AME 436	**Assigned: Friday 3/6/09**
Problem Set #4	• **Due Friday 3/13/09 at 4:30 pm in OHE 430J** • **Email to the grader (Thada Suksila, suksila@usc.edu) or fax to 213-740-8071 if you're off campus** • **DEN students submit through the usual channels**

For all problems, if useful, you can use the AirCycles.xls spreadsheet to guide your answers but you need to explain your results. Note: laptops or Pocket PCs running Excel spreadsheets will NOT be permitted on the exams.

Problem #1

For an Otto cycle with constant-volume combustion and the following parameters: $r = 9$, $\gamma = 1.3$, $M = 0.029$ kg/mole, $f = 0.062$, $Q_R = 4.3 \times 10^7$ J/kg, $T_2 = 300K$, $P_2 = 0.5$ atm, $P_{exh} = 1$ atm, $h = 0$, $\eta_{comp} = \eta_{exp} = 0.9$ (in other words, ideal **except for the compression and expansion efficiency, and the throttling**), determine the following:

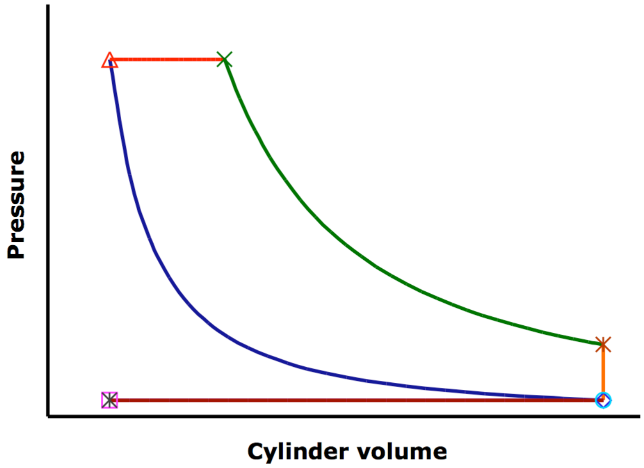
a) Temperature (T_3) and pressure (P_3) after compression, and the compression work per kg of mixture
b) Temperature (T_4) and pressure (P_4) after combustion
c) Temperature (T_5) and pressure (P_5) after expansion, and the expansion work per kg of mixture
d) Net work per kg of mixture (don't forget about the throttling loss!)
e) Thermal efficiency
f) IMEP

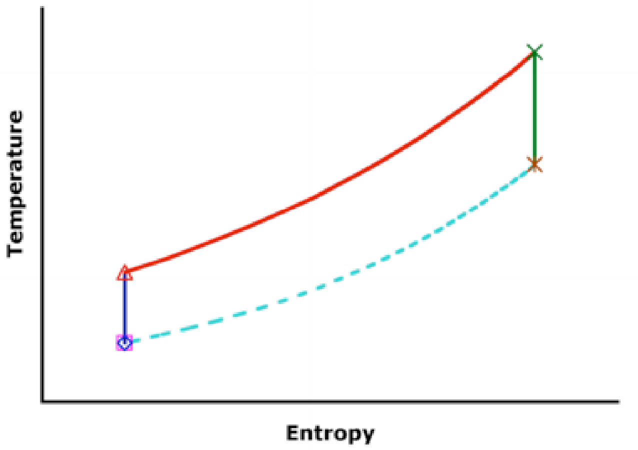
Note in this case that we cannot simply use P_2r^γ and $T_3 = T_2r^{\gamma-1}$; instead we have to use the definitions of compression and expansion efficiency given in lecture 6 to get the pressures and temperatures after compression or expansion.

Problem #2

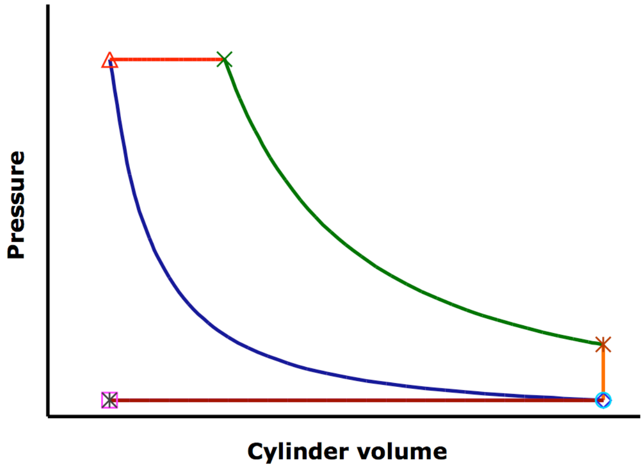
Consider the "baseline" ideal Diesel cycle shown on the P-V and T-s diagrams. Sketch modified P-V and T-s diagrams if the following changes are made. Unless otherwise noted, assume in each case the initial temperature and pressure, compression ratio, fuel mass fraction, heating value, etc. are unchanged. Where useful for clarity, label plots with phrases like "this area = that area," "these two temperatures are the same," etc. In some cases there may be no change to the P-V or T-s diagram.

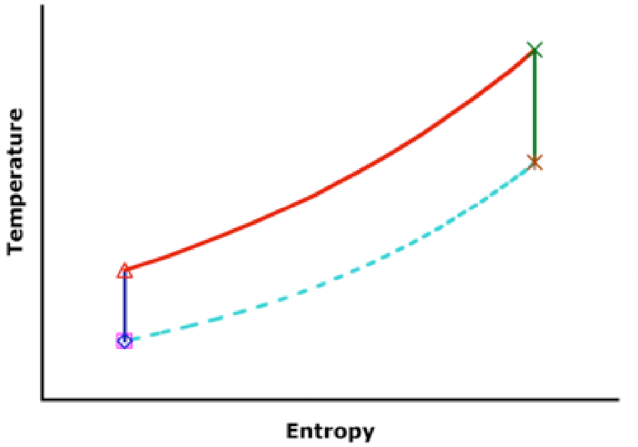
a) The displacement volume is increased by 50% (compression ratio is NOT changed)



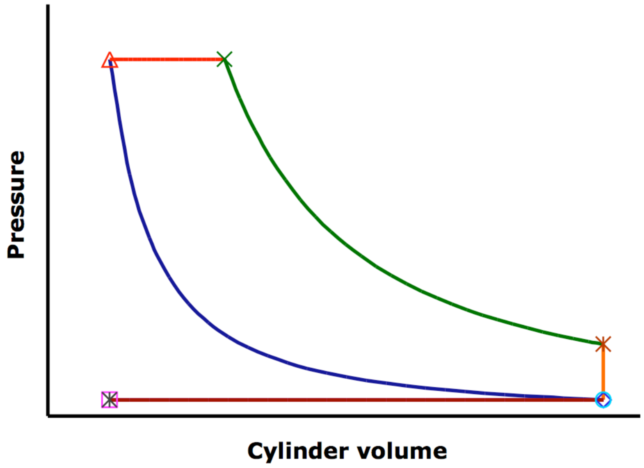


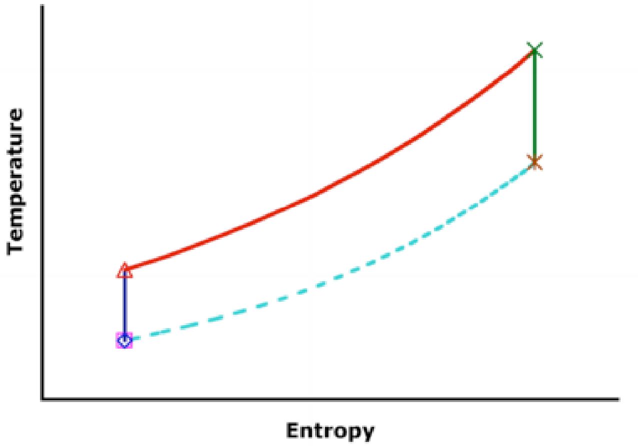
b) The cooling system fails so that the cylinder wall temperature becomes very high and there is heat transfer from the wall to the gas throughout the cycle



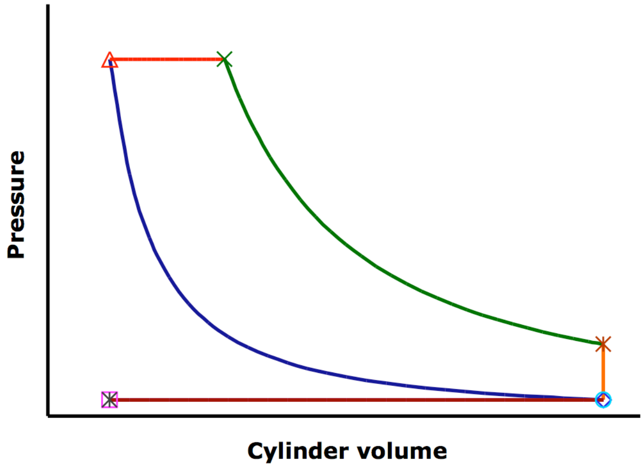


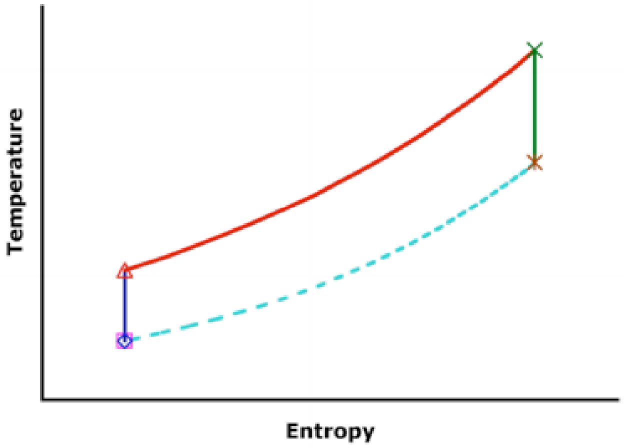
c) The fuel injector malfunctions and injects half of the fuel during the compression stroke; this part of the fuel burns instantaneously (but the other half of the fuel is still injected at the minimum cylinder volume and burns at constant pressure).





d) The ambient temperature (T_2 in my notation) is increased (ambient pressure not changed), but the fuel mass fraction (thus total heat release) is changed so that *the peak temperature is the same for the baseline cycle and the modified cycle.*





<u>Problem #3 (Cycle analysis)</u> (from last year's midterm)

For parts (a) - (d) in problem 2, will the change to the cycle cause the brake thermal efficiency to increase, decrease or remain the same? Explain briefly.

<u>Problem #4 (reciprocating engine performance)</u> (similar to a problem on a previous final exam)

An engine designer claims to have developed a naturally-aspirated (not turbocharged or supercharged) gasoline-fueled 4-stroke engine with a 100 cubic inch displacement volume, compression ratio of 8, operating at 4,000 rpm that produces 300 brake horsepower.

Possibly useful information: $C_P = 1400$ J/kgK; $\gamma = 1.3$; ambient air density 1.18 kg/m^3; Q_R (gasoline) = 4.3 x 10^7 J/kg; stoichiometric fuel mass fraction in air (f) (gasoline) = 0.062.

a) Do you believe this claim? Why or why not? (Hint: your answer should be NO.) Support your answer with calculations.

b) Would increasing compression ratio from 8 to 24 (without changing displacement volume) make the claim of 300 horsepower reasonable? Assume that somehow knocking is not a problem even at this high compression ratio. Again, support your answer with calculations.

c) Would increasing the intake pressure to 3 atm (with compression ratio 8) using a turbocharger make the claim of 300 hp reasonable? Again, support your answer with calculations.

d) Would changing the fuel to hydrogen (with intake pressure 1 atm and compression ratio 8) make the claim of 300 hp reasonable? Again, support your answer with calculations.

Problem #5

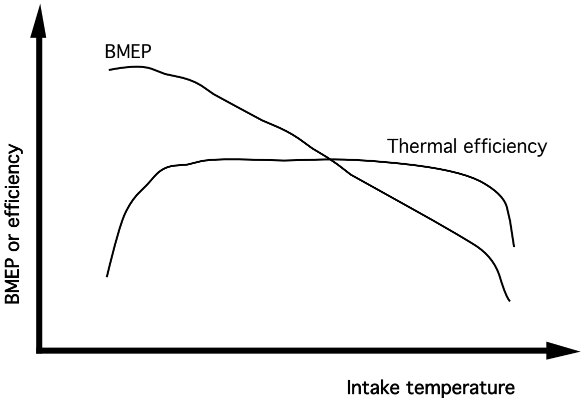
Engine A is a premixed-charge engine with a compression ratio of 10. It burns a stoichiometric mixture (ϕ = 1) at a throttled condition with intake pressure of 0.5 atm. Engine B is a non-premixed charge engine with a compression ratio of 20, having the **same displacement (V_d) and same rotation rate (N) as engine A.** Engine B operates with an intake pressure of 1 atm and an overall equivalence ratio of 0.5. State and **explain in a couple of sentences** which engine will have

- The larger fuel flow rate
- The larger volumetric efficiency
- The larger gross IMEP
- The larger brake power
- The higher thermal efficiency

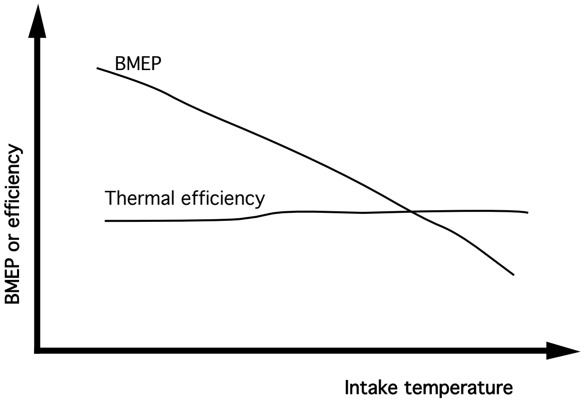
Problem #6

Explain the experimental observations shown in the figure below. For the premixed charge engine, assume that the equivalance ratio ϕ = 1 and that the spark timing is adjusted to that required for maximum power unless knock occurs, in which case the spark is retarded until knocking just stops. For the non-premixed charge engine, assume ϕ (overall) = 0.7 = constant. Hint: consider how intake temperature affects

a) Knock
b) Spark timing required to avoid knock
c) Intake air density
d) Burning velocity
e) Misfire
f) Cutoff ratio β (nonpremixed charge only)



PREMIXED CHARGE



NONPREMIXED CHARGE