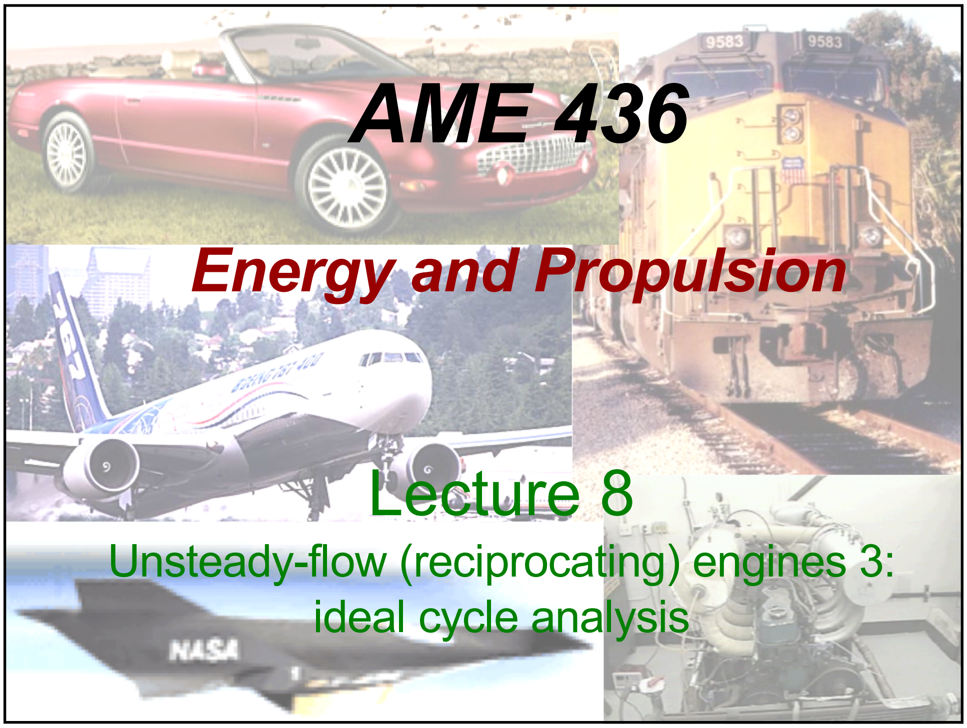

AME 436

Energy and Propulsion

Lecture 8

Unsteady-flow (reciprocating) engines 3:
ideal cycle analysis

Outline

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- Common cycle types
- Otto cycle
  - Why use it to model premixed-charge unsteady-flow engines?
  - Air-cycle processes
  - P-V & T-s diagrams
  - Analysis
  - Throttling and turbocharging/supercharging
- Diesel cycle
  - Why use it to model nonpremixed-charge unsteady-flow engines?
  - P-V & T-s diagrams
  - Air-cycle analysis
  - Comparison to Otto
- Complete expansion cycle
- Otto vs. Diesel - Ronney's Catechism
- Fuel-air cycles & comparison to air cycles & "reality"

Common cycles for IC engines

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- No real cycle behaves exactly like one of the ideal cycles, but for simple cycle analysis we need to hold one property constant during each process in the cycle

Process → ↓ Cycle Name↓	Comp-ression	Heat addition	Expan-sion	Heat rejection	Model for
Otto	s	v	s	v	Premixed-charge unsteady-flow engine
Diesel	s	P	s	v	Nonpremixed-charge unsteady-flow engine
Brayton	s	P	s	P	Steady-flow gas turbine
Complete expansion	s	v	s	P	"Late intake valve closing" premixed-charge engine
Stirling	v	T	v	T	"Stirling" engine
Carnot	s	T	s	T	Ideal reversible engine

Why use Otto cycle to model premixed-charge engines?

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- Volume compression ratio (r) = volume expansion ratio as reciprocating piston/cylinder arrangement provides
- Heat input at constant volume corresponds to infinitely fast combustion - not exactly true for real cycle, but for premixed-charge engine, burning time is a small fraction of total cycle time
- As always, constant entropy (s) compression/expansion corresponds to an adiabatic and reversible process - not exactly true but not bad either
- Recall that V on P-V diagram is cylinder volume (m^3), a property of the cylinder, NOT specific volume (v, units m^3/kg), a property of the gas
- Note that s is specific entropy (J/kgK) which IS a property of the gas, heat transfer = $\int$ Tds if mass doesn't change during heat addition

Ideal 4-stroke Otto cycle process

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- Compression ratio $r = V_2/V_1 = V_2/V_3 = V_5/V_4 = V_6/V_7$

Stroke	Process	Name	Constant	Mass in cylinder	Other info
A	$1 \rightarrow 2$	Intake	P	Increases	$P_2 = P_1$; $T_2 = T_1$ At 1, exhaust valve closes, intake valve opens
B	$2 \rightarrow 3$	Compression	s	Constant	$P_3/P_2 = r^{\gamma}$; $T_3/T_2 = r^{(\gamma-1)}$ At 2, intake valve closes
---	$3 \rightarrow 4$	Combustion	V	Constant	$T_4 = T_3 + fQ_R/C_v$; $P_4/P_3 = T_4/T_3$ At 3, spark fires
C	$4 \rightarrow 5$	Expansion	s	Constant	$P_4/P_5 = r^{\gamma}$; $T_4/T_5 = r^{(\gamma-1)}$
---	$5 \rightarrow 6$	Blowdown	V	Decreases	$P_6 = P_{ambient}$; $T_6/T_5 = (P_6/P_5)^{(\gamma-1)/\gamma}$ At 5, exhaust valve opens, exhaust gas "blows down"; gas remaining in cylinder experiences ≈ isentropic expansion
D	$6 \rightarrow 7$	Exhaust	P	Decreases	$P_7 = P_6$; $T_7 = T_6$

P-V & T-s diagrams for ideal Otto cycle

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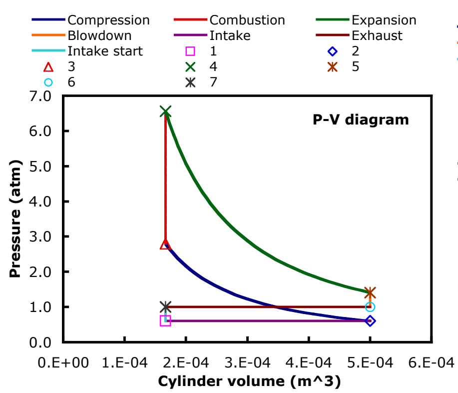
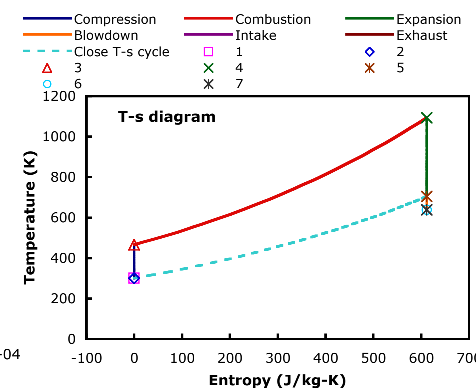
- Model shown is open cycle, where mixture is inhaled, compressed, burned, expanded then thrown away (not recycled)
- In a closed cycle with a fixed (trapped) mass of gas to which heat is transferred to/from, $6 \rightarrow 7$, $7 \rightarrow 1$, $1 \rightarrow 2$ would not exist, process would go directly $5 \rightarrow 2$ (Why don't we do this? Remember heat transfer is too slow!)





Otto cycle analysis

- Thermal efficiency (ideal cycle, no throttling or friction loss)

$$\eta_{th} = \frac{\text{what you get}}{\text{what you pay for}} = \frac{\text{work out} + \text{work in}}{\text{heat in}} = \frac{C_v(T_4 - T_5) + C_v(T_2 - T_3)}{C_v(T_4 - T_3)}$$

$$= \frac{T_4 - T_5 - T_3 + T_2}{T_4 - T_3} = \frac{T_4(1 - T_5/T_4) - T_3(1 - T_2/T_3)}{T_4 - T_3}$$

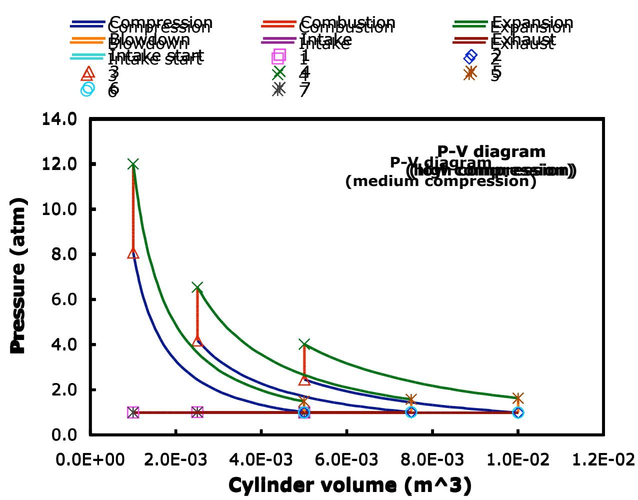
$$= \frac{T_4(1-(V_5/V_4)^{-(\gamma-1)}) - T_3(1-(V_2/V_3)^{-(\gamma-1)})}{T_4 - T_3} = \frac{T_4(1-r^{-(\gamma-1)}) - T_3(1-r^{-(\gamma-1)})}{T_4 - T_3}$$

$$= \frac{(T_4 - T_3)(1-r^{-(\gamma-1)})}{T_4 - T_3} = 1 - 1/r^{\gamma-1}$$

- Note η_{th} is independent of heat input (but in real cycle if mixture is too lean (too little heat input) it won't burn, if rich some fuel can't be burned since not enough O_2)
- Note that this η_{th} could have been determined by inspection of the T - s diagram - each Carnot cycle strip has same $1 - T_L/T_H = 1 - (T_2/T_3) = 1 - (V_3/V_2)^{\gamma-1} = 1 - (1/r)^{\gamma-1}$

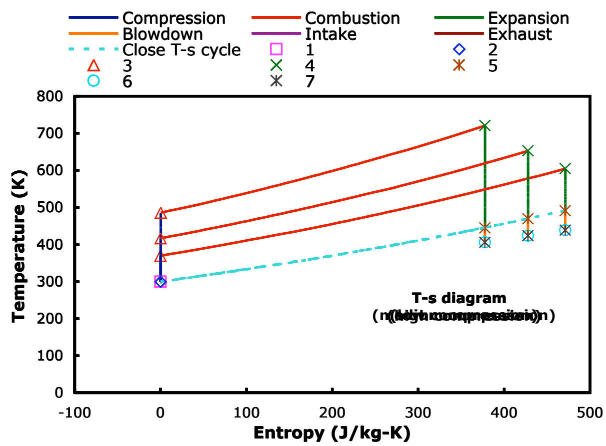
Effect of compression ratio (Otto)

- Animation: P-V diagrams, increasing compression ratio (same displacement volume, same fuel mass fraction (f), thus same heat input)



Effect of compression ratio (Otto)

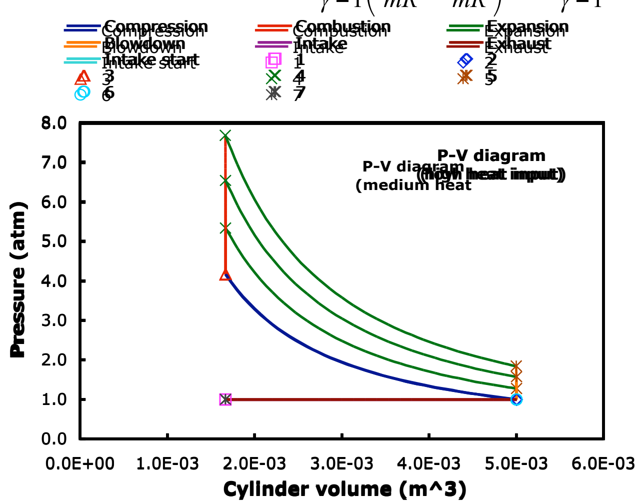
- Animation: T-s diagrams, increasing compression ratio (same displacement volume, same fuel mass fraction (f), thus same heat input)
- Higher compression clearly more efficient (taller Carnot strips)



Effect of heat input (Otto)

- Animation: P-V diagrams, increasing heat input via increasing f (same displacement volume, same compression ratio)

$$\text{Heat in} = mC_v\left(T_4 - T_3\right) = m\frac{R}{\gamma - 1}\left(\frac{P_4V_4}{mR} - \frac{P_3V_3}{mR}\right) = \frac{\left(P_4 - P_3\right)V}{\gamma - 1}$$



Effect of heat input (Otto)

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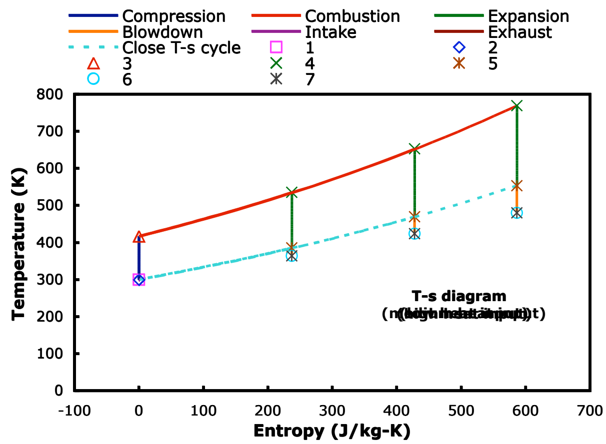
- Animation: T-s diagrams, increasing heat input via increasing f (same displacement volume, same compression ratio)
- Heat input does not affect efficiency (same T_L/T_H in Carnot strips)



Otto cycle analysis

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- η_{th} increases as r increases - why not use $r \to \infty$ ($\eta_{th} \to 1$)?
  - Main reason: KNOCK (lecture 10) - limits r to ≈ 10 depending on octane number of fuel
  - Also - heat losses increase as r increases (but this matters mostly for higher compression ratios as in Diesels discussed later)
- Typical premixed-charge engine with r = 8, γ = 1.3, theoretical η_{th} = 0.46; real engine ≈ 0.30 or less - why so different?
  - Heat losses - to cylinder walls, valves, piston
  - Friction
  - Throttling
  - Slow burn - combustion occurs over a finite time, thus a finite change in volume, not all at minimum volume (thus maximum T); as shown later this reduces η_{th}
  - Gas leakage past piston rings ("blow-by") & valves (minor issue)
  - Incomplete combustion (minor issue)

Throttling losses

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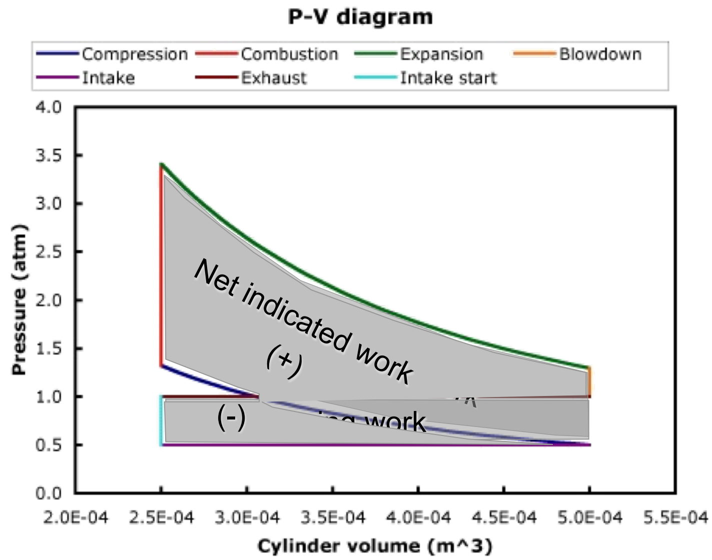
- Animation: gross & net indicated work, pumping work



Throttling losses

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- When you need less than the maximum IMEP available from a premixed-charge engine at Wide Open Throttle (WOT) (which is most of the time), a throttle is used to control IMEP, thus torque & power
- Throttling adjusts torque output by reducing intake ρ through decrease in intake P; when throttled, mass flow (thus volumetric efficiency η_v) decreases with $\eta_v \approx \eta_{v,WOT}(P_{intake}/P_{ambient})$; recall (Lecture 7)

$$\frac{IMEP_g}{P_{ambient}} = \frac{\eta_{th,i,g}\eta_v f Q_R}{RT_{\text{ambient}}} \Rightarrow IMEP_g = \frac{P_{ambient}}{RT_{ambient}}\eta_{th,i,g}\eta_v f Q_R$$

$$\Rightarrow IMEP_g = \frac{P_{ambient}}{RT_{ambient}}\eta_{th,i,g}\eta_{v,WOT}\frac{P_{\text{intake}}}{P_{ambient}} f Q_R = \left[\frac{\eta_{th,i,g}\eta_{v,WOT} f Q_R}{RT_{ambient}}\right] P_{\text{intake}} = K \cdot P_{\text{intake}}$$

where K ≈ constant (not a function of throttle position or P_{intake})

- Throttling loss significant at light loads (see next page)
- Control of fuel/air ratio can adjust torque, but cannot provide sufficient range of control - misfire problems with lean mixtures
- Diesel - nonpremixed-charge - use fuel/air ratio control - no misfire limit - no throttling needed

Throttling loss

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- How much work is lost to throttling for fixed work or power output, i.e. a fixed BMEP, if fuel mass fraction (f) and N are constant?

$$\frac{\eta_{th}\text{ (with throttling)}}{\eta_{th}\text{ (without throttling)}} = \frac{\left(\text{Brake power}/\dot{m}_{fuel}Q_R\right)_{with}}{\left(\text{Brake power}/\dot{m}_{fuel}Q_R\right)_{without}} = \frac{\left(\dot{m}_{fuel}\right)_{without}}{\left(\dot{m}_{fuel}\right)_{with}}$$

$$= \frac{\left(\dot{m}_{air}(f/(1-f))\right)_{without}}{\left(\dot{m}_{air}(f/(1-f))\right)_{with}} = \frac{\left(\dot{m}_{air}\right)_{without}}{\left(\dot{m}_{air}\right)_{with}} = \frac{\left(\rho_{intake}\right)_{without}V_d N/n}{\left(\rho_{intake}\right)_{with}V_d N/n} =$$

$$\frac{\left(P_{intake}/RT_{intake}\right)_{without}}{\left(P_{intake}/RT_{intake}\right)_{with}} = \frac{\left(P_{intake}\right)_{without}}{\left(P_{intake}\right)_{with}} \approx \frac{\left(IMEP_g\right)_{without}/K}{\left(IMEP_g\right)_{with}/K} = \frac{\left(IMEP_g\right)_{without}}{\left(IMEP_g\right)_{with}}$$

$$IMEP_{without} = BMEP + FMEP;$$

$$IMEP_{with} = BMEP + FMEP + PMEP = BMEP + FMEP + (P_{ex} - P_{in})$$

$$= BMEP + FMEP + (P_{ex} - IMEP_{with}/K)$$

$$\Rightarrow IMEP_{with} = (BMEP + FMEP + P_{ex})(K/(K+1))$$

$$\Rightarrow \frac{\eta_{th}\text{ (with throttling)}}{\eta_{th}\text{ (without throttling)}} = \frac{BMEP + FMEP}{(BMEP + FMEP + P_{ex})(K/(K+1))}$$

Throttling loss

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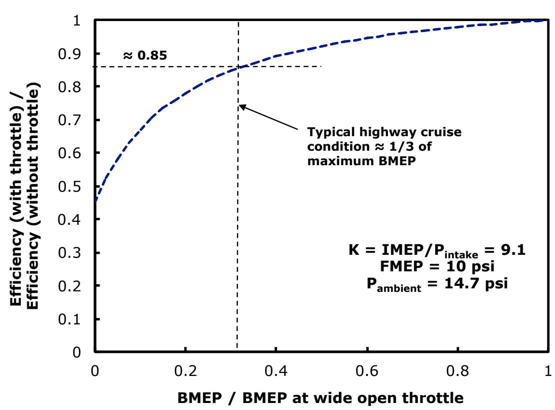
- Throttling loss increases from zero at wide-open throttle (WOT) to about half of all fuel usage at idle (other half is friction loss)
- At typical highway cruise condition (≈ 1/3 of BMEP at WOT), about 15% loss due to throttling
- Throttling isn't always bad, shifting to lower gear to reduce vehicle speed uses throttling loss (negative BMEP) and high N to maximize negative power

Double-click plot
To open Excel chart



AME 436 - Lecture 8 - Spring 2019 - Ideal cycle analysis

Throttling loss

- Another way to reduce throttling losses: close off some cylinders when low power demand
  - Cadillac had a 4-6-8 engine in the 1981 but it was a mechanical disaster
  - Mercedes had "Cylinder deactivation" on V12 engines in 2001 - 2002
  - GM uses a 4-8 "Active Fuel Management" (previously called "Displacement On Demand") engine
  - Nowadays several manufacturers have variable displacement engines (e.g Chrysler 5.7 L Hemi, "Multi-Displacement System")
  - Good summary articles on the mechanical aspects of variable displacement:

    http://www.autospeed.com/A_2618/xBXyt34qy_1/cms/article.html

    http://www.autospeed.com/cms/article.html?&title=Cylinder-Deactivation-Reborn-Part-2&A=2623
  - Certainly reduces throttling loss, but still have friction losses in inoperative cylinders

Throttling loss

- Aircycles4recips.xls (to be introduced in next lecture) analysis
  - Defaults: r = 9, V_d = 0.5 liter, P_{intake} = 1 atm, FMEP = 1 atm)
  - Predictions: P_{intake} = 1 atm, 13.45 hp, η = 29.96%
  - 1/3 of max. power via throttling: P_{intake} = 0.445 atm, 4.48 hp, η = 22.42%
  - 1/3 of max. power via halving displacement (double FMEP to account for friction losses in inoperative cylinders) P_{intake} = 0.806 atm, 4.48 hp, η = 24.78% (10.3% improvement over throttling)
  - Smaller engine operating at wide-open throttle to get same power: V_d = 0.5 liter / 3 = 0.167 liter, 4.48 hp, η = 29.96% (33.6% improvement over throttling bigger engine)
- Moral: if we all drove under-powered cars (small displacement) we'd get much better gas mileage than larger cars with variable displacement – could use turbocharging to regain maximum power (e.g. Ford EcoBoost)
- Hybrids use the "wide-open throttle, small displacement" idea and store surplus power in battery

Turbocharging & supercharging

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- Best way to increase power is to increase P_{in} above ambient using an air pump that forces air into engine
- Instead of pumping loss, you have a pumping gain! ☺
- Turbocharging: instead of blowdown (5 → 6), divert exhaust gas through a turbine & use shaft power to drive air pump; makes use of high pressure gas otherwise wasted during blowdown, thus thermal efficiency is increased
- Supercharging: air pump is driven directly from the engine rather than a separate turbine; if pump is 100% efficient (yeah, right…) then no loss or gain of overall cycle efficiency
- Limitations / problems
  - To get maximum benefit, need "intercooler" to cool intake air (thus increase density) after compression but before entering engine
  - Need time to overcome inertia of rotating parts & fill intake manifold with high-pressure air ("turbo lag")
  - Turbochargers: moving parts in hot exhaust system - not durable
  - Cost, complexity
- If an engine isn't turbocharged or supercharged, it's called "naturally aspirated"

Turbocharging & supercharging

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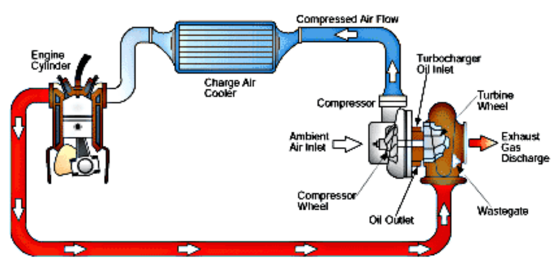
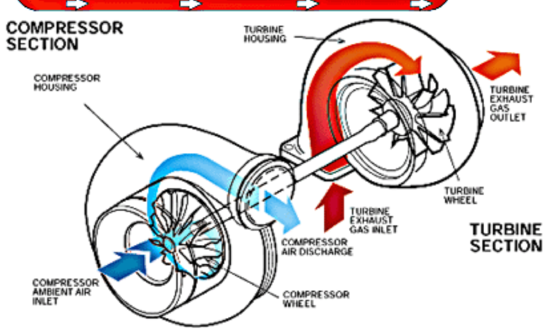
Source: http://auto.howstuffworks.com/turbo.htm





Turbocharging & supercharging

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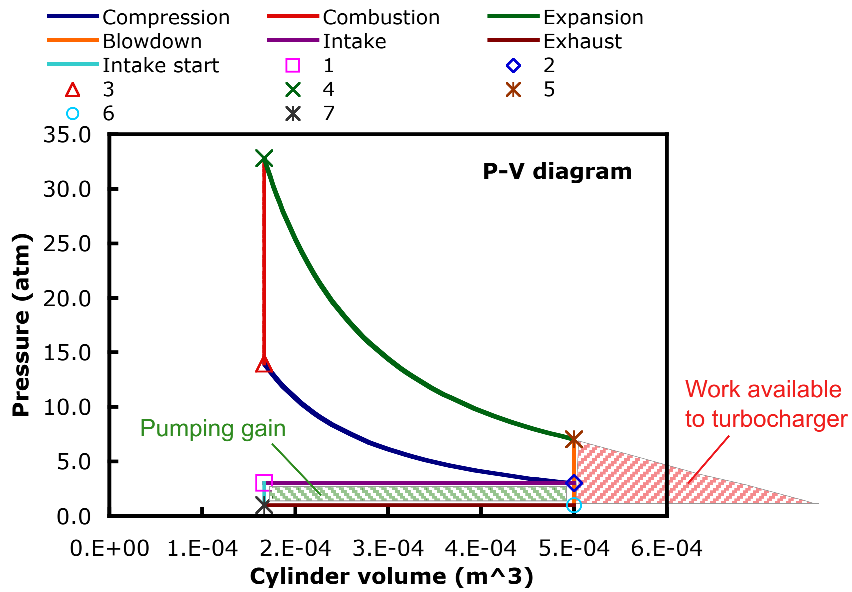

Turbocharging & supercharging

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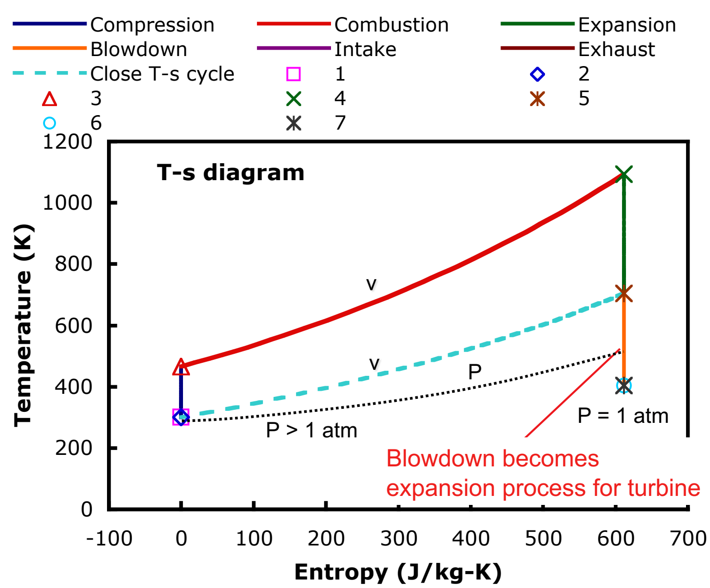

Why use Diesel cycle to model nonpremixed-charge engines?

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- Volume compression ratio = (volume ratio during heat addition) x (volume expansion ratio) as with reciprocating piston/cylinder arrangement (same reason as Otto/premixed)
- Heat input at constant pressure corresponds to slower combustion than constant-volume combustion - represents slower combustion than premixed-charge engine while still maintaining simple cycle analysis
- In reality, if burning were slow you would never wait until TDC to inject fuel, plus there is no way to ensure cylinder expansion rate exactly matches burning rate to obtain constant-P combustion
- Diesels have slower combustion since fuel is injected after compression, thus need to mix and burn, instead of just burn (already mixed before spark is fired) in premixed-charge engine
- Constant s compression/expansion corresponds to adiabatic & reversible process - not exactly true but a good first assumption
- Diesels not throttled (for reasons discussed later) (though often turbo/supercharged)

Ideal Diesel cycle analysis

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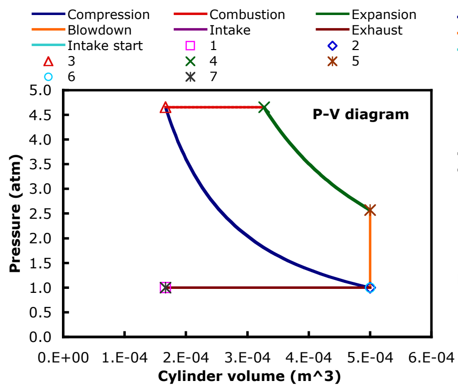
- Compression ratio r = $V_1/V_2 = V_2/V_3 = V_5/V_3 = V_6/V_7$
- New parameter: Cutoff ratio $\beta = V_4/V_3$; since 3 $\rightarrow$ 4 is const. P not const. V

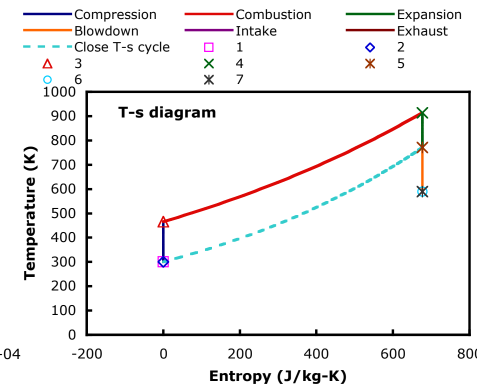
$\beta = V_4/V_3 = (mRT_4/P_4)/(mRT_3/P_3) = T_4/T_3$ (Cutoff ratio β not to be confused with non-dimensional activation energy β)

Stroke	Process	Name	Constant	Mass in cylinder	Other info
A	1 $\rightarrow$ 2	Intake	P	Increases	$P_2 = P_1$; $T_2 = T_1$ At 1, exhaust valve opens, intake valve closes
B	2 $\rightarrow$ 3	Compression	s	Constant	$P_3/P_2 = r^{\gamma}$; $T_3/T_2 = r^{(\gamma-1)}$ At 2, intake valve closes
---	3 $\rightarrow$ 4	Combustion	P	Constant	$T_4 = T_3 + fQ_R/C_P$; $T_4/T_3 = V_4/V_3$ At 3, fuel is injected
C	4 $\rightarrow$ 5	Expansion	s	Constant	$P_4/P_5 = (r/\beta)^{\gamma}$; $T_4/T_5 = (r/\beta)^{(\gamma-1)}$
---	5 $\rightarrow$ 6	Blowdown	V	Decreases	$P_6 = P_{ambient}$; $T_6/T_5 = (P_6/P_5)^{(\gamma-1)/\gamma}$ At 5, exhaust valve opens, exhaust gas "blows down" as with Otto
D	6 $\rightarrow$ 7	Exhaust	P	Decreases	$P_7 = P_6$; $T_7 = T_6$

P-V & T-s diagrams for ideal Diesel cycle

- Work is done during both 4 → 5 AND 3 → 4 (const. P combustion, volume increasing, thus $w_{3\rightarrow 4} = P_3(v_4 - v_3)$)
- Ambient intake pressure case shown (no pumping loop)





Diesel cycle analysis

- Thermal efficiency (ideal cycle, no throttling or friction loss)

$$\eta_{th} = \frac{\text{work out} + \text{work in}}{\text{heat in}} = \frac{C_v(T_4 - T_5) + P_4(v_4 - v_3) + C_v(T_2 - T_3)}{C_P(T_4 - T_3)}$$

$$= \frac{(T_4 - T_5) + (R/C_v)(T_4 - T_3) - (T_3 - T_2)}{(C_P/C_v)(T_4 - T_3)} = \frac{\gamma - 1}{\gamma} + \frac{T_4(1 - T_5/T_4) - T_3(1 - T_2/T_3)}{\gamma(T_4 - T_3)}$$

$$= 1 - \frac{1}{\gamma} + \frac{T_4(1 - (V_5/V_4)^{-(\gamma-1)}) - T_3(1 - (V_2/V_3)^{-(\gamma-1)})}{\gamma(T_4 - T_3)}$$

$$= 1 - \frac{1}{\gamma} + \frac{1}{\gamma} + \frac{T_4(-(V_5/V_4)^{-(\gamma-1)}) + T_3((V_2/V_3)^{-(\gamma-1)})}{\gamma(T_4 - T_3)}$$

$$= 1 + \frac{-T_4([(V_5/V_3)(V_3/V_4)]^{-(\gamma-1)}) + T_3((V_2/V_3)^{-(\gamma-1)})}{\gamma(T_4 - T_3)}$$

$$= 1 + \frac{-T_4([r/\beta]^{-(\gamma-1)}) + T_3(r^{-(\gamma-1)})}{\gamma(T_4 - T_3)} = 1 + \frac{-\beta T_3([r/\beta]^{-(\gamma-1)}) + T_3(r^{-(\gamma-1)})}{\gamma(\beta T_3 - T_3)}$$

$$= 1 - \frac{\beta([r/\beta]^{-(\gamma-1)}) - (r^{-(\gamma-1)})}{\gamma(\beta - 1)} = 1 - \frac{1}{r^{\gamma-1}}\frac{\beta^{\gamma} - 1}{\gamma(\beta - 1)}$$

Otto vs. Diesel cycle comparison

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- Thermal efficiency (ideal cycle, no throttling or friction loss)

$$\eta_{th,Diesel} = 1 - \frac{1}{r^{\gamma-1}}\left(\frac{\beta^{\gamma}-1}{\gamma(\beta-1)}\right); \text{ recall } \eta_{th,Otto} = 1 - \frac{1}{r^{\gamma-1}}(1)$$

$$\frac{\beta^{\gamma}-1}{\gamma(\beta-1)} > 1 \text{ for } \beta > 1, \gamma > 1; \frac{\beta^{\gamma}-1}{\gamma(\beta-1)} \to 1 \text{ as } \beta \to 1$$

$\Rightarrow$ For same r, η_{th} (Otto) > η_{th} (Diesel)

$\Rightarrow$ η_{th} (Otto) ≈ η_{th} (Diesel) as $\beta \to 1$ (small heat input)

- Lower η_{th} is due to burning at increasing volume, thus decreasing T - thus less efficient Carnot-cycle strips; most efficient burning strategy is at minimum volume, thus maximum T
- Note that (unlike Otto cycle) η_{th} is dependent on the heat input

$$\beta \equiv \frac{V_4}{V_3} = \frac{T_4}{T_3} = 1 + \frac{T_4 - T_3}{T_3} = 1 + \frac{fQ_R / C_P}{T_2 r^{\gamma-1}} = 1 + \frac{fQ_R}{C_P T_2 r^{\gamma-1}}$$

Higher heat input $\Rightarrow$ higher f $\Rightarrow$ larger β $\Rightarrow$ lower η_{th}

Otto vs. Diesel cycle comparison

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- Must have $V_4/V_3 \leq V_5/V_3$ (otherwise burning is still occurring at bottom of piston travel) thus we require $\beta \leq r$

$$\beta \leq r \Rightarrow 1 + \frac{fQ_R}{C_P T_2 r^{\gamma-1}} \leq r \Rightarrow f \leq \frac{(r-1)C_P T_2 r^{\gamma-1}}{Q_R}$$

For r = 20, Q_R = 4.3 x 10^7 J/kg, C_P = 1400 J/kgK, γ = 1.3, T_2 = 300K, requirement is f < 0.417, which is much greater than stoichiometric f (≈ 0.065) anyway so in practice this limit is never reached

- Typical f ≈ 0.04 (other parameters as above): β ≈ 2.67, η_{th} ≈ 0.515 (Diesel) vs. 0.593 (Otto), so difference not large for realistic conditions
- As with Otto, η_{th} increases as r increases - why not use $r \to \infty$ ($\eta_{th} \to 1$)? Unlike Otto, knock is not an issue - Diesel compresses air, not fuel/air mixture; main reason: heat losses
  - No knock issue so Diesels use much higher r
  - As gas is compressed more, T increases and V decreases, increasing temperature gradient $\Delta T/\Delta X$ for conduction loss
  - As conduction loss increases, compression work lost increases
  - At some point, lost work outweighs higher η_{th} of cycle having higher r
  - Also - as r increases, peak pressure increases - larger mechanical stresses for little improvement in η_{th}

Otto vs. Diesel cycle comparison

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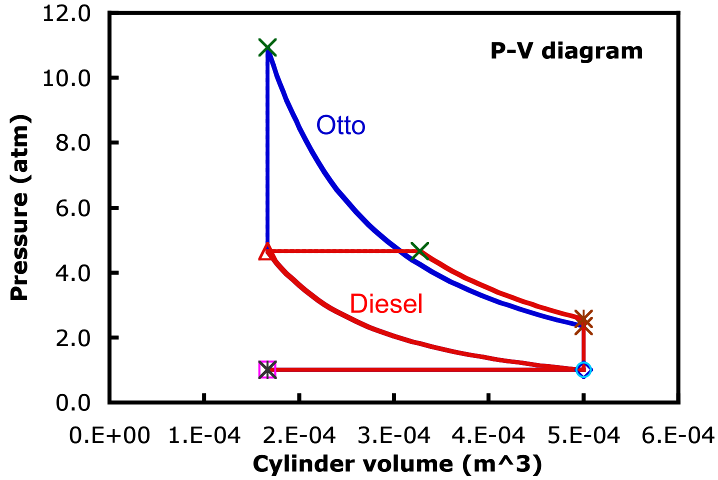

- Unthrottled Otto & Diesel with same compression ratio & heat input: Otto has higher peak P & T, more work output

Otto vs. Diesel cycle (animation)

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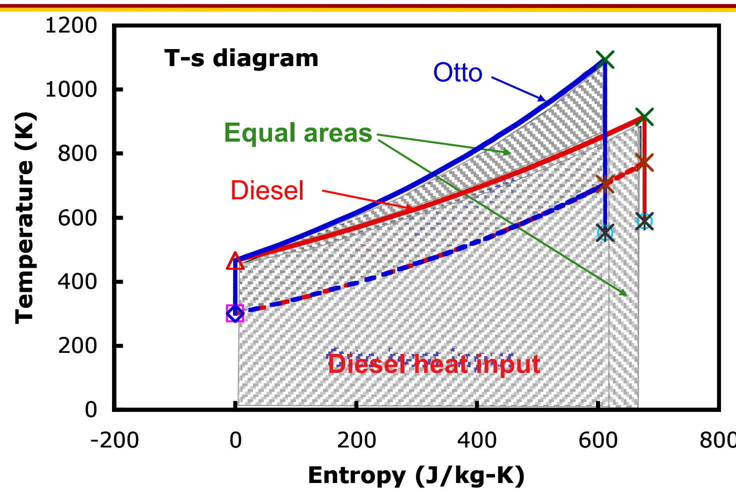

- Otto clearly has higher η_{th} - every Carnot strip has same T_L for both cycles, but every Otto strip has higher T_H
- Unlike Otto cycle, η_{th} for Diesel cannot be determined by inspection of the T - s diagram since each Carnot cycle strip has a different $1 - T_L/T_H$

Effect of compression ratio (Diesel)

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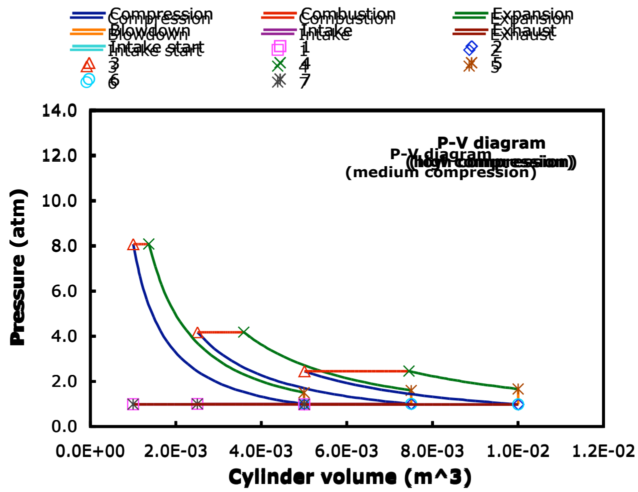
- Animation: P-V diagrams, increasing compression ratio (same displacement volume, same fuel mass fraction (f), thus same heat input)



Effect of compression ratio (Diesel)

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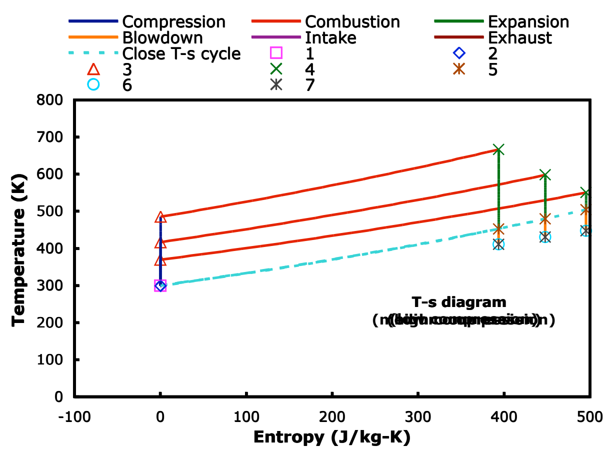
- Animation: T-s diagrams, increasing compression ratio (same displacement volume, same fuel mass fraction (f), thus same heat input)
- Higher compression clearly more efficient (taller Carnot strips)



Effect of heat input (Diesel)

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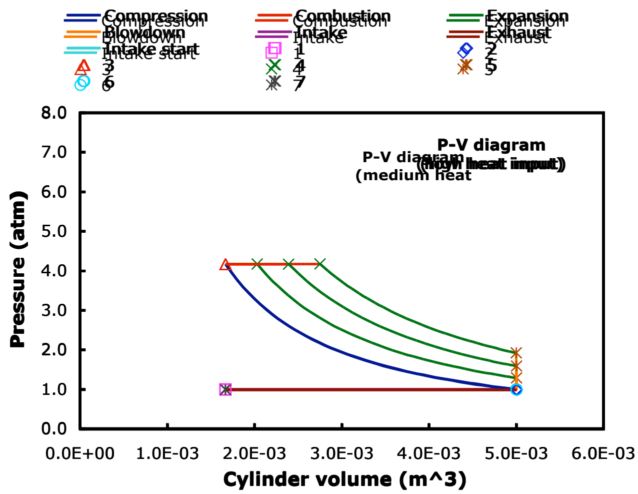
- Animation: P-V diagrams, increasing heat input via increasing f (same displacement volume, same compression ratio)



Effect of heat input (Diesel)

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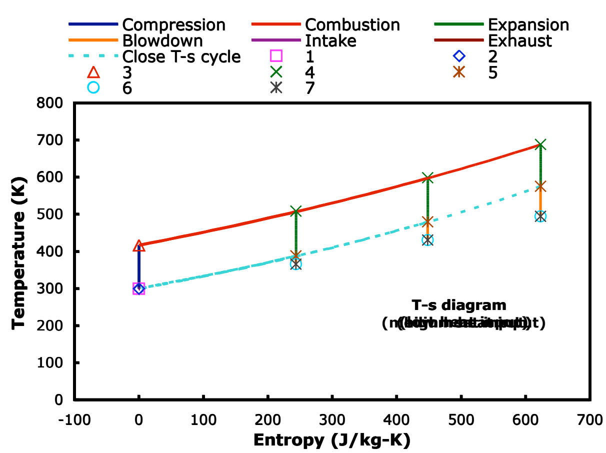
- Animation: T-s diagrams, increasing heat input via increasing f (same displacement volume, same compression ratio)
- Heat input does affect efficiency (shrinking T_L/T_H in Carnot strips as f increases)



Example

For an ideal Diesel cycle with the following parameters: r = 20, γ = 1.3, M = 0.029 kg/mole, f = 0.05, Q_R = 4.45 x 10^7 J/kg, initial temperature T_2 = 300K, initial pressure P_2 = 1 atm, P_{exh} = 1 atm, determine:

a) Temperature (T_3) & pressure (P_3) after compression & compression work per kg

$$\frac{P_3}{P_2} = r^\gamma \Rightarrow P_3 = P_2 r^\gamma = 1atm(20)^{1.3} \Rightarrow P_3 = 49.1atm$$

$$\frac{T_3}{T_2} = r^{\gamma-1} \Rightarrow T_3 = T_2 r^{\gamma-1} = 300K\left(20\right)^{1.3-1} \Rightarrow T_3 = 737K$$

$$W_{comp} = -C_v(T_3 - T_2) = -\frac{R}{\gamma-1}(T_3 - T_2) = -\frac{\Re}{M}\frac{1}{\gamma-1}(737-300) = -\frac{8.314J/moleK}{0.029kg/mole}\frac{1}{1.3-1}(737K-300K)$$

$$W_{comp} = -(955.6J/kgK)(737K-300K) = -417.6kJ/kg \text{ (negative since work is into system)}$$

b) Temperature (T_4) and pressure (P_4) after combustion, and the work output during combustion per kg of mixture

$$T_4 = T_3 + \frac{fQ_R}{C_p}; C_p = \frac{\gamma}{\gamma-1}R = \frac{1.3}{1.3-1}\frac{8.314J/moleK}{0.029kg/mole} = \frac{1242J}{kgK} \Rightarrow T_4 = 737K + \frac{(0.05)(4.45\times10^7 J/kg)}{1242J/kgK} = 2528K$$

$$P_4 = P_3 = 49.1atm \Rightarrow W_{comb} = P(v_4 - v_3) = Pv_3\left(\frac{v_4}{v_3} - 1\right) = RT_3\left(\frac{T_4}{T_3} - 1\right) = R\left(T_4 - T_3\right) = \frac{\Re}{M}\left(T_4 - T_3\right)$$

$$W_{comb} = \frac{8.314J/moleK}{0.029kg/mole}\left(2528K - 737K\right) = +513.5kJ/kg$$

Example (continued)

c) Cutoff Ratio

$$\beta = \frac{V_4}{V_3} = \frac{V_4}{m}\frac{m}{V_3} = \left(\frac{RT_3}{P_3}\frac{P_4}{RT_4}\right)^{-1} = \frac{T_4}{T_3} = \frac{2528K}{737K} = 3.43$$

d) Temperature (T_5) and pressure (P_5) after expansion, and the expansion work per kg of mixture

$$\frac{P_4}{P_5} = \left(\frac{r}{\beta}\right)^\gamma = \left(\frac{20}{3.43}\right)^{1.3} = 9.89 \Rightarrow P_5 = \frac{P_4}{9.89} = \frac{49.1atm}{9.89} = 4.96atm$$

$$\frac{T_4}{T_5} = \left(\frac{r}{\beta}\right)^{\gamma-1} = \left(5.83\right)^{1.3-1} = 1.7 \Rightarrow T_5 = \frac{T_4}{1.7} = \frac{2528K}{1.7} = 1489K$$

$$W_{exp} = -C_V\left(T_5 - T_4\right) = -955.63J/kgK\left(1489K - 2528K\right) = 992.9kJ/kg$$

e) Net work per kg of mixture

Net work = Compression work + Work during combustion + Expansion work
= -417.6 + 992.9 + 513.5 = 1089 kJ/kg

Example (continued)

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f) Thermal Efficiency

η_{th} = (Net work per unit mass) / (Heat input per unit mass)
= (Net work)/fQ_R = (1.089×10^6 J/kg)/(0.05)(4.45×10^7 J/kg) = 0.489 = 48.9%

compare this to the theoretical efficiency (should be the same since this is an ideal cycle analysis)

$$\eta_{th} = 1 - \frac{1}{r^{\gamma-1}}\left(\frac{\beta^{\gamma}-1}{\gamma(\beta-1)}\right) = 1 - \frac{1}{20^{1.3-1}}\left(\frac{3.43^{1.3}-1}{1.3(3.43-1)}\right) = 0.489 = 48.9\%$$

g) IMEP

$$IMEP = \frac{\text{Net work}}{V_d} = \frac{\frac{\text{Net work}}{\text{mass}}}{\frac{V_d}{\text{mass}}} = \frac{\frac{\text{Net work}}{\text{mass}}}{\frac{V_d}{\rho_2 V_d}} = \rho_2 \frac{\text{Net work}}{\text{mass}}$$

$$\Rightarrow IMEP = \frac{P_2}{(\Re/M)T_2}\frac{\text{Net work}}{\text{mass}} = (1atm)\frac{1.089\times10^6 J/kg}{\left(\frac{8.314J}{moleK}\Big/\frac{0.029kg}{mole}\right)300K} = 12.7atm$$

Note that the "mass" does not include the mass in the clearance volume; it is assumed that this mass is inert (i.e. exhaust gas) which does not yield additional heat release, plus its compression/expansion work cancels out

Examples of using P-V & T-s diagrams

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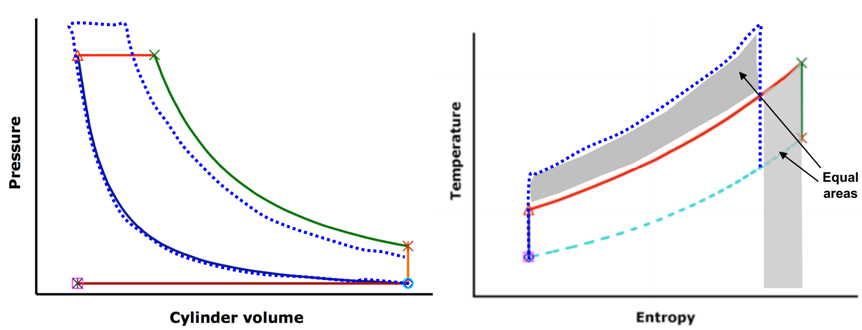
Consider the "baseline" ideal Diesel cycle shown on the P-V and T-s diagrams. Sketch modified diagrams if the following changes are made. Unless otherwise noted, assume in each case the initial T & P, r, f, Q_R, etc. are unchanged.

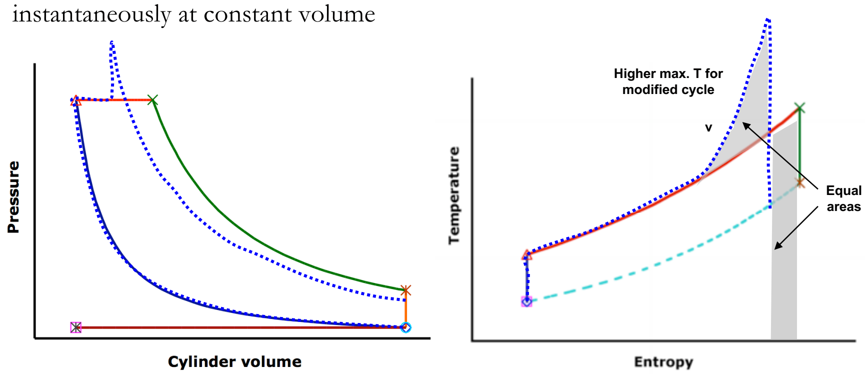
a) The compression ratio is increased (same **maximum** volume)



The minimum volume must decrease since r increases but the maximum volume does not. The cutoff ratio = 1 + $fQ_R/C_PT_2r^{\gamma-1}$ decreases. The temperature after compression T_3 as well as the maximum temperature $T_4 = T_3 + fQ_R/C_P$ increase. In order to maintain equal heat addition and thus equal areas on the T-s diagram, s_4 decreases.

Examples of using P-V & T-s diagrams

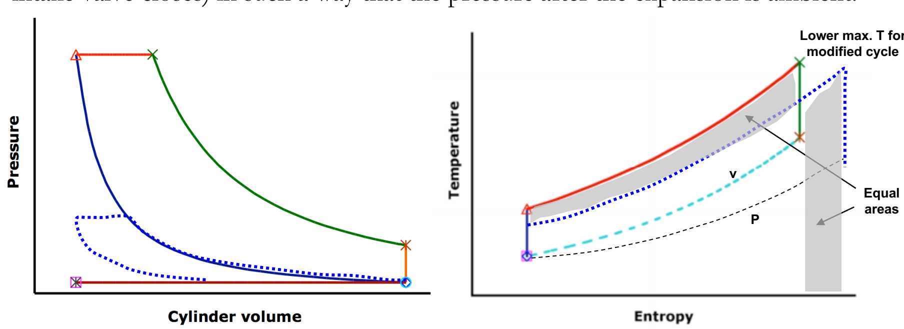
b) Part way through the constant-pressure burn, the remainder of the burn occurs instantaneously at constant volume



The cycle is the same until the 2nd part of the burn occurs at constant volume rather than constant pressure, which has a higher slope on the T-s diagram. To maintain the same total heat release, s_4 must decrease. Also, because constant volume combustion results in higher T than constant pressure combustion (since $C_V < C_P$), the maximum T increases.

Examples of using P-V & T-s diagrams

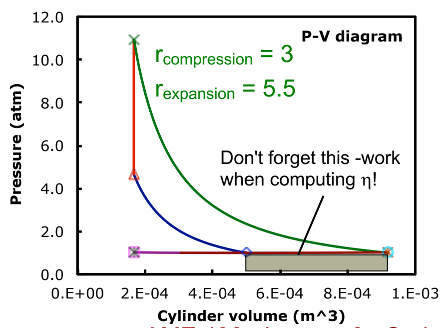
c) The intake valve closes late (i.e. after part of the compression stroke has started; the pressure stays at ambient pressure and no compression occurs until after the intake valve closes) in such a way that the pressure after the expansion is ambient.

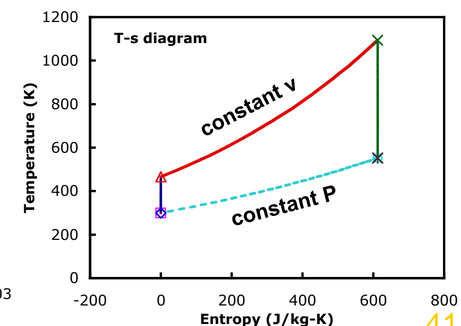


In order to have the pressure after expansion be ambient, the P-V curve for the expansion part of the modified cycle must be the same as the compression part of the base cycle. The pressure remains at ambient part way through the compression stroke, until the intake valve closes. The compression ratio is lower, thus the P and T after compression (state 3) are lower. s_4 must increase in order for the heat release (thus area) of the modified cycle to be the same as that of the baseline cycle.

Complete Expansion cycle

- Highest efficiency cycle consistent with piston/cylinder engine has constant-V combustion but expansion back to ambient P – Complete Expansion or Humphrey cycle (caution: different sources have different cycle naming conventions – Atkinson, Humphrey, Miller etc. – wikipedia.com is becoming the new default standard!)
- Needs different compression & expansion ratios - can be done by closing the intake valve AFTER the "compression" starts or by extracting power in a turbine whose work is somehow connected to the main shaft power output





Ideal 4-stroke Complete expansion cycle

- Compression ratio $r_c = V_2/V_3$; Expansion ratio $r_e = V_5/V_4$

Stroke	Process	Name	Constant	Mass in cylinder	Other info
A	$1 \rightarrow 2$	Intake	P	Increases then decreases	$P_2 = P_1$; $T_2 = T_1$ At 1, exhaust valve closes, intake valve opens; volume increases from V_1 (minimum) to V_5 (maximum) then back to V_2
B	$2 \rightarrow 3$	Compression	s	Constant	$P_3/P_2 = r_c^{\gamma}$; $T_3/T_2 = r_c^{(\gamma-1)}$ At 2, intake valve closes
---	$3 \rightarrow 4$	Combustion	V	Constant	$T_4 = T_3 + fQ_R/C_v$; $P_4/P_3 = T_4/T_3$ At 3, spark fires
C	$4 \rightarrow 5$	Expansion	s	Constant	$P_5 = P_2 = P_1$; $P_4/P_5 = r_e^{\gamma}$; $T_4/T_5 = r_e^{(\gamma-1)}$; $V_5 = V_1$
	$5 \rightarrow 6$	"Blowdown"	Every-thing	Constant	No blowdown in complete expansion cycle, nothing happens
D	$6 \rightarrow 7$	Exhaust	P	Decreases	$P_7 = P_6 = P_{ambient}$; $T_7 = T_6$; $V_7 = V_1$ At 6, exhaust valve opens

Complete Expansion cycle analysis

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Isentropic compression: $V_3 = V_2 / r_c;\ T_3 = T_2 r_c^{\gamma-1}; P_3 = P_2 r_c^{\gamma}$

Constant volume combustion: $V_4 = V_3$

$$T_4 = T_3 + \frac{fQ_R}{C_V} = T_2 r_c^{\gamma-1} + \frac{fQ_R}{C_V} = T_2 r_c^{\gamma-1}\left(1+\Gamma\right); \Gamma = \frac{fQ_R}{C_V T_2 r_c^{\gamma-1}} \text{ (dimensionless heat input)}$$

$$\left\{\text{Recall from Diesel cycle analysis: } \beta = 1 + \frac{fQ_R}{C_P T_2 r_c^{\gamma-1}}, \text{ so } \Gamma = \gamma\left(\beta - 1\right)\right\}$$

$$P_4 = P_3\left(\frac{T_4}{T_3}\right) = P_2 r_c^{\gamma}\left(\frac{T_2 r_c^{\gamma-1}\left(1+\Gamma\right)}{T_2 r_c^{\gamma-1}}\right) = P_2 r_c^{\gamma}\left(1+\Gamma\right)$$

Isentropic expansion: $P_5 = P_2$, expansion ratio $r_e = V_5 / V_4 > r_c$

$$P_4 = P_5 r_e^{\gamma} \Rightarrow r_e = \left(\frac{P_4}{P_5}\right)^{1/\gamma} = \left(\frac{P_2 r_c^{\gamma}\left(1+\Gamma\right)}{P_2}\right)^{1/\gamma} = r_c\left(1+\Gamma\right)^{1/\gamma} \text{ or } \frac{r_e}{r_c} = \left(1+\Gamma\right)^{1/\gamma}$$

$$T_4 = T_5 r_e^{\gamma-1} \Rightarrow T_5 = \frac{T_4}{r_e^{\gamma-1}} = \frac{T_2 r_c^{\gamma-1}\left(1+\Gamma\right)}{\left[r_c\left(1+\Gamma\right)^{1/\gamma}\right]^{\gamma-1}} = T_2\left(1+\Gamma\right)^{1/\gamma}$$

Complete expansion cycle analysis

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$$\eta_{th} = \frac{\text{net work}}{\text{heat in}} = \frac{C_v(T_4 - T_5) + C_v(T_2 - T_3) + P_2(v_2 - v_5)}{C_v(T_4 - T_3)}$$

$$\text{Note } P_2(v_2 - v_5) = P_2 v_2\left(1 - \frac{v_5}{v_4}\frac{v_4}{v_2}\right) = RT_2\left[1 - P_2 v_2\left(1 - \frac{v_5}{v_4}\frac{v_4}{v_2}\right)\right] = RT_2\left(1 - \frac{r_e}{r_c}\right) = RT_2\left[1 - \left(1+\Gamma\right)^{1/\gamma}\right]$$

$$\eta_{th} = \frac{T_4 - T_5 + T_2 - T_3 + \frac{RT_2}{C_v}\left[1 - \left(1+\Gamma\right)^{1/\gamma}\right]}{T_4 - T_3}$$

$$= \frac{T_2 r_c^{\gamma-1}\left(1+\Gamma\right) - T_2\left(1+\Gamma\right)^{1/\gamma} + T_2 - T_2 r_c^{\gamma-1} + (\gamma-1)T_2\left[1 - \left(1+\Gamma\right)^{1/\gamma}\right]}{fQ_R / C_v}$$

$$= \frac{\left(1+\Gamma\right) - \frac{\left(1+\Gamma\right)^{1/\gamma}}{r_c^{\gamma-1}} + \frac{1}{r_c^{\gamma-1}} - 1 + \frac{\gamma-1}{r_c^{\gamma-1}}\left[1 - \left(1+\Gamma\right)^{1/\gamma}\right]}{fQ_R / C_v T_2 r_c^{\gamma-1}} = \frac{\Gamma - \frac{1}{r_c^{\gamma-1}}\left\{\left[\left(1+\Gamma\right)^{1/\gamma} - 1\right] + (\gamma-1)\left[\left(1+\Gamma\right)^{1/\gamma} - 1\right]\right\}}{\Gamma}$$

$$\boxed{\eta_{th} = 1 - \frac{1}{r_c^{\gamma-1}}\frac{\gamma\left[\left(1+\Gamma\right)^{1/\gamma} - 1\right]}{\Gamma}} \text{(finally!)}$$

Otto vs. Complete Exp. cycle comparison

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- Thermal efficiency (ideal cycles, no throttling or friction loss)

$$\eta_{th,CompleteExpansion} = 1 - \frac{1}{r_c^{\gamma-1}} \frac{\gamma\left[(1+\Gamma)^{1/\gamma} - 1\right]}{\Gamma};\quad \frac{\gamma\left[(1+\Gamma)^{1/\gamma} - 1\right]}{\Gamma} < 1 \text{ for } \Gamma > 0$$

$$\text{thus } \eta_{th,CompleteExpansion} = 1 - \frac{1}{r_c^{\gamma-1}}\left(\frac{\gamma\left[(1+\Gamma)^{1/\gamma} - 1\right]}{\Gamma}\right) > 1 - \frac{1}{r_c^{\gamma-1}}(1) = \eta_{th,Otto}$$

$$\text{Also as } \Gamma \to 0,\ \frac{\gamma\left[(1+\Gamma)^{1/\gamma} - 1\right]}{\Gamma} \to 1 \Rightarrow \eta_{th,CompleteExpansion} = 1 - \frac{1}{r_c^{\gamma-1}}(1) \text{ (same as Otto as } \Gamma \to 0)$$

$$\text{Also as } \Gamma \to \infty,\ \frac{\gamma\left[(1+\Gamma)^{1/\gamma} - 1\right]}{\Gamma} \to \gamma\frac{\gamma\ \Gamma^{1/\gamma}}{\Gamma} = \frac{1}{\Gamma^{\gamma-1/\gamma}} \to 0 \ \Rightarrow \eta_{th,CompleteExpansion} \to 1 \text{ as } \Gamma \to \infty$$

Otto vs. Complete Exp. cycle comparison

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- For same r_c, η_{th} (Complete Expansion) > η_{th} (Otto) (obviously, since more expansion work with compression work and heat input is the same)
- η_{th} (Complete Expansion) ≈ η_{th} (Otto) as $\Gamma \to 0$ (small heat input)
- Like Diesel, Complete Expansion and Otto converge to same cycle at small heat input due to complete expansion (same T_H & T_L as Otto)
- Unlike Otto cycle, but like Diesel cycle) η_{th} in Complete Expansion cycle is dependent on the heat input (through Γ), but unlike Diesel cycle, $\eta_{th,CompleteExpansion}$ increases (rather than decreases, as in Diesel) with increasing heat input

Complete Expansion cycle

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- Thermodynamically, ideal Complete Expansion cycle is same as turbocharged cycle if you take turbine work as net work output rather than driving an air pump
- Gain due to complete expansion is substantial
  - Low-r cycle ideal shown on page 41: η_{th} = 0.424 vs. 0.356
  - Realistic cycle parameters (next lecture): η_{th} = 0.366 vs. 0.295
- No muffler needed if exhaust always leaves cylinder at 1 atm!
- Late intake valve closing is preferable (higher η_{th}) to throttling for part-load operation, but
  - Mechanically much more complex
  - Every power level requires a different intake valve closing time, thus a different $r_{compression}$, thus a different $r_{expansion}$
  - Since intake valve is closed late, maximum mass flow is less than conventional Otto cycle, so less power - at higher power levels can't sustain complete expansion cycle
- This is an alternative to throttling for premixed-charge engines, but why don't we throttle Diesel engines? (Answer in 2 slides…)

Complete Expansion cycle

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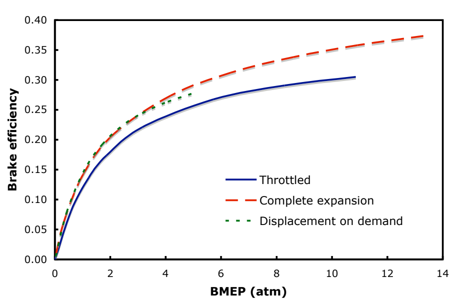
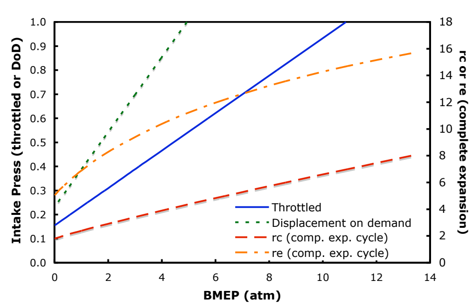
- What is kept constant in comparison below? Clearance volume
- Complete Expansion cycle provides higher BMEP (based on V_d for maximum-power cycle) AND higher efficiency, but at the expense of greatly increased mechanical complexity (and need for 2x larger piston stroke for wide-open throttle operation)
- Displacement-on-demand (1/2 displacement) similar to Complete Expansion, but doesn't benefit high end of power range





r_c = 8 (throttled & DoD), γ = 1.3, f = 0.068, Q_R = 4.5 x 10^7 J/kg, T_{in} = 300K, P_{in} = 1 atm (CE), P_{exh} = 1 atm, ExhRes = TRUE, Const-v comb, BurnStart = 0.045, BurnEnd = 0.105, h = 0.01, η_{comp} = η_{exp} = 0.9, FMEP = 1 atm

Ronney's catechism (1/4)

- Why do we throttle in a premixed charge engine despite the throttling losses it causes?

  Because we have to reduce power & torque when we don't want the full output of the engine (which is most of the time in LA traffic, or even on the open road)
- Why don't we have to throttle in a nonpremixed charge engine?

  Because we use control of the fuel to air ratio (i.e. to reduce power & torque, we reduce the fuel for the (fixed) air mass)
- Why don't we do that for the premixed charge engine and avoid throttling losses?

  Because if we try to burn lean in the premixed-charge engine, when the equivalence ratio (ϕ) is reduced below about 0.7, the mixture misfires and may stop altogether
- Why isn't that a problem for the nonpremixed charge engine?

  Nonpremixed-charge engines are not subject to flammability limits like premixed-charge engines since there is a continuous range of fuel-to-air ratios varying from zero in the pure air to infinite in the pure fuel, thus someplace there is a stoichiometric ($\phi = 1$) mixture that can burn. Such variation in ϕ does not occur in premixed-charge engines since, by definition, ϕ is the same everywhere.

Ronney's catechism (2/4)

- So why would anyone want to use a premixed-charge engine?

  Because the nonpremixed-charge engine burns its fuel slower, since fuel and air must mix before they can burn. This is already taken care of in the premixed-charge engine. This means lower engine RPM and thus less power from an engine of a given displacement
- Wait - you said that the premixed-charge engine is slower burning.

  Only if the mixture is too lean. If it's near-stoichiometric, then it's faster because, again, mixing was already done before ignition (ideally, at least). Recall that as ϕ drops, T_{ad} drops proportionately, and burning velocity (S_L) drops exponentially as T_{ad} drops
- Couldn't I operate my non-premixed charge engine at overall stoichiometric conditions to increase burning rate?

  No. In nonpremixed-charge engines it still takes time to mix the pure fuel and pure air, so (as discussed previously) burning rates, flame lengths, etc. of nonpremixed flames are usually limited by mixing rates, not reaction rates. Worse still, with initially unmixed reactants at overall stoichiometric conditions, the last molecule of fuel will never find the last molecule of air in the time available for burning in the engine - one will be in the upper left corner of the cylinder, the other in the lower right corner. That means unburned or partially burned fuel would be emitted. That's why diesel engines smoke at heavy load, when the mixture gets too close to overall stoichiometric.

Ronney's catechism (3/4)

- So what wrong with operating at a maximum fuel to air ratio a little lean of stoichiometric?

  That reduces maximum power, since you're not burning every molecule of O_2 in the cylinder. Remember - O_2 molecules take up a lot more space in the cylinder that fuel molecules do (since each O_2 is attached to 3.77 N_2 molecules), so it behooves you to burn every last O_2 molecule if you want maximum power. So because of the mixing time as well as the need to run overall lean, Diesels have less power for a given displacement / weight / size / etc.
- So is the only advantage of the Diesel the better efficiency at part-load due to absence of throttling loss?

  No, also you can use higher compression ratios, which increases efficiency at any load. This helps alleviate the problem that slower burning in Diesels means lower inherent efficiency (more burning at increasing cylinder volume)
- Why can the compression ratio be higher in the Diesel engine?

  Because you don't have nearly as severe problems with knock. That's because you compress only air, then inject fuel when you want it to burn. In the premixed-charge case, the mixture being compressed can explode (since it's fuel + air) if you compress it too much

Ronney's catechism (4/4)

- Why is knock so bad?

  We'll discuss that in the section on knock in lecture 10.
- So, why have things evolved such that small engines are usually premixed-charge, whereas large engines are nonpremixed-charge?

  In small engines (lawn mowers, automobiles, etc.) you're usually most concerned with getting the highest power/weight and power/volume ratios, rather than best efficiency (fuel economy). In larger engines (trucks, locomotives, tugboats, etc.) you don't care as much about size and weight but efficiency is more critical
- But unsteady-flow aircraft engines, even large ones, are almost always premixed-charge, because weight is always critical in aircraft

  You got me on that one! (Though there is at least one production Diesel powered aircraft.) But of course most large aircraft engines are steady-flow gas turbines, which kill unsteady-flow engines in terms of power/weight and power/volume (but not efficiency.)

Fuel-air cycles

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- So far we've studied "air cycles" where gas properties (C_P, C_v, γ…) are assumed constant so simple relations like Pv^{γ} = constant & $T_{ad} = T_{\infty} + fQ_R/C_v$) can be used
- This yields simple closed-form results, but isn't realistic
- More realistic estimate obtained using "real" gas properties from GASEQ or other chemical equilibrium program
  - Uses variable gas properties and compositions
  - Shows that rich mixtures have low η_{th} due to throwing away fuel that can't be burned due to lack of O_2
  - Shows limitation on work output because of limitation on heat input provided by combustion
- Still doesn't consider effects of
  - Finite burning time (especially lean & rich mixtures)
  - Incomplete combustion (crevice volumes, flame quenching, etc.)
  - Heat & friction losses
  - Hydrodynamic (pressure) losses in intake/exhaust
  - Etc., etc.

Fuel-air cycle analysis using GASEQ

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- Under "Units" menu, check "mass", "Joules" and "atm"
- At the top of the page, under "Problem type" select "adiabatic compression/expansion" and make sure the box "frozen composition" is checked
- Under "Reactants" select appropriate set of reactants, e.g. "Iso-octane-air flame"
- To set the stoichiometry, first click on one of the species (e.g. the fuel), then to the right of the "Stoichiometry" box, click the button called "set"; in the dialog box that pops up, enter the equivalence ratio you want, then close the box
- Note (i.e. write down) the mass fraction of fuel
- In the box below the reactants box, enter the reactant temperature and pressure (e.g. 298K, 1 atm) and the volume ratio of the compression process (e.g. 1/8 = 0.125 for compression ratio of 8) OR the final pressure (NOT pressure ratio) (e.g. 5 atm for a pressure ratio of 10 with reactants at 0.5 atm) in the locations provided
- Click on the "calculate" box. Note the internal energy, enthalpy and specific volume of both the reactants (call them u_2, h_2, v_2) and the products (u_3, h_3, v_3)
- Click on the "R<<P" button to make the products from this calculation the reactants for the next calculation
- At the top of the page, under "Problem type" select "adiabatic T and composition at const P" or "adiabatic T and composition at const v" depending on whether the assumption is constant pressure or constant volume combustion
- On the right side of the page, under "Products" select the appropriate products (e.g. "HC/O2/N2 products (extended)")
- Click on the "calculate" button. Note the internal energy, enthalpy and specific volume of the products (u_4, h_4, v_4). If Diesel cycle, compute the cutoff ratio $\beta = v_4/v_3$.

Fuel-air cycle analysis using GASEQ

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- Click on the "R<<P" button to make the products from this calculation the reactants for the next calculation
- At the top of the page, under "Problem type" select "adiabatic compression/expansion" and make sure the box "frozen composition" is either checked or not checked, depending on your assumption (usually you want to assume NOT frozen)
- In the box below the products box, enter the required volume expansion ratio (e.g. same as r_c for Otto, or r_c/β for Diesel) or the final pressure (again, NOT pressure ratio) in the location provided
- Click on the "calculate" button. Note product internal energy, enthalpy and specific volume (u_5, h_5, v_5)
- Calculate the heat input per unit mass of reactants
  = [fuel mass fraction (f)] x [heating value of fuel (Q_R)]
- Determine work for each process according to the following table

Process	Compression work in	Heat addition, constant P	Heat addition, constant v	Expansion work out
Control mass (unsteady flow)	$u_2 - u_3$	$P_3(v_4 - v_3)$	0	$u_4 - u_5$
Control volume (steady flow)	$h_2 - h_3$	0	N/A	$h_4 - h_5$

- Determine net work done per unit mass = work in + work out
- Determine thermal efficiency = (net work per unit mass)/(heat input per unit mass)

Fuel-air cycle analysis

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- Air cycle: doesn't know about combustion limitations, so $\eta_{th} = 1 - (1/r)^{\gamma-1}$, but γ decreases as ϕ increases (more "heavy" fuel molecules with high molecular mass M, thus low γ)

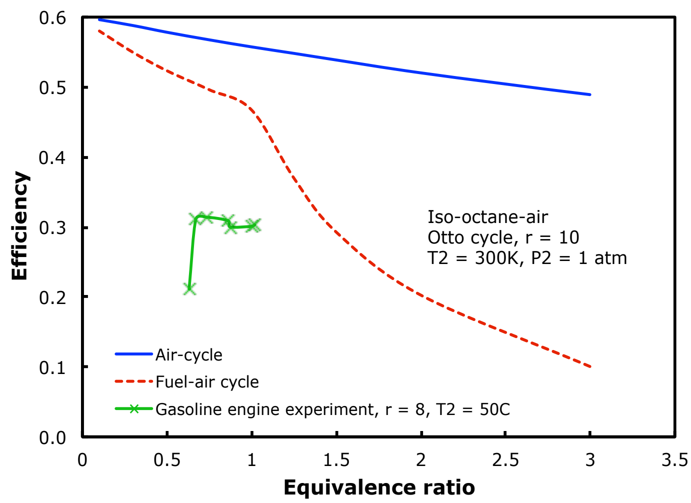
$$\gamma \equiv \frac{C_P}{C_V} = \frac{C_V + R}{C_V} = 1 + \frac{\Re / M}{C_V} \Rightarrow \gamma \downarrow as\ M \uparrow$$

- Fuel-air cycle
  - At low ϕ, just like air cycle since less fuel or anything other than air
  - As ϕ increases, η_{th} decreases because T_{ad} increases and thus γ decreases

$$\gamma \equiv \frac{C_P}{C_V} = \frac{C_V + R}{C_V} = 1 + \frac{\Re / M}{C_V}; C_V \uparrow as\ T \uparrow \quad \Rightarrow \gamma \downarrow as\ T \uparrow$$

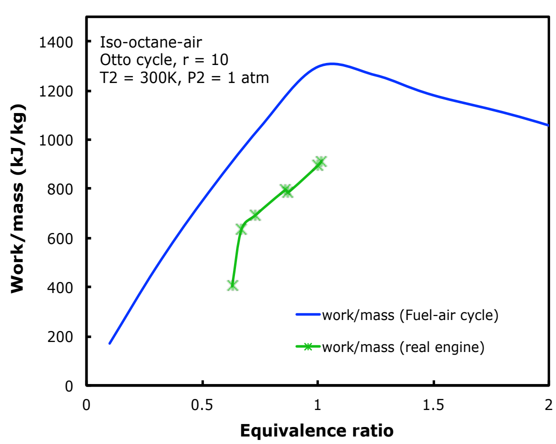
  - Also as ϕ increases,
  - $\phi > 1$: not enough O_2 to burn all fuel, so η_{th} decreases rapidly
- Real engine
  - Lower η_{th} even at $\phi = 1$, drops off rapidly below ≈ 0.65 due to lean misfire (lower $\phi \Rightarrow$ lower $T_{ad} \Rightarrow$ much lower S_L, not enough burn time)
  - η_{th} peaks at slightly < 1 since ϕ < 1 ensures there's enough air to burn every fuel molecule)

Fuel-air cycle analysis



Fuel-air cycle analysis

- BMEP = Work/V_d = (Work/mass)/(V_d/mass) = ρ_{intake}(Work/mass)
- Work/mass or BMEP peaks at ϕ slightly > 1 (unlike η_{th} which peaks slightly < 1) - why? BMEP is limited by the ability to burn all O_2, which takes up most of the space (each O_2 attached to 3.77 N_2, fuel occupies only ≈ 2% by moles or volume), so burning rich ensures all O_2 is burned



Example

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Repeat the previous example of the ideal Diesel air-cycle analyis using a fuel-air cycle analysis (using GASEQ) with a lean iso-octane air mixture with ϕ = 0.792.

P_2 = 1 atm, T_2 = 300K, f = 0.05 (ϕ = 0.792), u_2=-172.54 kJ/kg, ρ_2=1.2176 kg/m^3

From GASEQ with "Adiabatic Compression/Expansion," "Frozen Chemistry," and "Volume Products/Volume Reactants" = 0.05 (r = 20), we obtain

P_3=53.7 atm, T_3=805K, u_3=260.17 kJ/kg, ρ_3=24.35 kg/m^3

Click "R<<P," select "Adiabatic T and Composition at constant P" which yields

P_4=53.7 atm, T_4=2422K, u_4=-220.57 kJ/kg, ρ_4=7.725 kg/m^3

The final volume must be the same as the initial volume, and the volume ratio during combustion is v_4/v_3 = ρ_3/ρ_4 = 24.366/7.7288 = 3.153, so the volume ratio during expansion must be 20/3.153 = 6.343. Thus, click "R<<P," select "Adiabatic Compression/Expansion," uncheck "Frozen Chemistry," and in "Volume Products/Volume Reactants" enter 6.343. This yields

P_5=5.34 atm, T_5=1530 K, u_5=-1270.00 kJ/kg, ρ_5=1.2178 kg/m^3

(which is essentially the same as the initial density of 1.2176 kg/m^3, thus we did the expansion right, because the initial mass and volume are the same as the final mass and volume.)

Example

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Compression work = $u_2 - u_3$ = -172.54 – 260.17 = -432.71 kJ/kg

Expansion work = $u_4 - u_5$ = -220.57 – (-1270.00) = 1049.43 kJ/kg

Work during combustion = $P_3(v_4 - v_3) = P_3(1/\rho_4 - 1/\rho_3)$
= (53.7 atm)(1.01325 x 10^5 N/m^{2}atm)(1/7.725 – 1/24.35)m^3/kg (1 kJ/1000J)
= 480.90 kJ/kg

Net Work = -432.71 + 1028.11 + 480.90 = 1076.3 kJ/kg = 1.0763 x 10^6 J/kg

Cut off Ratio = $v_4/v_3 = \rho_3/\rho_4$ = 24.35/7.725 = 3.152

Thermal efficiency = Net work / heat in
= (1.0763 x 10^6 J/kg)/ (0.05 x 4.45 x 10^7 J/kg) = 48.4%

$$IMEP = \frac{\text{Net work}}{V_d} = \frac{\text{(Net work)/mass}}{V_d / mass} = \frac{\text{Net work/mass}}{V_d / (\rho_2 V_d)} = \rho_2 \text{(Net work/mass)}$$

$$\Rightarrow IMEP = \frac{(1.2176 kg/m^3)(1.076\times 10^6 J/kg)}{101325 N/m^2 atm} = 12.9 atm$$

Comment: GASEQ considers dissociation of products, and C_P and C_V are increase with temperature, both of which greatly decrease T_4 compared to the air cycle analysis. The net work, efficiency and IMEP are similar to the previous air-cycle example because of the use of an effective γ = 1.3; if we had used γ = 1.4, the air-cycle analysis would have yielded considerably higher efficiency and IMEP

Summary

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- Air-cycle analysis is very useful for understanding ICE performance
- P-V & T-s diagrams and numerical analysis are complementary
- Various cycles used to model different "real" engines, but most involve isentropic compression/expansion & constant V or P heat addition
- Engines are air processors; more air processed $\Rightarrow$ more power produced
- The differences between premixed (gasoline-type) and non-premixed charge (diesel-type) combustion lead to major differences in performance and thus which applications are most appropriate
  - Power is controlled by air flow (via throttle) in premixed charge engines but via fuel/air ratio (FAR) in nonpremixed charge engines
  - Compression ratio is limited by knock in premixed charge engines but by heat losses in nonpremixed charge engines
- Fuel-air cycles, where "real" gas properties are employed, are more realistic approximations than air cycles but still removed from reality
- The ideal cycle analyses do NOT show the limitations of performance imposed by heat losses, slow burning, knock, friction, etc. - coming up in next 2 lectures