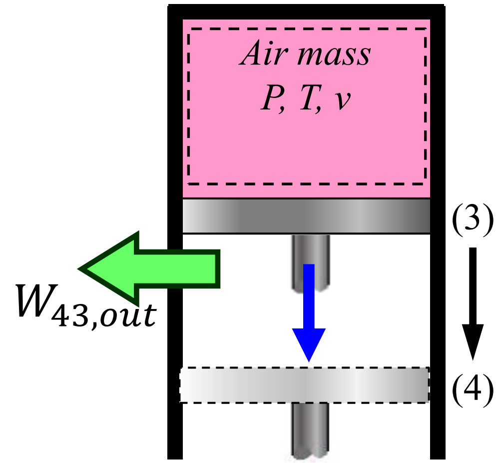
(a) P-υdiagram

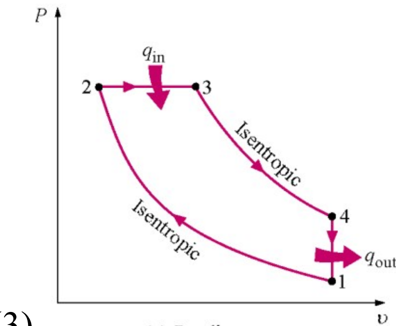
4.4.3 Ideal Diesel Cycle

- Air is treated as an ideal gas

Process 3-4: isentropic expansion

- Adiabatic, reversible ($s_4 = s_3$)
- 1st Law: $\Delta U = Q - W$
  - $W_{43,out} = m(u_3 - u_4)$
  - $W_{43,out} = mC_V(T_3 - T_4)$





(a) P-v diagram

- 2nd Law analysis
  - $s_4 - s_2 = \int \frac{\delta d}{T} + s_{gen} \rightarrow s_4 - s_3 = 0$
  - Ideal gas: (lecture 12)
    - $s_4 - s_3 = \int_3^4 C_V \frac{dT}{T} + Rln\left(\frac{v_4}{v_3}\right)$ OR $s_4 - s_3 = \int_3^4 C_P \frac{dT}{T} - Rln\left(\frac{P_4}{P_3}\right)$
  - If constant specific heat, T, P, v relationship
    - $C_V ln\left(\frac{T_4}{T_3}\right) = Rln\left(\frac{v_3}{v_4}\right) \rightarrow \frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$
    - $C_P ln\left(\frac{T_4}{T_3}\right) = Rln\left(\frac{P_4}{P_3}\right) \rightarrow \frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{k-1/k}$
  - If variable specific heat, T, P, v relationship
    - $s^o_{T4} = s^o_{T2} + Rln\left(\frac{P_4}{P_3}\right)$

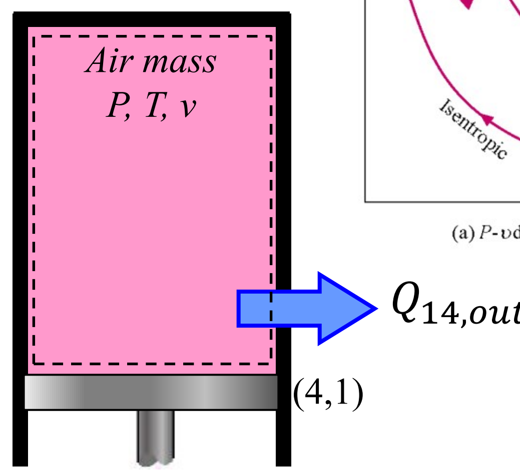
- Isentropic relations: $\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{(k-1)/k} = \left(\frac{V_3}{V_4}\right)^{k-1}$ & $\frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^{k}$

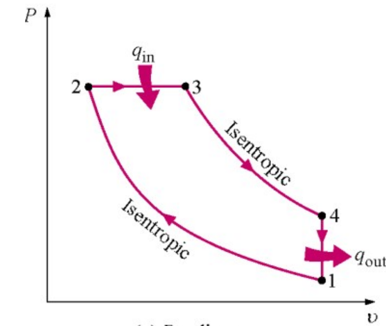
4.4.3 Ideal Diesel Cycle

- Air is treated as an ideal gas

Process 4-1: constant volume heat rejection

- Heat rejection → $P_1 < P_4$, $T_1 < T_4$
- 1st Law: $\Delta U = Q - W$
  - $Q_{14,out} = m(u_4 - u_1)$
  - $Q_{14,out} = mC_V(T_4 - T_1)$





(a) P-υ diagram

4.4.4 Diesel Cycle – Example

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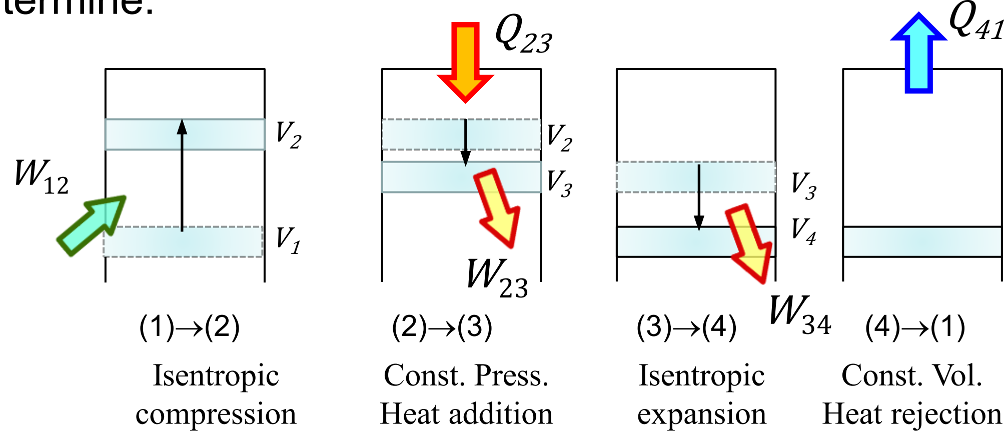
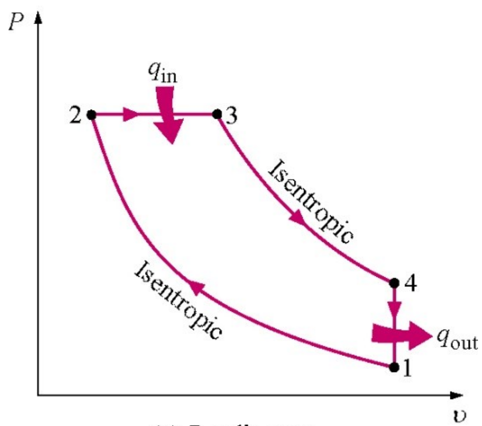
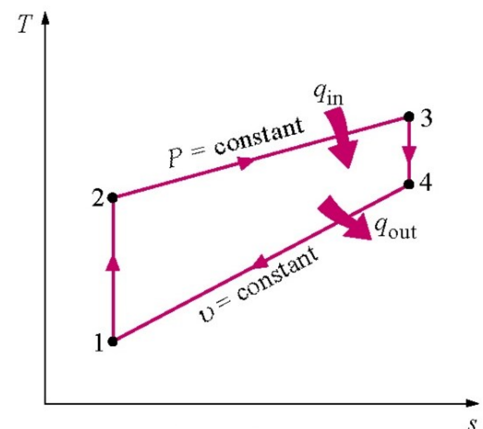
Example 4-6: An *ideal* Diesel cycle operates with a compression ratio (r) of 20 and a mass of 1kg. At the beginning of the compression stroke, air exists at 100 kPa, 15^{o}C. The heat addition from combustion (Q_{23}) is 880 kJ (same as Otto cycle Ex. 4-5). Assume are can be treated as an ideal gas with constant specific heats. Determine:

a) The maximum working pressure
b) The maximum temperature
c) The heat rejected
d) The MEP and net thermal efficiency





(a) P-υdiagram

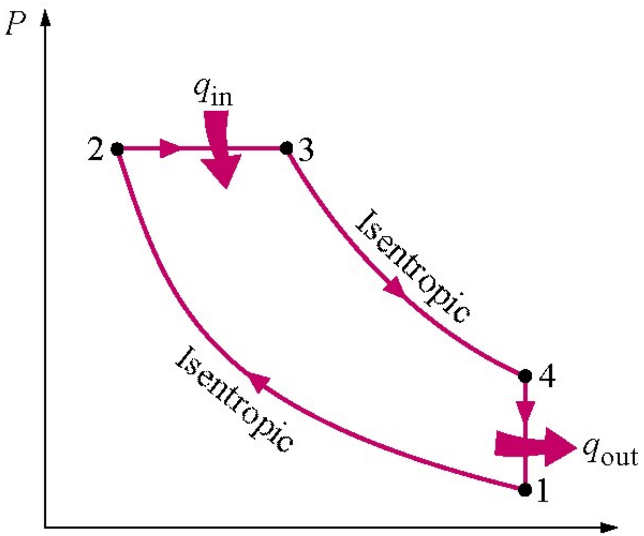


(b) T-s diagram

Air properties: R = 0.287 kJ/kgK, C_V = 0.717kJ/kgK, C_P = 1.004 kJ/kgK

4.4.4 Diesel Cycle – Thermal Efficiency

- Net thermal efficiency of an ideal diesel cycle can be expressed as a function of the temperatures in the cycle.



- $\eta_{th,Diesel} = \frac{W_{net}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$
- $\eta_{th,Diesel} = 1 - \frac{mC_v(T_4 - T_1)}{mC_P(T_3 - T_2)}$
- $\eta_{th,Diesel} = 1 - \frac{1}{k}\frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$

4.4.4 Diesel Cycle – Thermal Efficiency

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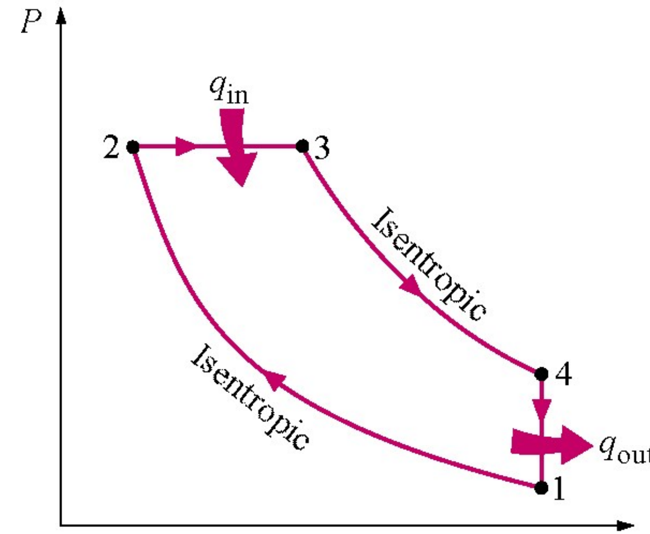
- Define cutoff ratio (r_c): $\boldsymbol{V_3 / V_2}$
  - Measure of the volume duration of the heat addition at constant pressure

Ideal gas behavior:

- $\frac{P_2 V_2}{mRT_2} = \frac{P_3 V_3}{mRT_3}$
  - $P_2 = P_3 \rightarrow \cdots \rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$
- $\frac{P_4 V_4}{mRT_4} = \frac{P_1 V_1}{mRT_1}$
  - $V_4 = V_1 \rightarrow \cdots \rightarrow \frac{T_4}{T_1} = \frac{P_4}{P_1}$
- Processes 1-2 and 3-4 are isentropic
  - $P_1 V_1^k = P_2 V_2^k \quad \& \quad P_3 V_3^k = P_4 V_4^k$
  - $\rightarrow \cdots \rightarrow \frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^k = r_c^k$



$$\eta_{th,Diesel} = 1 - \frac{1}{k}\frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)} = 1 - \frac{1}{k}\frac{T_1}{T_2}\frac{r_c^k - 1}{(r_c - 1)} = 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)}$$

4.4.4 Diesel Cycle – Thermal Efficiency

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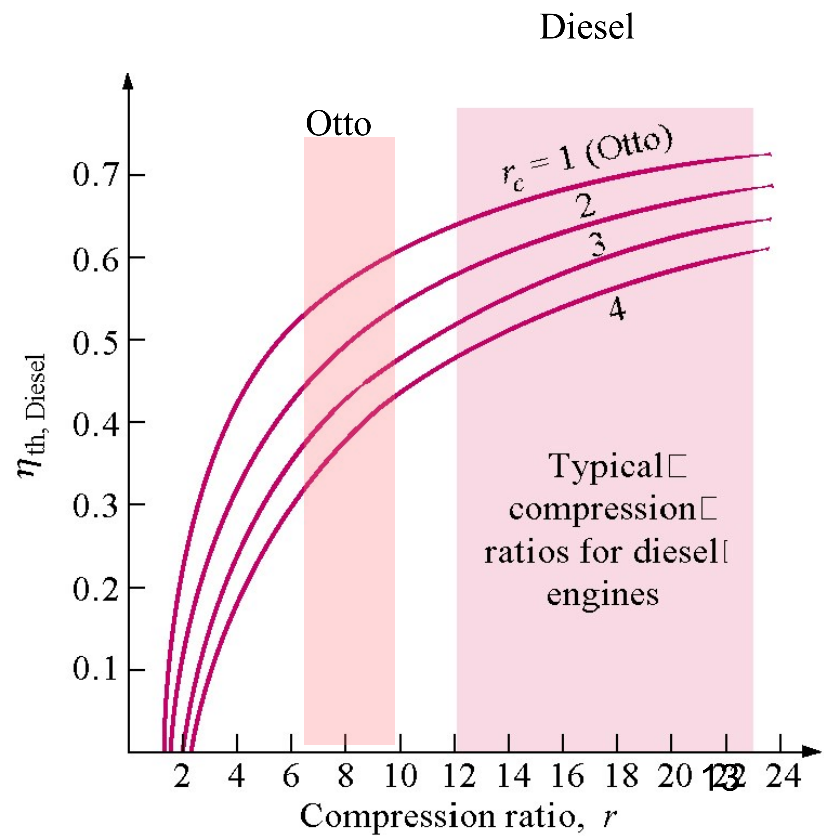
$$\eta_{th,Diesel} = 1 - \frac{1}{k}\frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)} = 1 - \frac{1}{k}\frac{T_1}{T_2}\frac{r_c^k - 1}{(r_c - 1)} = 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)}$$

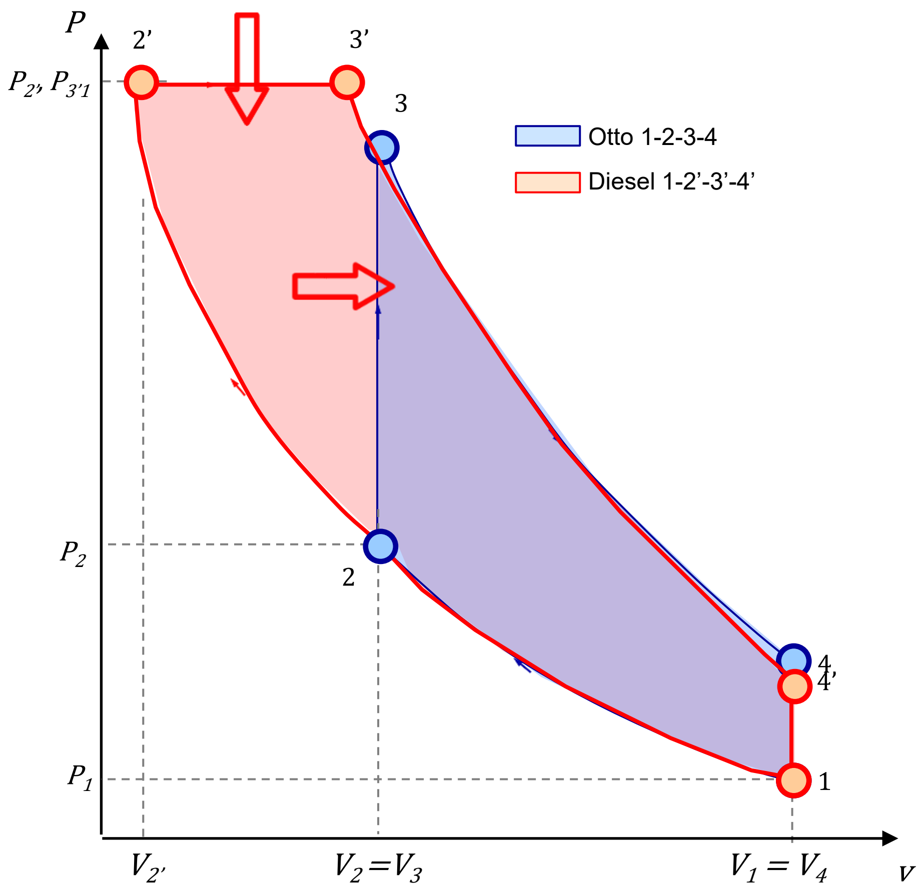
- As $r_c \rightarrow 1$ (i.e. towards $V_3 = V_2$)
  - $\eta_{th,Diesel} = \eta_{th,Otto}$

- Diesel engines operate at much higher compression ratios and thus are more efficient than the spark-ignition engines.



4.4.5 Diesel vs. Otto Thermal Efficiency

- Otto vs. Diesel: (Ex. 4-5 vs. Ex. 4-6)

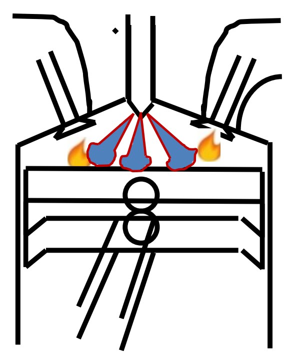


P_1 = 100 kPa, T_1 = 288K, Q_{23} = 880 kJ

	Otto	Diesel
Comp. Ratio	9	20
P_{max}	6000 kPa	6631 kPa
T_{max}	1921 K	1831 K
W_{12}	-291 kJ	-478.2 kJ
W_{exp}	746 kJ	1051 kJ
Q_{41}	-365 kJ	-307.5 kJ
n_{th}	52%	65%
IMEP	620 kPa	728 kPa

4.4.5 Diesel vs. Otto Thermal Efficiency

- Operating conditions typically much different between Otto and Diesel
- Compare cycles based on fixed operating parameters
- Fix state (1): T, P, v



4.4.5 Diesel vs. Otto Thermal Efficiency

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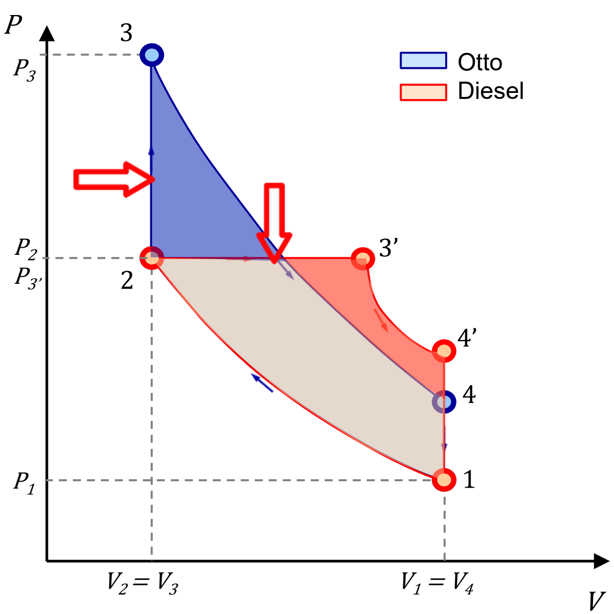
- Operating conditions typically much different between Otto and Diesel
- Compare cycles based on fixed operating parameters
- Fix state (1): T, P, v

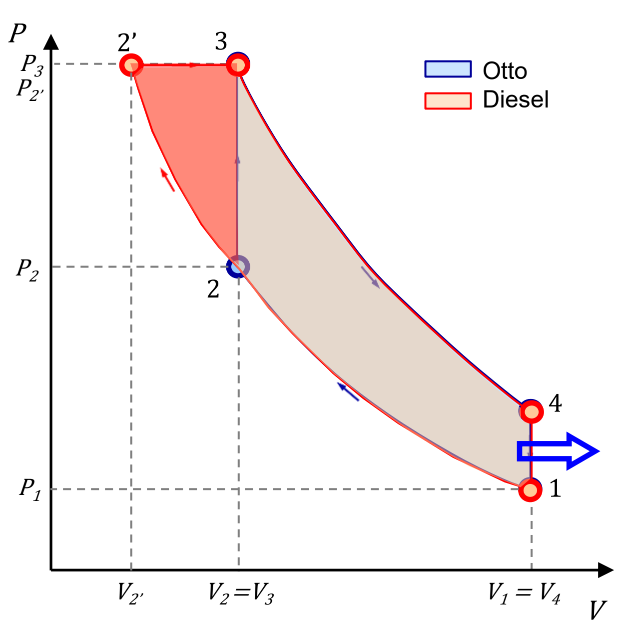
Fix: r & Q_H

- $P_{MAX,\,Otto} > P_{MAX,\,Diesel}$
- $W_{NET,\,Otto} > W_{NET,\,Diesel}$
- $\eta_{Otto} > \eta_{Diesel}$
- Diesel needs to operate with greater C_R



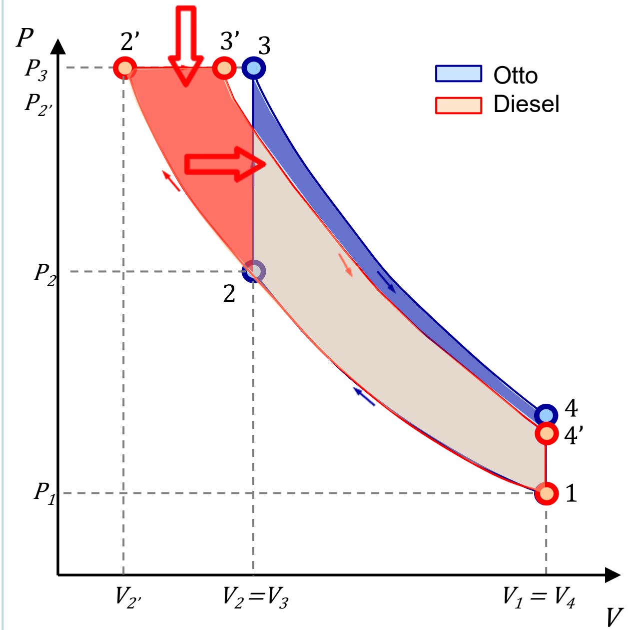
Fix: P_{MAX}, Q_L

- Comparison under similar thermal stresses
- $r_{Diesel} > r_{Otto}$
- $W_{NET,\,Diesel} > W_{NET,\,Otto}$
- $\eta_{Diesel} > \eta_{Otto}$



Fix: P_{MAX} & Q_H

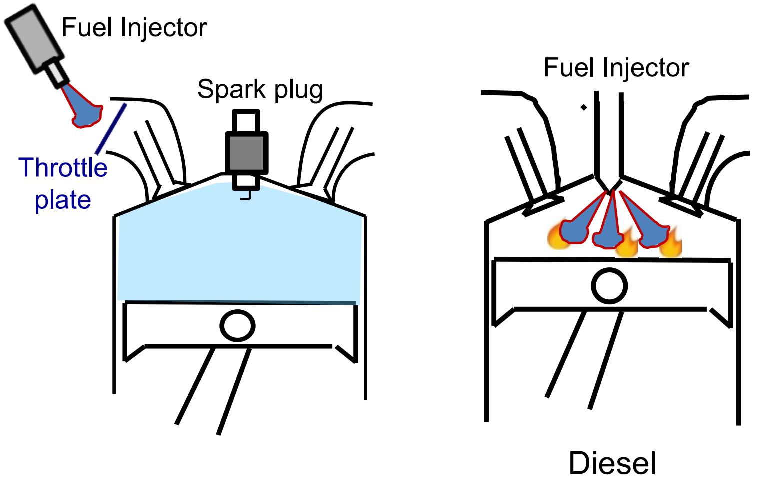
- Fix heat addition and r to reach P_{MAX}
- $r_{Diesel} > r_{Otto}$
- $W_{NET,\,Diesel} > W_{NET,\,Otto}$
- $\eta_{Diesel} > \eta_{Otto}$



4.4.5 Diesel vs. Otto Thermal Efficiency

- How can Otto approach the efficiency of diesel?
  - Therory: $\eta_{Diesel} = 0.6 - 0.7$; $\eta_{Otto} = 0.5 - 0.6$
  - Realistic: $\eta_{Diesel} \cong 0.45 - 0.50$; $\eta_{Otto} \cong 0.35 - 0.40$

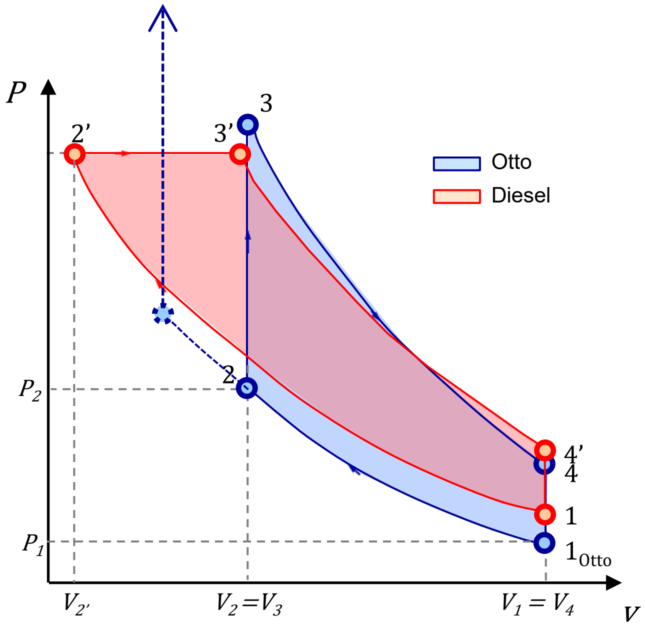
- Why not just drive diesels?
  - Emissions
  - Clean diesels
    - Current research topic

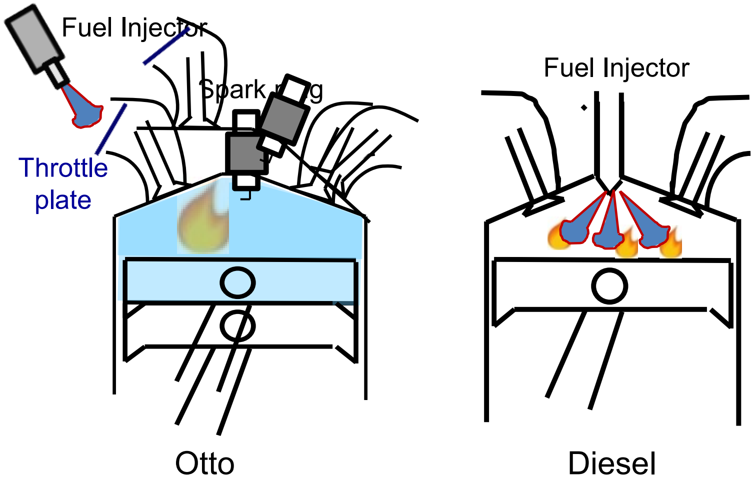


4.4.5 Diesel vs. Otto Thermal Efficiency

- How can Otto approach the efficiency of diesel?
  - Therory: $\eta_{Diesel} = 0.6 - 0.7$; $\eta_{Otto} = 0.5 - 0.6$

- Compression ratio
  - Higher *r* increases efficiency

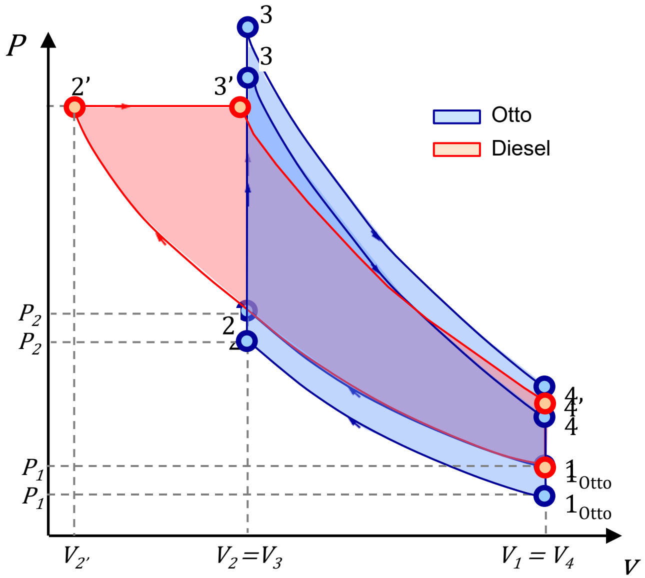


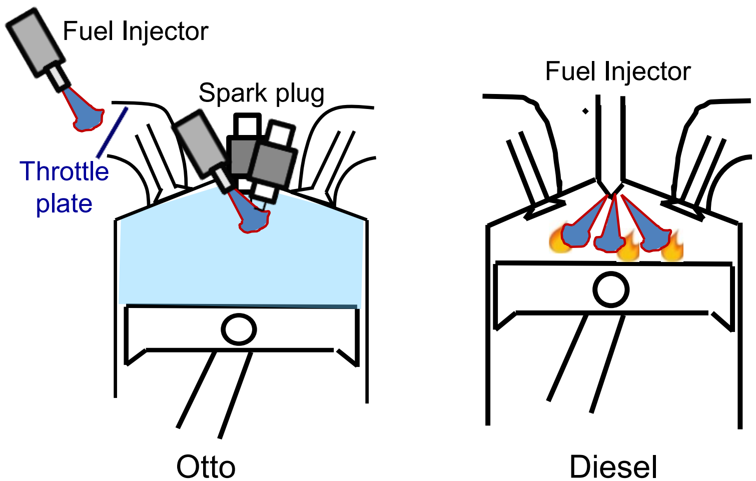


4.4.5 Diesel vs. Otto Thermal Efficiency

- How can Otto approach the efficiency of diesel?
  - Therory: $\eta_{Diesel} = 0.6 - 0.7$; $\eta_{Otto} = 0.5 - 0.6$
  - Realistic: $\eta_{Diesel} \cong 0.45$; $\eta_{Otto} \cong 0.3$

- Remove Throttle Plate
  - Reduce pumping losses
    - See open cycle analysis

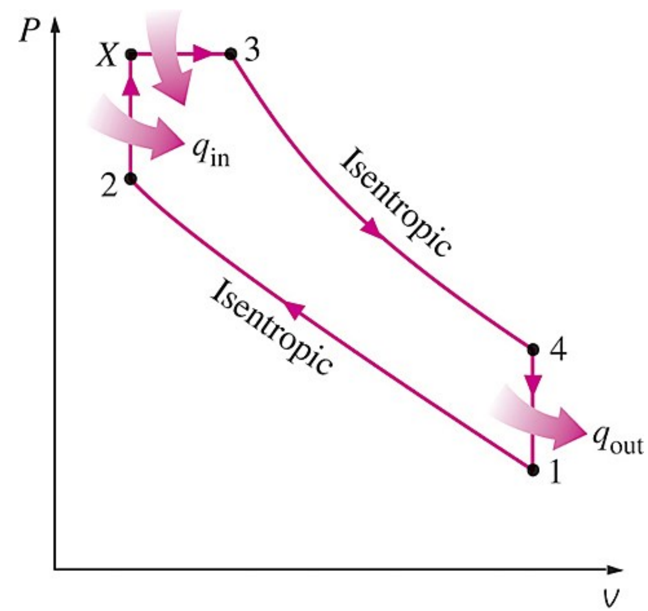




- Direct-injection spark-ignition
- Next generation SI engines
- Current research topic
- Ignition stability concerns

4.4.6 Dual Cycle

- **Dual Cycle**
  Approximating the combustion process in internal combustion engines as constant-volume (Otto cycle) or constant-pressure (Diesel cycle) 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 petrol 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.



4.4.6 Dual Cycle

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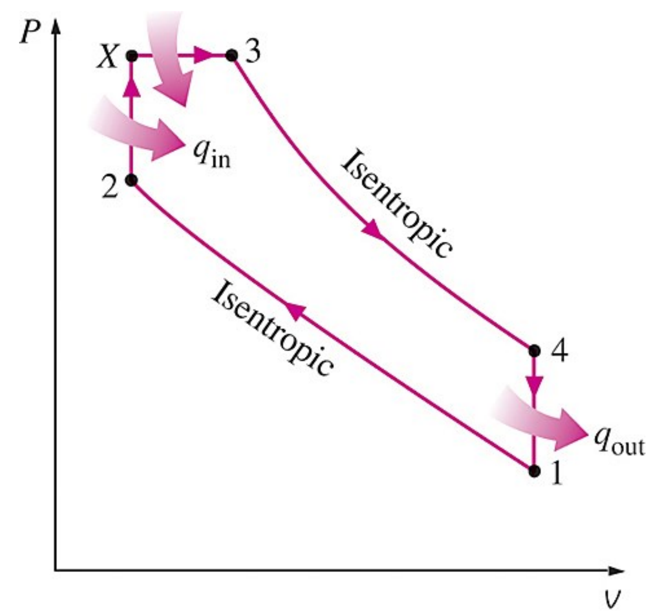
Exercise 4-7: The performance of a given diesel cycle is approximated by the dual cycle. The compression ratio (r) is 16 and the total heat added to the working fluid is 1935 kJ/kg. The inlet conditions are P_1 = 150 kPa and T_1 = 320 K. Assume air as an ideal gas is the working fluid with k = 1.4 and R = 0.287 kJ/kgK.

a) Assuming that half of the total heat is added at constant volume and half at constant pressure, compute the thermal efficiency

b) For the assumptions in (a) calculate the cut-off ratio

c) Compare the efficiency and peak pressure of the dual cycle with the efficiency and peak pressure that would be obtained if the same total heat were added at constant volume or at constant pressure



ans:

a) η_{th} = 0.55

b) 1.45

c) Dual cycle: Pmax = 13098 kPa, n_{th} = 0.55, Const. Vol: P_{max} = 20683 kPa, η_{th} = 0.565, Const. Press: P_{max} = 5514 kPa, η_{th} = 0.465