2.61 Final Exam solution, May 2014

Problem 1

Ethanol cooling effects

a) Ideal gas law $MAP \cdot V = \left(\frac{m_a}{W_a} + \frac{m_f}{W_f}\right) RT$ W are the molecular wts; R is universal gas constant

$= m_a \left(\frac{1}{W_a} + \frac{m_f}{m_a}\frac{1}{W_f}\right) RT$ V is volume at IVC

Therefore $m_a = \frac{MAP \cdot V}{RT\left[\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_f}\right]}$

Temperature effect → RT; Fuel vapor displacement effect → $\left[\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_f}\right]$

b) At the same T, $\frac{(m_a)_{E100}}{(m_a)_{E0}} = \frac{\left(\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_f}\right)_{E0}}{\left(\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_f}\right)_{E100}} = \frac{\frac{1}{28.96} + \frac{1}{14.7}\frac{1}{110}}{\frac{1}{28.96} + \frac{1}{9.01}\cdot\frac{1}{46}} = \underline{\underline{0.951}}$

↑ Displacement effect lowers η_v

E_0 : $CH_{1.85} + 1.46(O_2 + 3.773N_2) \rightarrow CO_2 + \frac{1.85}{2}H_2O + 1.46\times3.773\,N_2$; $(A/F)_{stoi} = 14.6$

E_{100} : $C_2H_5OH + 3(O_2 + 3.773N_2) \rightarrow 2CO_2 + 3H_2O + 3\times3.773\,N_2$; $(A/F)_{stoi} = 9.01$

(c) Assume that the fuel does not contribute substantially to charge temp.

Then $m_a c_p \Delta T = x\, m_f\, h_{fg}$

where x is the fraction of fuel vaporized in flight.

$$\Delta T = \frac{x\, m_f\, h_{fg}}{m_a c_p} = \frac{x\, F/A\, h_{fg}}{c_p} = \begin{cases} 0.5\,\frac{1}{9.01}\,\frac{840\times10^3}{1000} = \underline{\underline{46.6\,K}} \text{ for E100} \\ 0.7\,\frac{1}{14.6}\,\frac{305\times10^3}{1000} = \underline{\underline{14.6\,K}} \text{ for E10} \end{cases}$$

(d) $m_a = \frac{MAP \cdot V}{RT\left[\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_f}\right]}$

$$\frac{(m_a)_{E100}}{(m_a)_{E10}} = \frac{\{RT}{\{RT} \frac{\left[\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_F}\right]\}_{E0}}{\left[\frac{1}{W_a} + \frac{F}{A}\frac{1}{W_f}\right\}_{E100}} = \overbrace{\left(\frac{313-14.6}{313-46.6}\right)}^{1.12} \cdot 0.95 = \underline{\underline{1.064}}$$

0.95 from part (b)

1.064 ← 6.4% better η_v

(e) Polytropic:
$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = (80)^{\frac{0.32}{1.32}} = 2.843$$

For E100, $T_1 = 313 - 46.6 = 266.4K$; $T_2 = \underline{\underline{770.7K}}$

For E0, $T_1 = 313 - 14.6 = 298.4K$; $T_2 = \underline{\underline{863.3K}}$

> difference 92.6K

↓

knock less, less NO_x production

Problem 2
Piston crevice knock

(i) Detonation is more severe in the piston crevice gas is because the gas there is denser (due to the lower crevice gas temperature) than the combustion chamber gas. Hence the energy density is significantly higher (by the temperature ratio).

(ii) The detonation of the crevice gas is fast; hence, the release of energy may be considered as instantaneous at constant volume. For ideal gas, applying the first law

$$\dot{E}_{CV} = \dot{Q} - \overset{\text{Const. vol.}}{\cancel{p\dot{V}}} \quad \text{or} \quad \frac{d}{dt}(m c_v T) = \dot{Q} \Rightarrow \frac{\dot{p}V}{\gamma - 1} = \dot{Q}$$

Thus $\dot{p} = (\gamma-1)\dfrac{\dot{Q}}{V} \equiv (\gamma-1)\dot{q}$ where $\dot{q}$ is the volumetric heat release rate

Integrating over the heat release period

$$\Delta p = (\gamma-1)\, q$$

$$q = \rho\left(\frac{1}{1+A/F}\right)(1-x_r)\,LHV = \left(\frac{p}{\frac{R}{W}T}\right)\left(\frac{1-x_r}{1+A/F}\right)LHV$$

↑ crevice temperature

↑ m.w. of unburned mixture

Thus $$\Delta p = (\gamma-1)\left(\frac{p}{\frac{R}{W}T}\right)\left(\frac{1-x_r}{1+A/F}\right)LHV$$

For $A/F = 14.6$, $p = 40$ bar, $W = 29$, $\gamma = 1.32$, $T = 500$K, $x_r = 0$

$$\Delta p = (1.32-1)\left(\frac{40\times10^5}{\frac{8314}{29}\times 500}\right)\frac{1}{1+14.6}\times 44\times10^6 = \underline{\underline{252 \text{ bar}}}$$

$$p = (40+252)\text{bar} = \underline{\underline{292 \text{ bar}}} \quad \text{Very High pressure!}$$

Note that the pressure is independent of the crevice geometry but the total energy from the detonation is

Problem 3
Two Stroke Opposed Piston Engine

Comparing the two-stroke opposed piston engine with the 4 stroke engine:

1. It is a two-stroke, therefore, the power density is higher (theoretically twice; but lower in practice because of incomplete scavenging).
2. For a two-stroke engine, the scavenging of the engine would lead to air feed through. Therefore, a three way catalyst cannot be used.
3. The air feed through is also a problem for diesel operation because it reduces the exhaust temperature and affect the after-treatment operation.
4. To prevent fuel feed through, direct injection is needed.
5. The combustion geometry is not favorable for air fuel mixing (for both spark and diesel configurations) since the injector has to be mounted on the side of the combustion chamber. Having more than one injector would be expensive.
6. If the engine is a spark ignition engine, the flame propagation geometry is not favorable because it is started with the spark plug mounted on the side.
7. No valve train is needed: substantially lower friction, lower cost, and lower weight to power ratio.
8. Lower heat transfer because there is no “head” surface.
9. Opposed piston configuration leads to better engine balancing and reduces noise-vibration-harshness.
10. The ring pack of one of the two pistons has to side over the hot exhaust port. The process leads to lubrication and durability problems.

The concept has been around for a long time. People recognize the benefits (1, 7, 8 and 9). It has been implemented in some engines; historically the large ones. However (2) renders its application mainly to diesels, and (10) is a significant hurdle.

(1) and (7) make the configuration attractive as engines for UAVs.

Problem 4

Engine downsizing

(a) $\mathrm{BMEP}=\frac{4\pi\Gamma}{V_D}$ where Γ is the torque and V_D is the displacement. Thus at Γ = 175 N-m:

$$\mathrm{BMEP}_a=\frac{4\pi\ 175}{2\mathrm{x}10^{-3}}=11\ \mathrm{bar};\qquad \mathrm{BMEP}_b=\frac{4\pi\ 175}{1.2\mathrm{x}10^{-3}}=18.3\ \mathrm{bar};$$

(b) The BMEP value is proportional to the charge density at the intake manifold. For the same manifold temperature, the boost is proportional to the density ratio. Thus the pressure ratio of the compressor is

$$\text{Compressor pressure ratio}=\frac{18.3}{11}=1.66$$

(c) The heat transfer per cycle Q is equal to the heat transfer rate times the time available. In the following derivation, B is the bore; A is the combustion chamber area and is proportional to B^2; the stroke L is proportional to B; N is the revolution per second

$$Q\sim \mathrm{Nu}\left(\frac{\kappa\Delta T}{B}\right)A\tau$$

The Nusselt number scales as $\left(\frac{\rho(2NL)B}{\mu}\right)^{0.8}\propto\left(\rho NB^2\right)^{0.8}$; the available time $\tau\approx\frac{1}{N}$

The charge temperature is mainly a function of fuel air ratio and compression ratio, and is the same for the two engine configurations.

Thus $Q\propto\left(\rho NB^2\right)^{0.8}\frac{1}{B}B^2\frac{1}{N}=\rho^{0.8}N^{-0.2}B^{2.6}$

Comparing engine B to engine A:

$$\frac{\rho_b}{\rho_a}=1.66;\qquad \frac{B_b}{B_a}=\left(\frac{1.2}{2}\right)^{\frac{1}{3}}=0.84;$$

$$\text{Thus }\frac{Q_b}{Q_a}=\left(1.66\right)^{0.8}\left(0.84\right)^{2.6}=\left(1.5\right)\left(0.64\right)=0.96$$

Note the significant size reduction effect.

(d) The friction force is equal to the normal force F_n times the friction coefficient, which is proportional to the square root of the Sommerfeld number. For pistons that are geometrically scaled, the normal force is proportional to pB^2, thus:

$F_f\propto pB^2\sqrt{\frac{\mu N}{p}}$; since p is proportional to the manifold air density ρ, $F_f\propto B^2\sqrt{\rho N}$.

For simplicity, the oil viscosity μ is assumed to be the same. In practice μ is a function of temperature, which changes with BMEP.

Since work is force x distance travelled, the ratio of the piston friction work of engine B to engine A is

$$\frac{\left(W_F\right)_b}{\left(W_F\right)_a}=\frac{B\left\{B^2\sqrt{\rho N}\right\}_{\text{Engine B}}}{B\left\{B^2\sqrt{\rho N}\right\}_{\text{Engine A}}}=\frac{B_b^3}{B_a^3}\sqrt{\frac{\rho_b}{\rho_a}}=0.84^3\sqrt{1.66}=0.77$$

The ratio shows a significant reduction due to the size effect.

(e) The brake fuel conversion efficiency is

$$\eta_f=\frac{\mathrm{BMEP}*V_D}{m_f\mathrm{LHV}}=\frac{W_{i,g}}{m_f\mathrm{LHV}}-\frac{W_F}{m_f\mathrm{LHV}}-\frac{W_P}{m_f\mathrm{LHV}}-\frac{\text{Other losses}}{m_f\mathrm{LHV}}$$

The change in the gross indicated work $\Delta W_{i,g}$ is due to the change in heat transfer, -ΔQ. Therefore, the change of η_f due to change of heat transfer and mechanical friction is

$$\Delta\eta_f \approx -\frac{Q_a}{m_f LHV}\left(\frac{Q_b}{Q_a}-1\right)-\frac{(W_F)_a}{m_f LHV}\left(\frac{(W_F)_b}{(W_F)_a}-1\right)$$
$$= -(0.22)(0.96-1)-(0.035)(0.77-1)$$
$$= 0.88\% \qquad + \qquad 0.81\%$$
$$= 1.69\%$$

The gain of 1.69 percentage points may seem modest, however, since η_f at peak torque is about 33%, the improvement is 1.69/33 = 4.9%, a significant gain. It will be seen below that the gain at part load is even more because of the reduced pumping loss, and that the heat transfer and friction losses are a bigger fraction of the fuel energy.

(f) $BMEP_a = \frac{4\pi\ 50}{2x10^{-3}} = 3.14$ bar; $\qquad BMEP_b = \frac{4\pi\ 50}{1.2x10^{-3}} = 5.24$ bar;

(g) The BMEP value is proportional to the charge density at the intake manifold. For the same manifold temperature, the MAP is proportional to the density. Thus the MAP ratio is

$$\frac{(MAP)_b}{(MAP)_a} = \frac{5.24}{3.14} = 1.66$$

(h) Pumping work

$$W_P \approx (P_e - P_i)V_D$$
$$(W_P)_a \approx (1-0.35)x10^5 x2x10^{-3} = 130J$$

Engine B has MAP=0.35x1.66=0.58

$$(W_P)_b \approx (1-0.58)x10^5 x1.2x10^{-3} = 50.4J$$

(i) Same answer as in part (c). Note that the absolute values are different, but the ratio is the same since the density ratio ρ_b/ρ_a for the part load case is the same as that for the peak torque case.

(j) Same answer as in part (d).

(k) The brake fuel conversion efficiency is

$$\eta_f = \frac{BMEP * V_D}{m_f LHV} = \frac{W_{i,g}}{m_f LHV} - \frac{W_F}{m_f LHV} - \frac{W_P}{m_f LHV} - \frac{\text{Other losses}}{m_f LHV}$$

The change in the gross indicated work $\Delta W_{i,g}$ is due to the change in heat transfer, $-\Delta Q$. Therefore, the change of η_f due to change of heat transfer, mechanical friction, and pumping loss is

$$\Delta\eta_f \approx \frac{-\Delta Q}{m_f LHV} - \frac{\Delta W_F}{m_f LHV} - \frac{\Delta W_P}{m_f LHV}$$
$$= -\frac{Q_a}{m_f LHV}\left(\frac{Q_b}{Q_a}-1\right)-\frac{(W_F)_a}{m_f LHV}\left(\frac{(W_F)_b}{(W_F)_a}-1\right)-\frac{(W_P)_a}{m_f LHV}\left(\frac{(W_P)_b}{(W_P)_a}-1\right)$$
$$= -(0.28)(0.96-1)-(0.065)(0.77-1)-(0.04)\left(\frac{50.4}{130}-1\right) \qquad .$$
$$= 1.12\% \qquad +1.50\% \qquad +2.45\%$$
$$= 5.07\%$$

The gain of 5 percentage points is significant. At part load, the fuel conversion efficiency is about 20%. Thus, the improvement of specific fuel consumption is 5.07/20 = 25%