14 Experimental Facilities

14.1 Experiments are being conducted on a 3.9 litre 4 stroke diesel engine, which is being operated with three different inlet conditions:

i) naturally aspirated with air at an inlet temperature of 30°C

ii) naturally aspirated with air at an inlet temperature of 150°C

iii) naturally aspirated, but with a mixture of 30% O₂ and 70% CO₂ (molar basis) at an inlet temperature of 150°C.

At a speed of 2000 rpm in mode (iii), the following data were obtained:

<table>
<thead>
<tr>
<th></th>
<th>Mode (ii)</th>
<th>Mode (iii)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake power (kW)</td>
<td>16.2</td>
<td>33.2</td>
</tr>
<tr>
<td>bsfc (g/kWh)</td>
<td>405.9</td>
<td>345.0</td>
</tr>
<tr>
<td>Oxygen level (%)</td>
<td>18.0</td>
<td>7.8</td>
</tr>
</tbody>
</table>

At a bmep of 2.6 bar, the engine performance for cases (ii) and (iii) are:

<table>
<thead>
<tr>
<th></th>
<th>Mode (ii)</th>
<th>Mode (iii)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure at end of compression (bar)</td>
<td>38</td>
<td>31</td>
</tr>
<tr>
<td>Effective phasing of combustion (°atdc)</td>
<td>0</td>
<td>12</td>
</tr>
</tbody>
</table>

a) Use the Willans’ line construction to estimate the fmep of the engine.

b) Stating clearly any assumptions, calculate the volumetric efficiency of the engine, and comment on the result.

Assume that the effective compression ratio of the engine is 14:1 (instead of the nominal value of 15.5:1, to allow for the inlet valves closing slightly after bottom dead centre).

c) If the pressure at the start of compression is 1 bar, calculate the value of the polytropic index, and compare this with the value for the ratio of the specific heat capacities (use the data from the tables in Appendix A).

d) By assuming that the combustion can be treated as instantaneous at the ‘effective phasing of combustion’, use an Otto cycle analysis to apportion the reduction in cycle efficiency form case (i) to case (ii), between the effects of: the lower value of the ratio of the heat capacities, and (ii) the increased ignition delay.

a)

To construct the Willans’ line, we need to calculate the bmep and fuel mass flow rate (\(m_f\)) for each of the operating points. The bmep is found from a rearrangement of equation 2.17:

\[
bmep = \frac{\text{brake power}}{(V_s \times N^*)}
\]

where: \(V_s\) = swept volume, and
\(N^*\) = number of times the swept volume is exchanged per second, remembering that this is a four-stroke engine, in which the volume is exchanged in alternate revolutions.

\[
V_s \times N^* = 3.9 \times 10^{-3} \times 2000/120 = 0.065 \text{ m}^3/\text{s}
\]

The fuel mass flowrate is the product of the brake power (\(W_b\)) and the brake specific fuel consumption (bsfc):

\[
m_f = W_b \times \text{bsfc}
\]
Load I   Load II
\[\begin{array}{lll}
\text{Brake power} & 16.2 & 33.2 \\
\text{bsfc} & 405.9 & 345.0 \\
\text{bmep} & 2.49 & 5.11 \\
\text{fuel mass flow rate (m)} & 6.58 & 11.45 \\
\end{array}\]

\[\begin{align*}
\text{fuel flow rate (kg/h)} & \\
15.0 & 12.5 & 10.0 & 7.5 & 5.0 & 2.5 & 0 & 2 & 1 \\
\end{align*}\]

\[\begin{align*}
\text{fmeep (bar)} & \\
\text{bmep (bar)} & \\
\end{align*}\]

The straight line can be fitted by:
\[y = mx + c\]

For the two data points:
\[y_1 = mx_1 + c\text{ and } y_2 = mx_2 + c\]

so \[m = (y_1 - y_2)/(x_1 - x_2)\]

\[m = (11.45 - 6.58)/(5.11 - 2.49) = 1.859\]

so \[c = y_1 - mx_1 = 11.45 - 1.859\times5.11\]

and \[c = 1.95.\]

\[\text{Fmep is when } y = 0\]

\[\text{fmeep} = -c/m = -1.95/1.859\]

\[\text{fmeep} = -\text{bmeep} = 1.05 \text{ bar}\]

b)

To calculate the volumetric efficiency, we need to assume complete combustion, and that the fuel composition can be represented by \(\text{C}_x\text{H}_{1.8}x\). We can then calculate the gravimetric 'air' fuel ratio, and as we have already calculated the fuel mass flow rate, this gives us the 'air' mass flow rate, from which we can calculate the volumetric efficiency. 'Air' is in quotes because its molar composition is 30% \(\text{O}_2\) and 70% \(\text{CO}_2\).

The generalised combustion equation, for 1 kmol of fuel is:
\[\text{C}_x\text{H}_{1.8}x + zx(0.3 \text{ O}_2 + 0.70 \text{ CO}_2) \rightarrow x(1 + 0.7z)\text{CO}_2 + 0.9x\text{H}_2\text{O} + x(0.3z - 1.45)\text{O}_2\]

In a dry gas analysis, the mole fraction of oxygen in the exhaust \((a)\) is:
\[a = x(0.3z - 1.45)/[(x(1+ 0.7z) + x(0.3z - 1.45)]\]

or \[a = (0.3z - 1.45)/(z - 0.45)\]

or \[az - 0.45a = 0.3z - 1.45\]

or \[z = (1.45 - 0.45a)/(0.3 - a)\]

Since the 'air' is measured on a volumetric basis, while the fuel consumption has been measured on a mass basis, we will use the following unconventional definition of AFR:

The 'air' fuel ratio (kmols of gas/kg of fuel): \[zx(0.3 + 0.7) : x(M_c + 1.8M_H)\]

or \[z : (12 + 1.8 \times 1) \text{ or } z/13.8 : 1\]
For Load I: \( a = 0.18 \), so \( z = (1.45 - 0.45 \times 0.18)/(0.3 - 0.18) = 11.41 \), and AFR = \( z/13.8 = 0.827 \).

the molar flow rate of 'air', \( n_{a,I} = m_f \times AFR = 6.58 \times 0.827 = 5.44 \text{ kmol/h or } 1.51 \text{ g mol/s} \)

For Load II: \( a = 0.078 \), so \( z = (1.45 - 0.45 \times 0.078)/(0.3 - 0.078) = 6.37 \), and AFR = \( z/13.8 = 0.462 \).

the molar flow rate of 'air', \( n_{a,II} = m_f \times AFR = 11.45 \times 0.462 = 5.29 \text{ kmol/h or } 1.47 \text{ g mol/s} \)

As would be expected in a diesel engine, the molar (and thus volume) flowrate into the engine has decreased slightly as the engine load has increased.

The volume flow rate of 'air', 
\[ V = n_a R_o T/p \]

and the volumetric efficiency, 
\[ \eta_{vol} = \frac{V}{(V_s x N/120)} = \frac{120 n_a R_o T/(p x V_s x N/120)}{x} \]

so:
\[ \eta_{vol,I} = 120 \times 1.51 \times 10^{-3} \times 8314.3 \times (273.15 + 150)/(10^5 \times 3.9 \times 10^{-3} \times 2000) = 0.817 \]
\[ \eta_{vol,II} = 120 \times 1.47 \times 10^{-3} \times 8314.3 \times (273.15 + 150)/(10^5 \times 3.9 \times 10^{-3} \times 2000) = 0.796 \]

c) The pressure at the end of compression is solely due to piston motion, since in both cases (ii) and (iii) the combustion does not commence until after top dead centre.

For a polytropic process, 
\[ pV^n = \text{const.} \]

Taking logs and rearranging gives:
\[ n = \ln(p_2/p_1)/\ln(V_1/V_2) \]

For case (ii), 
\[ n = \ln(38)/\ln(14) = 1.38, \]

for case (iii), 
\[ n = \ln(31)/\ln(14) = 1.30 \]

To estimate the mean value of the ratio of specific heat capacities, we need to know the temperature range. If we assume the start of compression to be at 150°C, then we can apply the equation of state (assuming no leakage) to estimate the temperature at the end of compression.

\[ p_1 V_1/T_1 = p_2 V_2/T_2 \]
\[ T_2 = p_2 V_2 T_1/(p_1 V_1) \]

For case (ii), 
\[ T_2 = 38 \times (150 + 273.15)/14 = 1149 \text{ K}; \]

for case (iii), 
\[ T_2 = 31 \times (150 + 273.15)/14 = 937 \text{ K}. \]

Both these temperatures are rather high (since the 'air' will be cooled during the induction process), and because of the approximations in their evaluation, it will be sufficient to approximate the temperature ranges over which the ratio of the heat capacities is evaluated.

For case (ii) air, 400 to 1100 K:
\[ C_p = (H_{1100} - H_{400})(1100 - 400) = (24.984 - 2.973)/700 = 31.44 \text{ kJ/kmolK} \]

\[ C_v = C_p - R_o = 31.44 - 8.3143 = 23.130 \text{ kJ/kmolK} \]

\[ \gamma = C_p/C_v = 31.44 - 22.13 = 1.36 \] (compared to 1.38 for the polytropic index)

For case (iii), 30% O\(_2\) and 70% CO\(_2\), 400 to 900 K:
\[ C_p = 0.3 H_{900,O2} + 0.7 H_{900,CO2} - 0.3 H_{900,CO2} - 0.7 H_{900,CO2}(900 - 400) \]
\[ = (0.3 \times 19.244 + 0.7 \times [-365.477] - 0.3 \times 3.028 - [-389.509])/500 = 43.37 \text{ kJ/kmolK} \]

\[ C_v = C_p - R_o = 43.37 - 8.3143 = 35.06 \text{ kJ/kmolK} \]

\[ \gamma = C_p/C_v = 43.37 - 35.06 = 1.24 \] (compared to 1.30 for the polytropic index)

Since the polytropic index is higher than the ratio of heat capacities, then this suggests that the estimate of the effective compression ratio is low.
d)

With a compression ratio \( (r_v) \) of 14:

Otto cycle efficiency for case (ii) with \( \gamma = 1.36 \) is:

\[
1 - \frac{1}{r_v^{\gamma - 1}} = 0.613
\]

Otto cycle efficiency for case (iii) with \( \gamma = 1.24 \) is:

\[
1 - \frac{1}{r_v^{\gamma - 1}} = 0.469
\]

If the combustion is phased 12°-atdc, and we assume simple harmonic motion for the piston, then the clearance volume has increased by:

\[
V_c(1 - \cos 12°)/2 = 0.011V_s
\]

the clearance volume was originally \( V_c/(r_v - 1) = 0.077V_s \), so the new clearance volume is \( 0.088V_s \)

The start of the expansion stroke is delayed, so the effective swept volume is \( 0.989V_s \)

The compression ratio with delayed combustion is thus: \( 1 + 0.989/0.088 = 12.2 \).

The Otto cycle efficiency for case (iii) is now:

\[
1 - \frac{1}{12.2^{\gamma - 1}} = 0.451
\]

The Otto cycle efficiency is thus lowered 14.4 %points by the reduced ratio of heat capacities, and by a further 1.8 %points by the delayed combustion.

This question has been based on experimental data in 'The Effects of Non-Air Mixtures on the Operation of a Diesel Engine by Experiment and Simulation', by J G Hawley, S J Ashcroft and M A Patrick, *Proc Instrn Mech Engrs*, Vol 212 No A1, pp 55-68, 1998. The pressure volume diagrams show quite clearly the loss of work associated with the delayed combustion in case (iii).