2.61 Internal Combustion Engine
Final examination solution, May 2017

Problem 1: ethanol cooling effect

W are the molecular wts
R is universal gas constant
V is volume at IVC

a) Ideal gas law $MAP \cdot V = \left(\frac{m_a}{W_a} + \frac{m_f}{W_f}\right) RT$

$= m_a \left(\frac{1}{W_a} + \frac{m_f}{m_a}\frac{1}{W_f}\right) RT$

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

Temperature effect — Fuel vapor displacement effect

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

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.

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

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

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

6.4% better η_v

(e) Polytropic:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = (80)^{\frac{0.32}{1.32}} = 2.893$$

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

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

difference of 92.6K

↓

knock less, less NO_x production

Problem 2 soluiton

Scaling of losses

(a) Heat transfer per cycle

surface area $\propto B^2$

$$Q = Nu \cdot \frac{k\Delta T}{B} \cdot A \cdot \frac{1}{N} \propto Nu\, B \cdot \frac{1}{N}$$

$L \propto B$

$$Nu \propto Re^{0.8} = \left(\frac{\rho \cdot 2NL \cdot B}{\mu}\right)^{0.8} \propto (\rho N B^2)^{0.8}$$

Thus $Q \propto (\rho N B^2)^{0.8} \frac{B}{N}$

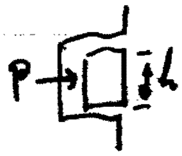
work done/cycle

$$Torq = \frac{\eta_f m_f LHV}{4\pi} = \frac{\eta_f m\left(\frac{1}{1+A/F}\right) LHV}{4\pi} \propto m \propto \rho V_D \propto \rho B^3$$

So for the same torque, $\rho \propto 1/B^3$

Thus $Q \propto \left(\frac{1}{B^3} \cdot B^2\right)^{0.8} \cdot B \propto B^{0.2} \Rightarrow Q \propto B^{0.2}$

(b) Ring friction



$F_f = F_N f$; $F_N = p\pi B h$; $f \propto \sqrt{S_R}$

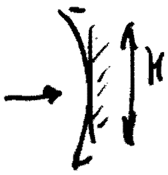
$S_R = \frac{\mu U}{\sigma_R h} = \frac{\mu (2NL)}{\sigma_R h}$; $L \propto B$; $\sigma_R = p \Rightarrow S_R \propto \sqrt{\frac{B}{p}}$

Thus $F_f \propto pB\sqrt{\frac{B}{p}}$

For the same torque output $p \propto \frac{1}{V_D} \propto \frac{1}{B^3}$ Thus $F_f \propto \sqrt{p}\, B^{3/2}$ is independent of B.

Friction work done $\propto F_f \cdot B$ Thus $W_{\text{Ring friction}} \propto B$

(c) Skirt friction



gas force on piston

$F_f = F_N f$; $F_N \propto pB^2$

$f \propto \sqrt{S_s}$; $S_s = \frac{\mu U}{\sigma_s H}$; $U \propto B$, $H \propto B$; $\sigma_s \propto \frac{pB^2}{\pi B H} \propto p$

Thus $S_s \propto \frac{1}{p}$

Skirt friction work

$F_f \propto pB^2 \frac{1}{\sqrt{p}} \propto p^{1/2}B^2$; Same torq $p \propto \frac{1}{B^3} \Rightarrow F_f \propto \frac{1}{B^{3/2}} B^2 \propto B^{1/2}$; $W \propto F_f B \Rightarrow W \propto B^{3/2}$

Note that since for the same torque output, if the engine efficiency does not change much, the amount of fuel used is approximately the same. Then the losses (a), (b) and (c) go down with the engine size B.

Problem 3 solution

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

Const. vol.

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

Thus $\dot{p} = (\gamma-1)\frac{\dot{Q}}{V} \equiv (\gamma-1)\dot{q}$ 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, $x_r = 0$

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

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

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

Problem 4: Split cycle engine

Advantages:

i. The splitting of the intake/compression and expansion/exhaust strokes to be performed by two independent cylinders enables one to have a higher expansion ratio ε than compression ratio. The high expansion ratio gives higher fuel conversion efficiency. Note that the bore of the expansion cylinder could be larger than that of the compression one; hence larger ε can be made large without going to longer stroke. This feature makes the packaging of the engine easier (do not have to have a tall engine).
ii. Lower compression ratio (at the same expansion ratio) allows lower NOx emissions.
iii. Lower compression ratio also prevents knocking.
iv. The transfer of the charge into the expansion cylinder is just before ignition. Hence the turbulence generated by the fluid motion has little time to decay. The high turbulence promotes combustion speed.

Disadvantages

v. Substantial pumping loss in the valves and crossover passage in the charge transfer process. The loss is especially severe at high speed and high load.
vi. Ditto for heat transfer loss.
vii. The air intake is determined by the sweep volume of the intake/compression cylinder. Have to turbo-charge to increase power.
viii. The valve stem seal for the two transfer valves must sustain the compression pressure and temperature. The tradeoff between leakage and friction is difficult.
ix. Because ignition is in the expansion cylinder, ignition timing is essentially at or after TDC. Since the heat release is not instantaneous, the later combustion partly negates the effect of (ii) and efficiency suffers.
x. If the fuel is injected in the crossover passage, there is not sufficient time for evaporation and mixing. So the charge will be inhomogeneous. There will be soot formation in the locally fuel rich region.

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