2.61 Internal Combustion Engines

Problem Set 4
Tuesday, March 2, 2004

Due: Thursday, March 11, 2004

1.   Hydrogen is a possible future fuel for SI engines, considering both emissions (only water vapor is present in the exhaust) and consumption (high heating value). A disadvantage of hydrogen fuel in SI engines is that the partial pressure of hydrogen in the mixture with air reduces engine's volumetric efficiency, which is proportional to the partial pressure of air.
   (a)   Find the partial pressure of air in the intake manifold downstream of the hydrogen fuel injection location at WOT (the engine is working with stoichiometric mixture).
   (b)   Estimate the ratio of the fuel energy per unit time entering a hydrogen-fueled engine operating with a stoichiometric mixture to the fuel energy per unit time entering an identical gasoline-fueled operating at the same speed with a stoichiometric mixture.
   (c)   How does the power output compare for both engines? What can you do to ensure that the output of each engine is the same? What are your limitations? (for reference see BMW paper: SAE 2003-01-3210, on 2.61 webpage)

2.   A conventional spark-ignition engine operating with gasoline will not run smoothly (due to incomplete combustion) with an equivalence ratio leaner than about $\phi=0.8$. It is desirable to extend the smooth operating limit of the engine to leaner equivalence ratios so that at part throttle operation (with intake pressure less than 1 atmosphere) the pumping work is reduced. Leaner than normal operation can be achieved by adding hydrogen gas (H_2) to the mixture in the intake system. The addition of H2 makes the fuel-air mixture easier to burn.
   (a)   The fuel composition with "mixed" fuel operation is $H_2 + C_8H_{18}$ --- one mole of hydrogen to every mole of gasoline, which is assumed the same as isooctane. What is the stoichiometric air/fuel ratio for the "mixed" fuel?
   (b)   The lower heating value of H2 is 120 MJ/kg and for isooctane is 44.4 MJ/kg. What is the heating value per kilogram of fuel mixture?
   (c)   Engine operation with isooctane and the mixed ($H_2 + C_8H_{18}$) fuel is compared in a particular engine at a part-load condition (brake mean effective pressure of 275 kPa and 1400 rev/min).
   You are given the following information about the engine operation:

Fuel	C_8H_{18}	$H_2 + C_8H_{18}$
Equivalence ratio	0.8	0.5
Gross indicated fuel conversion efficiency	0.35	0.4
Mechanical rubbing friction mep	138 kPa	138 kPa
Inlet manifold pressure	46 kPa	?
Pumping mep	55 kPa	?

   Estimate approximately the inlet manifold pressure and the pumping mean effective pressure with ($H_2 + C_8H_{18}$) fuel. Explain your METHOD and ASSUMPTIONS clearly. Note that mechanical efficiency η_m is defined as

$$\eta_m = bmep/imep_g = bmep/(bmep + rfmep + pmep)$$

3. For four stroke engines, the inlet and exhaust valve opening and closing crank angles are typically:

IVO: 15 CAD BTDC
IVC: 50 CAD ABDC
EVO: 55 CAD BBDC
EVC: 10 CAD ATDC

(a) Explain why these valve timings improve engine breathing relative to valve opening and closing at the beginnings and ends of the intake and exhaust strokes.

(b) Mention at least one modification you would do to the above valve timings if the engine you are designing was
-   intended for a race car
-   turbo-charged
-   working at low speeds

What other additional design issues are important?

4. In a spark-ignition engine, a turbulent flame propagates through the uniform premixed fuel-air mixture within the cylinder and extinguishes at the combustion chamber walls.

(a) Draw a carefully proportioned qualitative graph of cylinder pressure p and mass fraction burned x_b as a function of crank angle θ for $-90° < \theta < 90°$ for a typical SI engine at wide-open throttle with the spark timing adjusted for maximum brake torque. Mark in the crank angles of spark discharge, and of the flame development period (0 to 10 percent burned) and end of combustion, on both p and x_b versus θ curves relative to the top-center crank position.

(b) Estimate approximately the fraction of they cylinder volume occupied by burned gases when the mass fraction burned is 0.5 (i.e., halfway through the burning process).

(c) A simple model for this turbulent flame is shown below in Fig. 1. The rate of burning of the charge dmb/dt is given by

$$\frac{dm_b}{dt} = A_f \rho_u S_T$$

Where A_f is the area of the flame front (which can be approximated by a portion of a cylinder whose axis is at the spark plug position), ρ_u is the unburned mixture density, and S_T is the turbulent flame speed (the speed at which the front moves relative to the unburned mixture ahead of it). The rate of mass burning is influenced therefore by combustion chamber geometry (through A_f) as well as those factors that influence S_T (turbulent intensity, fuel/air ratio, residual gas fraction, and EGR). Compare combustion chambers A and B shown in Fig. 1. Sketch the approximate location of the flame front when 50 percent of the mass has been burned. (A careful qualitative sketch is sufficient; however, provide a quantitative justification for your sketch.) Sketch the mass fraction burned versus crank angle curves for these two combustion chambers on the same graph, each with its spark timing set for maximum brake torque. You may assume the value of S_T is the same for A and B.

(d) Compare combustion chambers A and C in the figure below, which have the same flame travel distance but have different chamber shapes. Which chamber has
-   the faster rate of mass burning during the first half of the combustion process;
-   the faster rate of mass burning during the second half of the combustion process;
-   the more advanced timing for maximum brake torque?
Explain your answers

Figure 1
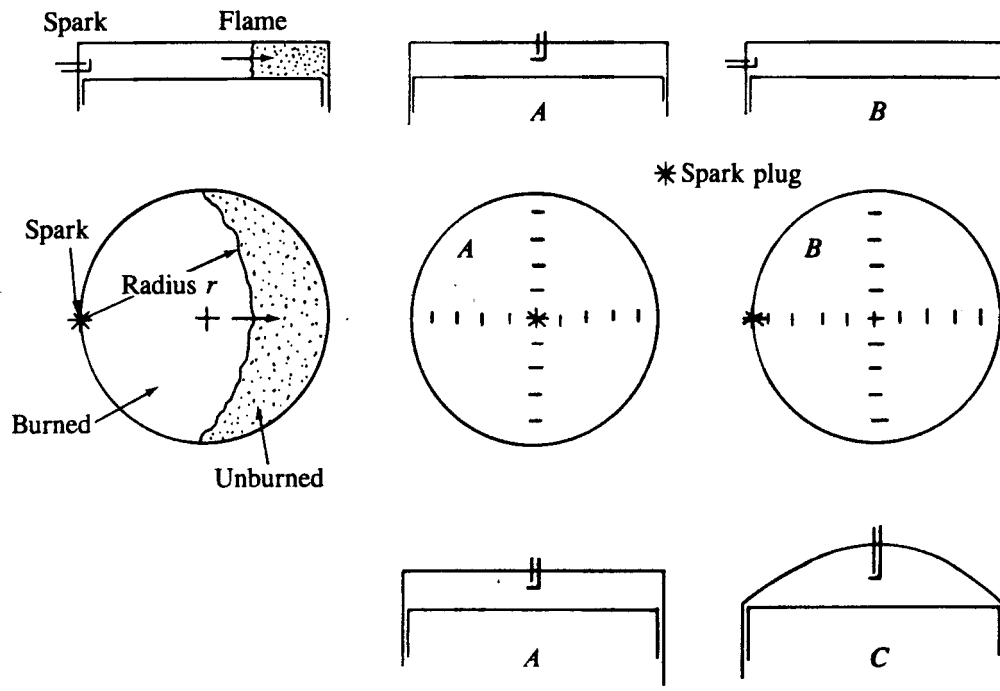
5. An approximate way to calculate the pressure in the end gas (the unburned gas ahead of the flame) just after knock occurs is to assume that all the end gas burns instantaneously at constant volume. We assume that the inertia of the burned gases prevents significant gas motion while the end gas autoignites. Use the data in Figure 9.5 from the text and assume that autoignition occurs at 10 CAD ATDC. You may also need to use the equilibrium charts of Chapter 4 from the text.
   (a) Determine the maximum pressure reached in the end gas after knock occurs
   (b) Estimate the volume occupied by the end gas as a fraction of the cylinder volume just before autoignition occurs

6. Use the GM Vortec 4.2L DOHC engine, which appeared on problem set 2 to calculate the ratio of actual gross indicated engine performance to the equivalent fuel-air cycle predictions in Figures 5-9 and 5-10.
   (a) Maximum power, assume total friction mep = 200 kpa
   (b) Maximum torque, assume total friction mep = 150 kpa
   (c) Best brake specific fuel consumption: the best measured value is 250 g/kWh at part throttle and total friction mep = 140 kpa.
   At WOT the engine operates rich with fuel-air equivalence ratio of 1.2. The engine operates stoichiometric at other load points for emission control reasons. Explain briefly the reasons why the fuel-air cycle predicts better performance than measured