DISCLAIMER
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1. A pickup truck has a five-liter, V6, SI engine operating at 2400 RPM. The compression ratio $r_c = 10.2:1$, the volumetric efficiency $\eta_{TV} = 0.91$, and the bore and stroke are related as stroke $S = 0.92 B$. Calculate: (a) Stroke length. [cm] (b) Average piston speed. [m/sec] (c) Clearance volume of one cylinder. [cm$^3$] (d) Air flow rate into engine. [kg/sec]

2. A construction vehicle has a diesel engine with eight cylinders of 5.375-inch bore and 8.0-inch stroke, operating on a four-stroke cycle. It delivers 152-shaft horsepower at 1000 RPM, with a mechanical efficiency of 0.60. Calculate: (a) Total engine displacement. [in.$^3$] (b) Brake mean effective pressure. [psia] (c) Torque at 1000RPM. [lbf-ft] (d) Indicated horsepower. (e) Friction horsepower.

3. Methanol is burned in an engine with air at an equivalence ratio of $\varphi = 0.75$. Exhaust pressure and inlet pressure are 101 kPa. Write the balanced chemical equation for this reaction. Calculate: (a) Air-fuel ratio. (b) Dew point temperature of the exhaust if the inlet air is dry. [0°C] (c) Dew point temperature of the exhaust if the inlet air has a relative humidity of 40% at 25°C. [0°C] (d) Antiknock index of methanol.

4. Compute the indicated power generated at WOT by a three-liter, four-cylinder, four stroke cycle SI engine operating at 4800 RPM using either gasoline or methanol. For each case, the intake manifold is heated such that all fuel is evaporated before the intake ports, and the air-fuel mixture enters the cylinders at 60°C and 100 kPa. Compression ratio $r_c = 8.5$, fuel equivalence ratio $\varphi = 1.0$, combustion efficiency $\eta_T/c = 98\%$, and volumetric efficiency $\eta_{TV} = 100\%$. Calculate the indicated specific fuel consumption for each fuel. [gm/kW-hr]

5. A six-cylinder, four-stroke cycle SI engine with multipoint fuel injection has a displacement of 204 liters and a volumetric efficiency of 87% at 3000 RPM, and operates on ethyl alcohol with an equivalence ratio of 1.06. Each cylinder has one port injector which delivers fuel at a rate of 0.02 kg/sec. The engine also has an auxiliary injector upstream in the intake manifold which delivers fuel at a rate of 0.003 kg/sec to change the air-fuel ratio and give a richer mixture when needed. When in use, the auxiliary injector operates continuously and supplies all cylinders. Calculate: (a) Time of one injection pulse for one cylinder for one cycle. [sec] (b) AF if the auxiliary injector is not being used. (c) AF if the auxiliary injector is being used.

6. A 3.6-liter, V6 SI engine is designed to have a maximum speed of 7000 RPM. There are two intake valves per cylinder, and valve lift equals one-fourth valve diameter. Bore and stroke are related as $S = 1.06B$. Design temperature of the air-fuel mixture entering the cylinders is 60°C. Calculate! (a) Ideal theoretical valve diameter. [cm]. (b) Maximum flow velocity through intake valve. [m/sec] (c) Do the valve diameters and bore size seem compatible?

7. A 6.8-liter, in-line, eight-cylinder CI engine has a compression ratio $r_c = 18.5$ and a crevice volume equal to 3% of the clearance volume. During the engine cycle pressure ill. The
crevice volume equals combustion chamber pressure while remaining at the cylinder wall
temperature of 190°C. Cylinder conditions at the start of compression are 75°C and 120kPa,
and peak pressure is 11,000kPa. Cutoff ratio is \( f_3 = 2.3 \). Calculate: (a) Crevice volume of
one cylinder. \([\text{em}^3]\) (b) Percent of air-fuel mixture in the crevice volume at the end of
compression. \([\%]\) (c) Percent of air-fuel mixture in the crevice volume at the end of
combustion. \([\%]\)

8. A 2.6-liter, four-cylinder, stratified charge SI engine with a compression ratio of 10.5:1
operates on an Otto cycle. The engine has divided combustion chambers, with a secondary
chamber containing 18% of the clearance volume in each cylinder. A 1-cm² orifice
connects the secondary chamber with the main combustion chamber. AF = 13.2 in the
secondary chamber where the spark plug is located, and AF = 20.8 in the main chamber.
The fuel is gasoline with a 98% combustion efficiency. When operating at 2600 RPM, the
conditions in both chambers at the start of combustion are 700 K and 2100 kPa.
Combustion can be modeled as an instantaneous heat addition in the secondary chamber,
followed by a gas expansion into the main chamber which lasts for about 7°of engine
rotation. Additional heat is then added from combustion in the main chamber. Calculate: (a)
Overall AF. (b) Peak temperature and pressure in the secondary chamber. \( r^\circ C, \text{kPa} \) (c)
Approximate velocity of gas flow into the main chamber immediately after combustion in
the secondary chamber. \([\text{m/sec}]\)

9. A CI engine with a 3.2-inch bore and 3.9-inch stroke operates at 1850 RPM. In each cycle,
fuel injection starts at 16°bTDC and lasts for 0.0019 second. Combustion starts at 8° bTDC.
Due to the higher temperature, the ignition delay of any fuel injected after combustion starts
is reduced by a factor of two from the original ID. Calculate: (a) ID of first fuel injected.
[see] (b) ID of first fuel injected in degrees of engine rotation. (c) Crank angle position
when combustion starts on last fuel droplets injected.

10. The engine in Problem 7-7 has a volumetric efficiency of 92%, an overall combustion
efficiency of 99%, an indicated thermal efficiency of 52%, and a mechanical efficiency of
86% when operating at 3500 RPM. Calculate: (a) Brake power at this condition. \([\text{KW}]\) (b)
bmep. \([\text{kPa}]\) (c) Amount of unburned fuel exhausted from the engine.\([\text{kg/hr}]\) (d) bsfc.
\([\text{g/mkW-hr}]\)
SOLUTION

QUESTION 1:
John’s automobile has a three-liter SI V6 engine that operates on a four-stroke cycle at 3600 RPM. The compression ratio is 9.5, the length of connecting rods is 16.6 cm, and the engine is square ($B = S$). At this speed, combustion ends at 20° aTDC. Calculate:

1. cylinder bore and stroke length
2. average piston speed
3. clearance volume of one cylinder
4. piston speed at the end of combustion
5. distance the piston has traveled from TDC at the end of combustion
6. volume in the combustion chamber at the end of combustion

1) For one cylinder, using Eq. (2-8) with $S = B$:
$$V_d = V_{total}/6 = 3L/6 = 0.5 L = 0.0005 \text{ m}^3 = (\pi/4)B^3S = (\pi/4)B^3$$
$$B = 0.0860 \text{ m} = 8.60 \text{ cm} = S$$

2) Using Eq. (2-2) to find average piston speed:
$$\bar{U}_p = 2SN = (2 \text{ strokes/rev})(0.0860 \text{ m/stroke})(3600/60 \text{ rev/sec})$$
$$= 10.32 \text{ m/sec}$$

3) Using Eq. (2-12) to find the clearance volume of one cylinder:
$$r_c = 9.5 = (V_d + V_c)/V_c = (0.0005 + V_c)/V_c$$
$$V_c = 0.000059 \text{ m}^3 = 59 \text{ cm}^3$$

4) Crank offset, $a = S/2 = 0.0430 \text{ m} = 4.30 \text{ cm}$
$$R = r/a = 16.6 \text{ cm}/4.30 \text{ cm} = 3.86$$
Using Eq. (2-5) to find instantaneous piston speed:
\[
\frac{U_p}{\bar{U}_p} = \frac{(\pi/2) \sin \theta [1 + (\cos \theta / \sqrt{R^2 - \sin^2 \theta})]}{(\pi/2) \sin (20^\circ) [1 + (\cos (20^\circ) / \sqrt{3.86^2 - \sin^2 (20^\circ)})]}
\]
\[
= 0.668
\]
\[
U_p = 0.668 \bar{U}_p = (0.668)(10.32 \text{ m/sec}) = 6.89 \text{ m/sec}
\]

5) Using Eq. (2-3) to find piston position:
\[
s = a \cos \theta + \sqrt{r^2 - a^2 \sin^2 \theta}
\]
\[
= (0.0430 \text{ m}) \cos (20^\circ) + \sqrt{(0.166 \text{ m})^2 - (0.0430 \text{ m})^2 \sin^2 (20^\circ)}
\]
\[
= 0.206 \text{ m}
\]

Distance from TDC:
\[
x = r + a - s = (0.166 \text{ m}) + (0.043 \text{ m}) - (0.206 \text{ m})
\]
\[
= 0.003 \text{ m} = 0.3 \text{ cm}
\]

6) Using Eq. (2-14) to find instantaneous volume:
\[
\frac{V}{V_c} = 1 + \frac{1}{2} (r_c - 1) [R + 1 - \cos \theta - \sqrt{R^2 - \sin^2 \theta}]
\]
\[
= 1 + \frac{1}{2} (9.5 - 1) [3.86 + 1 - \cos (20^\circ) - \sqrt{3.86^2 - \sin^2 (20^\circ)}]
\]
\[
= 1.32
\]
\[
V = 1.32 V_c = (1.32)(59 \text{ cm}^3) = 77.9 \text{ cm}^3 = 0.0000779 \text{ m}^3
\]

This indicates that, during combustion, the volume in the combustion chamber has only increased by a very small amount and shows that combustion in an SI engine occurs at almost constant volume at TDC.

QUESTION 2:
The engine in Example Problem 2-1 is connected to a dynamometer which gives a brake output torque reading of 205 N-m at 3600 RPM. At this speed air enters the cylinders at 85 kPa and
60°C, and the mechanical efficiency of the engine is 85%. Calculate:

1. brake power
2. indicated power
3. brake mean effective pressure
4. indicated mean effective pressure
5. friction mean effective pressure
6. power lost to friction
7. brake work per unit mass of gas in the cylinder
8. brake specific power
9. brake output per displacement
10. engine specific volume

1) Using Eq. (2-43) to find brake power:
   \[ \dot{W}_b = 2 \pi N \tau = (2 \pi \text{ radians/rev})(3600/60 \text{ rev/sec})(205 \text{ N-m}) \]
   \[ = 77,300 \text{ N-m/sec} = 77.3 \text{ kW} = 104 \text{ hp} \]

2) Using Eq. (2-47) to find indicated power:
   \[ \dot{W}_i = \dot{W}_b / \eta_m = (77.3 \text{ kW}) / (0.85) = 90.9 \text{ kW} = 122 \text{ hp} \]

3) Using Eq. (2-41) to find the brake mean effective pressure:
   \[ \text{bme}p = 4 \pi \tau / V_d = (4 \pi \text{ radians/cycle})(205 \text{ N-m}) / (0.003 \text{ m}^3/\text{cycle}) \]
   \[ = 859,000 \text{ N/m}^2 = 859 \text{ kPa} = 125 \text{ psia} \]
4) Equation (2-37c) gives indicated mean effective pressure:

\[ \text{imep} = \frac{\text{bmept}}{\eta_m} = \frac{(859 \text{ kPa})}{(0.85)} = 1010 \text{ kPa} = 146 \text{ psia} \]

5) Equation (2-37d) is used to calculate friction mean effective pressure:

\[ \text{fmept} = \text{imept} - \text{bmept} = 1010 - 859 = 151 \text{ kPa} = 22 \text{ psia} \]

6) Equations (2-15) and (2-44) are used to find friction power lost:

\[ A_p = \left( \frac{\pi}{4} \right) B^2 = \left( \frac{\pi}{4} \right)(0.086 \text{ m})^2 = 0.00581 \text{ m}^2 \text{ for one cylinder} \]

\[ \dot{W}_f = \frac{1}{2n}(\text{fmept})A_p \bar{U}_p \]

\[ = \frac{1}{4}(151 \text{ kPa})(0.00581 \text{ m}^2/\text{cyl})(10.32 \text{ m/sec})(6 \text{ cyl}) \]

\[ = 13.6 \text{ kW} = 18 \text{ hp} \]

Or, it can be obtained from Eq. (2-49):

\[ \dot{W}_f = \dot{W}_i - \dot{W}_b = 90.9 - 77.3 = 13.6 \text{ kW} \]

7) First brake work is found for one cylinder for one cycle using Eq. (2-29):

\[ W_b = (\text{bmept})V_d = (859 \text{ kPa})(0.0005 \text{ m}^3) = 0.43 \text{ kJ} \]

It can be assumed the gas entering the cylinders at BDC is air:

\[ m_a = \frac{PV_{\text{BDC}}}{RT} = \frac{P(V_d + V_c)}{RT} \]

\[ = (85 \text{ kPa})(0.0005 + 0.000059) \text{ m}^3/(0.287 \text{ kJ/kg-K})(333 \text{ K}) \]

\[ = 0.00050 \text{ kg} \]

Brake specific work per unit mass:

\[ w_b = \frac{W_b}{m_a} = (0.43 \text{ kJ})/(0.00050 \text{ kg}) = 860 \text{ kJ/kg} = 370 \text{ BTU/lbm} \]

8) Equation (2-51) gives brake specific power:

\[ \text{BSP} = \frac{\dot{W}_b}{A_p} = \frac{(77.3 \text{ kW})}{[(\pi/4)(0.086 \text{ m})^2/(6 \text{ cylinders})]} \]

\[ = 2220 \text{ kW/m}^2 = 0.2220 \text{ kW/cm}^2 = 1.92 \text{ hp/in.}^2 \]

9) Equation (2-52) gives brake output per displacement:

\[ \text{BOPD} = \frac{\dot{W}_b}{V_d} = \frac{(77.3 \text{ kW})}{(3 \text{ L})} \]

\[ = 25.8 \text{ kW/L} = 35 \text{ hp/L} = 0.567 \text{ hp/in.}^3 \]

10) Equation (2-53) gives engine specific volume:

\[ \text{BSV} = \frac{V_d}{\dot{W}_b} = \frac{1}{\text{BOPD}} = 1/25.8 \]

\[ = 0.0388 \text{ L/kW} = 0.0286 \text{ L/hp} = 1.76 \text{ in.}^3/\text{hp} \]

Extracted from example problem 2-2
QUESTION 3:

The engine in Example Problem 2-2 (Question 2) is running with an air-fuel ratio $AF = 15$, a fuel heating value of $44,000\, \text{kJ/kg}$, and a combustion efficiency of $97\%$. Calculate:

1. rate of fuel flow into engine
2. brake thermal efficiency
3. indicated thermal efficiency
4. volumetric efficiency
5. brake specific fuel consumption

1) From Example Problem 2-2, the mass of air in one cylinder for one cycle is $m_a = 0.00050 \, \text{kg}$. Then:

$$m_f = m_a / AF = 0.00050 / 15 = 0.000033 \, \text{kg of fuel per cylinder per cycle}$$

Therefore, the rate of fuel flow into the engine is:

$$\dot{m}_f = (0.000033 \, \text{kg/cyl-cycle})(6 \, \text{cyl})(3600 / 60 \, \text{rev/sec})(1 \, \text{cycle/2 rev})$$

$$= 0.0060 \, \text{kg/sec} = 0.0132 \, \text{lbm/sec}$$

2) Using Eq. (2-64) to find brake thermal efficiency:

$$\left( \eta_b \right)_b = \dot{W}_b / \dot{m}_f Q_{HV} \eta_c = (77.3 \, \text{kW}) / (0.0060 \, \text{kg/sec})(44,000 \, \text{kJ/kg})(0.97)$$

$$= 0.302 = 30.2\%$$

Or, using Eq. (2-68) for one cycle of one cylinder:

$$\left( \eta_b \right)_i = W_b / \dot{m}_f Q_{HV} \eta_c = (0.43 \, \text{kJ}) / (0.000033 \, \text{kg})(44,000 \, \text{kJ/kg})(0.97)$$

$$= 0.302$$

3) Indicated thermal efficiency using Eq. (2-65):

$$\left( \eta_i \right)_i = \left( \eta_b \right)_b / \eta_m = 0.302 / 0.85 = 0.355 = 35.5\%$$

4) Using Eq. (2-69) with standard air density for volumetric efficiency:

$$\eta_v = m_a / \rho_a V_d = (0.00050 \, \text{kg}) / (1.181 \, \text{kg/m}^3)(0.0005 \, \text{m}^3)$$

$$= 0.847 = 84.7\%$$

5) Using Eq. (2-59) for brake specific fuel consumption:

$$\text{bsfc} = \dot{m}_f / \dot{W}_b = (0.0060 \, \text{kg/sec}) / (77.3 \, \text{kW})$$

$$= 7.76 \times 10^{-5} \, \text{kg/kW-sec} = 279 \, \text{gm/kW-hr} = 0.459 \, \text{lbm/hp-hr}$$

Extracted from example problem 2-3.
Isooctane is burned with 120% theoretical air in a small three-cylinder turbocharged automobile engine. Calculate:

1. air–fuel ratio
2. fuel–air ratio
3. equivalence ratio

Stoichiometric reaction:

\[ C_8H_{18} + 12.5 \ O_2 + 12.5(3.76) \ N_2 \rightarrow 8 \ CO_2 + 9 \ H_2O + 12.5(3.76) \ N_2 \]

With 20% excess air:

\[ C_8H_{18} + 15 \ O_2 + 15(3.76) \ N_2 \rightarrow 8 \ CO_2 + 9 \ H_2O + 15(3.76) \ N_2 + 2.5 \ O_2 \]

With 20% excess air, all the fuel gets burned, and the same amount of CO₂ and H₂O is found in the products. In addition, there is some oxygen and additional nitrogen in the products (the excess air).

1) Equations (2-55) and (4-1) are used to find the air–fuel ratio:

\[ AF = \frac{m_a}{m_f} = \frac{N_a M_a}{N_f M_f} = \frac{[(15)(4.76)(29)]}{[(1)(114)]} \]

\[ = 18.16 \]

2) Equation (2-56) is used to find fuel–air ratio:

\[ FA = \frac{m_f}{m_a} = \frac{1}{AF} = \frac{1}{18.16} = 0.055 \]

3) Fuel–air ratio of stoichiometric combustion:

\[ (FA)_{stoich} = \frac{[(1)(114)]}{[(12.5)(4.76)(29)]} = 0.066 \]

Equivalence ratio is obtained using Eq. (4-2):

\[ \phi = \frac{(FA)_{act}}{(FA)_{stoich}} = \frac{0.055}{0.066} = 0.833 \]

Equation
At what temperature will water start to condense out of exhaust gases of the engine in Example Problem 4-1? Exhaust pressure is one atmosphere. Calculate this for:

1. dry inlet air
2. inlet air with relative humidity of 55%

1) Reaction equation from Example Problem 4-1:

\[ C_8H_{18} + 15 O_2 + 15(3.76) N_2 \rightarrow 8 CO_2 + 9 H_2O + 2.5 O_2 + 15(3.76) N_2 \]

Mole fraction of water vapor in exhaust products:

\[ x_v = \frac{N_v}{N_{total}} = \frac{9}{[8 + 9 + 2.5 + 15(3.76)]} = 0.1186 \]

Partial pressure of water vapor:

\[ P_v = x_v P_{total} = (0.1186)(101 \text{ kPa}) = 11.98 \text{ kPa} \]

The dew point is the temperature at which this water vapor pressure becomes saturated. From steam tables [90]:

\[ T_{DP} = 49^\circ \text{C} \]

2) If the relative humidity (rh) of the inlet air is 55% at \( T = 25^\circ \text{C} \), then the vapor pressure of water at the inlet will be

\[ P_v = (rh)P_{Sat \ at \ 25^\circ C} = (0.55)(3.169 \text{ kPa}) = 1.743 \text{ kPa} \]

Using psychrometric equations and steam tables from any thermodynamics textbook (e.g. [90]), the specific humidity is:

\[ \omega_v = \frac{m_v}{m_a} = 0.622\left[\frac{P_v}{(P - P_v)}\right] = (0.622)\left[\frac{(1.743)}{(101 - 1.743)}\right] = 0.0109 \text{ kg}_{v}/\text{kg}_{a} \]

Changing this mass ratio to a molar ratio using the molecular weights of air (29) and water vapor (18) gives the number of moles of water carried in with the air for one mole of fuel:

\[ N_v = N_{air} \omega_v (M_{air}/M_v) = [(15)(4.76)](0.0109)(29)/(18) = 1.25 \]

The reaction equation then becomes:

\[ C_8H_{18} + 15 O_2 + 15(3.76) N_2 + 1.25 H_2O \rightarrow \]

\[ 8 CO_2 + 10.25 H_2O + 2.5 O_2 + 15(3.76) N_2 \]

The mole fraction of water vapor in the exhaust is

\[ x_v = \frac{10.25}{[8 + 10.25 + 2.5 + 15(3.76)]} = 0.1329 \]

The partial pressure of water vapor is

\[ P_v = x_v P_{total} = (0.1329)(101 \text{ kPa}) = 13.42 \text{ kPa} \]

Dew point temperature:

\[ T_{DP} = 52^\circ \text{C} \]

Extracted from example problem 4-1 & 4-2
QUESTION 5:
Find the adiabatic flame temperature of isoctane burned with an equivalence ratio of 0.833 in dry air as in Example Problem 4-1. It can be assumed that the reactants are at a temperature of 427°C (700 K) after the compression stroke.

From Example Prob. 4-1:

\[ C_8H_{18} + 15 \text{O}_2 + 15(3.76) \text{N}_2 \rightarrow 8 \text{CO}_2 + 9 \text{H}_2\text{O} + 2.5 \text{O}_2 + 15(3.76) \text{N}_2 \]

Equations (4-5) and (4-8) are used for adiabatic combustion. Enthalpy values can be obtained from most thermodynamic textbooks. The values used here are from [90]:

\[ \sum_{\text{PROD}} N_i (h_f^o + \Delta h)_i = \sum_{\text{REACT}} N_i (h_f^o - \Delta h)_i \]

\[ 8[(-393,522) + \Delta h_{\text{CO}_2}] + 9[(-241,826) + \Delta h_{\text{H}_2\text{O}}] + 2.5[0 + \Delta h_{\text{O}_2}] + 15(3.76)[0 + \Delta h_{\text{N}_2}] = [(-259,280) + (73473)] + 15[0 + (12,499)] + 15(3.76)[0 + (11,937)] \]

Simplify:

\[ 8\Delta h_{\text{CO}_2} + 9\Delta h_{\text{H}_2\text{O}} + 2.5\Delta h_{\text{O}_2} + 56.4\Delta h_{\text{N}_2} = 5,999,535 \]

By trial and error, find the temperature that satisfies this equation. Try \( T = 2400 \text{ K} \):

\[ 8(115,779) + 9(93,741) + 2.5(74,453) + 56.4(70,640) = 5,940,130 \]

This is too low, so try \( T = 2600 \text{ K} \):

\[ 8(128,074) + 9(104,520) + 2.5(82,225) + 56.4(77,963) = 6,567,948 \]

This is too high, so the adiabatic flame temperature is found by interpolation:

\[ T_{\text{max}} = 2419 \text{ K} = 2146^\circ \text{C} \]

B) The four-cylinder engine of a light truck owned by a utility company has been converted to run on propane fuel. A dry analysis of the engine exhaust gives the following volumetric
percentages:

- CO$_2$ 4.90%  
- CO 9.79%  
- O$_2$ 2.45%

Calculate the equivalence ratio at which the engine is operating. The three components identified sum up to 4.90 + 9.79 + 2.45 = 17.14% of the total, which means that the remaining gas (nitrogen) accounts for 82.86% of the total. Volume percent equals molar percent, so if an unknown amount of fuel is burned with an unknown amount of air, the resulting reaction is:

$$x \text{ C}_3\text{H}_8 + y \text{ O}_2 + y(3.76) \text{ N}_2 \rightarrow 4.90 \text{ CO}_2 + 9.79 \text{ CO} + 2.45 \text{ O}_2 + 82.86 \text{ N}_2 + z \text{ H}_2\text{O}$$

where:  
- \(z\) = number of moles of water vapor removed before dry analysis

Conservation of nitrogen during reaction gives:

$$y(3.76) = 82.86 \quad \text{or} \quad y = 22.037$$

Conservation of carbon:

$$3x = 4.90 + 9.79 \quad \text{or} \quad x = 4.897$$

Conservation of hydrogen:

$$8x = 8(4.897) = 2z \quad \text{or} \quad z = 19.588$$

The reaction is:

$$4.90 \text{ C}_3\text{H}_8 + 22.037 \text{ O}_2 + 22.037(3.76) \text{ N}_2 \rightarrow 4.90 \text{ CO}_2 + 9.79 \text{ CO} + 2.45 \text{ O}_2 + 82.86 \text{ N}_2 + 19.588 \text{ H}_2\text{O}$$

Dividing by 4.90:

$$\text{C}_3\text{H}_8 + 4.50 \text{ O}_2 + 4.50(3.76)\text{N}_2 \rightarrow \text{CO}_2 + 2 \text{ CO} + 0.50 \text{ O}_2 + 16.92 \text{ N}_2 + 4 \text{ H}_2\text{O}$$

Actual air–fuel ratio:

$$\text{AF}_{\text{act}} = \frac{m_a}{m_f} = \frac{[(4.50)(4.76)(29)]}{[(1)(44)]} = 14.12$$

Stoichiometric combustion:

$$\text{C}_3\text{H}_8 + 5 \text{ O}_2 + 5(3.76) \text{ N}_2 \rightarrow 3 \text{ CO}_2 + 4 \text{ H}_2\text{O} + 5(3.76) \text{ N}_2$$

Stoichiometric air–fuel ratio:

$$\text{AF}_{\text{stoich}} = \frac{m_a}{m_f} = \frac{[(5)(4.76)(29)]}{[(1)(44)]} = 15.69$$

Equivalence ratio using Eq. (4-2):

$$\phi = \frac{\text{AF}_{\text{stoich}}}{\text{AF}_{\text{act}}} = \frac{15.69}{14.12} = 1.11$$

Extracted from example problem 4-3 & 4-4

**QUESTION 6:**

A) A 2.8-liter four-cylinder square engine (bore = stroke) with two intake valves per cylinder is designed to have a maximum speed of 7500 RPM. Intake temperature is 600°C. Calculate:
1. intake valve area
2. diameter of intake valves
3. valve lift

Using Eq. (3-i) for speed of sound at inlet conditions:

\[ c_i = \sqrt{kRT} = \sqrt{(1.40)(287 \text{ J/kg-K})(333 \text{ K})} = 366 \text{ m/sec} \]

where:
- \( R \) = gas constant
- \( T \) = temperature
- \( k = c_p/c_v = 1.40 \)

The value of 1.40 is used here because this corresponds to the inlet temperature of 333 K.

For one cylinder:

\[ V_d = (2.8 \text{ L})/4 = 0.7 \text{ L} = 0.0007 \text{ m}^3 \]

Use Eq. (2-8) to get stroke with \( B = S \):

\[ V_d = (\pi/4)B^2S = (\pi/4)S^3 = 0.0007 \text{ m}^3 \]

\[ S = 0.0962 \text{ m} = 9.62 \text{ cm} = B \]

Use Eq. (2-2) to get maximum average piston speed:

\[ (\bar{U}_p)_{max} = 2SN = (2 \text{ strokes/rev})(0.0962 \text{ m/stroke})(7500/60 \text{ rev/sec}) \]

\[ = 24.1 \text{ m/sec} \]

1) Equation (5-4) gives total intake valve area needed:

\[ A_i = 1.3B^2(\bar{U}_p)_{max}/c_i = (1.3)(0.0962 \text{ m})^2(24.1 \text{ m/sec})/(366 \text{ m/sec}) \]

\[ = 0.000792 \text{ m}^2 = 7.92 \text{ cm}^2 = 1.23 \text{ in}^2 \]

2) For each valve:

\[ A_i = (7.92/2) \text{ cm}^2 = (\pi/4)d_v^2 \]

Diameter of each valve:

\[ d_v = 2.25 \text{ cm} = 0.886 \text{ in.} \]

3) Equation (5-1) gives an upper limit to valve lift:

\[ l_{max} = d_v/4 = (2.25 \text{ cm})/4 = 0.56 \text{ cm} = 5.6 \text{ mm} = 0.22 \text{ in.} \]

B) A six-cylinder, 3.6-liter SI engine is designed to have a maximum speed of 6000RPM. At this speed the volumetric efficiency of the engine is 0.92. The engine will be equipped with a two-barrel carburetor, one barrel for low speeds and both barrels for high speed. Gasoline density can
be considered to be 750kg/m³. Calculate:

1. throat diameters for the carburetor (assume discharge coefficient \( C_{Dt} = 0.94 \))
2. fuel capillary tube diameters (assume discharge coefficient \( C_{Dc} = 0.74 \))

Use Eq. (2-70) to get air flow at maximum speed:

\[
(m_a)_{max} = \eta_v \rho_a V_d N/n
\]

\[
= (0.92)(1.181 \text{ kg/m}^3)(0.0036 \text{ m}^3/\text{cycle})(6000/60 \text{ rev/sec})/(2 \text{ rev/cycle})
\]

\[
= 0.1956 \text{ kg/sec}
\]

1) Equation (5-12) is used to find the throat area needed at maximum engine speed:

\[
(m_a)_{max} = 236.5 \ C_{Dt} A_t = (236.5)(0.94) A_t = 0.1956
\]

\[
A_t = 0.00088 \text{ m}^2 = 8.8 \text{ cm}^2
\]

For each barrel:

\[
A_t = (\pi/4) d_t^2 = 0.00044 \text{ m}^2
\]

\[
d_t = 0.0237 \text{ m} = 2.37 \text{ cm} = 0.93 \text{ in.}
\]

Extracted from example problem 5-1 & 5-2

**QUESTION 7:**
A six-cylinder, 4.8-liter, supercharged engine operating at 3500 RPM has an overall volumetric efficiency of 158%. The supercharger has an isentropic efficiency of 92% and a mechanical efficiency in its link with the engine of 87%. It is desired that air be delivered to the cylinders at 65°C and 180 kPa, while ambient conditions are 23°C and 98 kPa.

Calculate:

1. amount of aftercooling needed
2. engine power lost to run supercharger

Equation (2-70) gives mass flow rate of air to engine:

\[ m_a = \eta_r \rho_a V_d n / n \]

\[ = (1.58)(1.181 \text{ kg/m}^3)(0.0048 \text{ m}^3/\text{cycle})(3500/60 \text{ rev/sec})/(2 \text{ cycles/rev}) \]

\[ = 0.261 \text{ kg/sec} \]

Using Fig. 5-7 and Eq. (5-15):

\[ T_{2s} = T_1 (P_2/P_1)^{(k - 1)/k} = (296 \text{ K})(180/98)^{1.4-1/1.4} = 352 \text{ K} = 79°C \]

Equation (5-14) is used to find actual air temperature at compressor exit:

\[ (\eta_s)_{sc} = (T_{2s} - T_1)/(T_{2A} - T_1) = 0.92 = (352 - 296)/(T_{2A} - 296) \]

\[ T_{2A} = 357 \text{ K} = 84°C \]

1) Amount of aftercooling needed to reduce air temperature back to 65°C:

\[ \dot{Q} = m_a c_p (T_{2A} - T_{in}) \]

\[ = (0.261 \text{ kg/sec})(1.005 \text{ kJ/kg-K})(357 - 338) \text{K} = 5.0 \text{ kW} \]

2) Equations (5-13) and (5-16) are combined to find the engine power needed to drive the supercharger:

\[ \dot{W} = m_a c_p (T_{out} - T_{in}) / \eta_m \]

\[ = (0.261 \text{ kg/sec})(1.005 \text{ kJ/kg-K})(357 - 296) \text{K}/(0.87) \]

\[ = 18.4 \text{ kW} = 24.7 \text{ hp} \]

---

**Figure 5-7** Ideal flow process (1-2_s) and actual flow process (1-2_A) through a supercharger or a turbocharger compressor in (a) pressure–volume coordinates, and (b) temperature–entropy coordinates.
The diesel engine of Example Problem 5-4 (Question 6B) has a compression ratio of 18:1 and operates on an air-standard Dual cycle. At 2400 RPM, combustion starts at 7° bTDC and lasts for 42° of engine rotation. The ratio of connecting rod length to crank offset is \( R = 3.8 \).

Calculate:
1. time for one injection
2. fuel flow rate through an injector

For one cylinder for one cycle:

\[
V_d = \frac{(0.0032 \, \text{m}^3)}{5} = 0.00064 \, \text{m}^3
\]

Equation (2-69) gives the mass of air:

\[
m_a = \eta_v \rho_a V_d = (0.95)(1.181 \, \text{kg/m}^3)(0.00064 \, \text{m}^3) = 0.000718 \, \text{kg}
\]

Equations (2-56) and (2-57) give the mass of fuel needed:

\[
m_f = \phi m_a / (AF)_{\text{stoich}} = (0.80)(0.000718 \, \text{kg})/(14.5) = 0.0000396 \, \text{kg}
\]

Engine time = \((60 \, \text{sec/min})/(2400 \, \text{rev/min}) = 0.025 \, \text{sec/rev}
\]

\[= (0.025 \, \text{sec/rev})/(360^\circ/\text{rev}) = 6.9 \times 10^{-5} \, \text{sec/degree}\]

1. Time for injection:

\[t = (25^\circ/\text{injection})(6.9 \times 10^{-5} \, \text{sec/degree}) = 0.00173 \, \text{sec/injection}\]

2. Injection rate:

\[\dot{m}_f = (0.0000396 \, \text{kg})/(0.00173 \, \text{sec}) = 0.0229 \, \text{kg/sec} = 0.050 \, \text{lbm/sec}\]

C) The diesel engine of Example Problem 5-4 (Question 6B) has a compression ratio of 18:1 and operates on an air-standard Dual cycle. At 2400 RPM, combustion starts at 7° bTDC and lasts for 42° of engine rotation. The ratio of connecting rod length to crank offset is \( R = 3.8 \).

Calculate:
1. ignition delay
2. cycle cutoff ratio

1) Combustion starts at 7° bTDC and fuel injection starts at 20° bTDC (from Example Problem 5-4). Ignition delay in degrees of engine rotation:

\[
\text{ID} = 13^\circ \text{ of engine rotation}
\]

Ignition delay in seconds:

\[
\text{ID} = (13^\circ)/[(2400/60 \, \text{rev/sec})(360^\circ/\text{rev})] = 0.0009 \, \text{sec}
\]

2) Combustion stops at 35° aTDC. Equation (2-14) is used to find cutoff ratio:

\[
\beta = V/V_{\text{TDC}} = V/V_c = 1 + \frac{1}{2} (r_c - 1)[R + 1 - \cos \theta - \sqrt{R^2 - \sin^2 \theta}]
\]

\[= 1 + \frac{1}{2} (18 - 1)[3.8 + 1 - \cos (35^\circ) - \sqrt{(3.8)^2 - \sin^2 (35^\circ)}]
\]

\[= 2.91
\]

Extracted from example problem 7.3
QUESTION 8:

A four-cylinder, 3.2-liter engine running at 4500 RPM has a swirl ratio of 6 as defined by Eq. (6-1). The stroke and bore are related as $S = 1.06 \, B$.

Calculate:

1. angular velocity of gas mixture in the cylinder using the paddle wheel model
2. swirl ratio as defined by Eq. (6-2)

1) Equation (6-1) is used to find angular velocity:

\[
(SR)_1 = \frac{\omega}{N} = 6 = \frac{\omega}{(4500/60 \text{ rev/sec})}
\]

\[
\omega = 450 \text{ rev/sec}
\]

2) For one cylinder:

\[
V_d = \frac{(3.2 \text{ L})}{4} = 0.8 \text{ L} = 0.0008 \text{ m}^3
\]

\[
V_d = \frac{(\pi/4)B^2S}{(\pi/4)(1.06)B^3} = 0.0008 \text{ m}^3
\]

\[
B = 0.0987 \text{ m} = 9.87 \text{ cm}
\]

\[
S = (1.06)(9.87 \text{ cm}) = 10.46 \text{ cm} = 0.1046 \text{ m}
\]

Equation (2-2) is used to find average piston speed:

\[
\bar{U}_p = 2SN = (2 \text{ strokes/rev})(0.1046 \text{ m/stroke})(4500/60 \text{ rev/sec})
\]

\[
= 15.7 \text{ m/sec}
\]

Tangential speed of rotating gas:

\[
u_t = 2\pi\omega r = (2\pi \text{ radians/rev})(450 \text{ rev/sec})(0.0987/2 \text{ m}) = 139.5 \text{ m/sec}
\]

Using Eq. (6-2) to find swirl ratio:

\[
(SR)_2 = \frac{u_t}{\bar{U}_p} = \frac{139.5}{15.7} = 8.9
\]

QUESTION 9:
Crevice volume of an engine equals 2% of the total clearance volume. It can be assumed that pressure in the crevices is about the same as in the combustion chamber but the temperature stays at the cylinder wall temperature of 180°C. Cylinder inlet conditions are 60°C and 98 kPa, and the compression ratio is 9.6:1.

Calculate:

1. what percent of the fuel is trapped in the crevices at the end of the compression stroke
2. what percent of the fuel ends up in the exhaust due to being trapped in the crevice volume

It can be assumed that only 80% of fuel trapped in the crevice volume gets burned later in the power stroke.

Using Eqs. (3-4) and (3-5) for conditions at the end of compression stroke:

\[ P_2 = P_1(r_c)^k = (98 \text{ kPa})(9.6)^{1.35} = 2076 \text{ kPa} \]

\[ T_2 = T_1(r_c)^{k-1} = (333 \text{ K})(9.6)^{0.35} = 735 \text{ K} = 462°C \]

Mass in crevice as a fraction of clearance volume:

\[ m_{\text{crev}} = \frac{PV}{RT} = \frac{(2076 \text{ kPa})(0.02V_c \text{ m}^3)}{(0.287 \text{ kJ/kg-K})(453 \text{ K})} \]

\[ = 0.319 V_c \]

Mass in the combustion chamber at TDC:

\[ m_{\text{chamb}} = \frac{(2076 \text{ kPa})(V_c)}{(0.287 \text{ kJ/kg-K})(735 \text{ K})} = 9.841 V_c \]

1) Percent of fuel in crevice:

\[ \% = \left[ \frac{m_{\text{crev}}}{m_{\text{total}}} \right](100) \]

\[ = \left[ \frac{(0.319 V_c)(100)}{[(0.319 V_c) + (9.841 V_c)]} \right] = 3.14 \% \]

2) Twenty percent of this fuel does not get burned:

percent of total not burned = (0.20)(3.14) = 0.63 \%

This is fuel lost in the exhaust due to crevice flow. Additional fuel is lost due to mixing and combustion inefficiencies.

Extracted from example problem 6-2

QUESTION 10:
A) The spark plug is fired at 18°bTDC in an engine running at 1800 RPM. It takes 8° of engine rotation to start combustion and get into flame propagation mode. Flame termination occurs at 12°aTDC. Bore diameter is 8.4 cm and the spark plug is offset 8 mm from the centerline of the cylinder. The flame front can be approximated as a sphere moving out from the spark plug. Calculate the effective flame front speed during flame propagation.
Rotational angle during flame propagation is from 10°bTDC to 12°aTDC, which equals 22°.

Time of flame propagation:
\[ t = \frac{22°}{\left[\frac{360°}{\text{rev}}\left(\frac{1800}{60 \text{ rev/sec}}\right)\right]} = 0.00204 \text{ sec} \]

Maximum flame travel distance:
\[ D_{\text{max}} = \text{bore}/2 + \text{offset} = (0.084/2) + (0.008) = 0.050 \text{ m} \]

Effective flame speed:
\[ V_f = D_{\text{max}}/t = (0.050 \text{ m})/(0.00204 \text{ sec}) = 24.5 \text{ m/sec} \]

B) The engine in Example Problem 1A is now run at 3000RPM. As speed is increased in this engine, greater turbulence and swirl increase the flame front speed at a rate such that \( Vl < X 0.85 \) N. Flame development after spark plug firing still takes 8° of engine rotation. Calculate how much ignition timing must be advanced such that flame termination again occurs at 12°aTDC.

Flame speed:
\[ V_f = (0.85) \left(\frac{3000}{1800}\right) (24.5 \text{ m/sec}) = 34.7 \text{ m/sec} \]

With flame travel distance the same, the time of flame propagation is
\[ t = D_{\text{max}} / V_f = (0.050 \text{ m}) / (34.7 \text{ m/sec}) = 0.00144 \text{ sec} \]

Rotational angle during flame propagation:
\[ \text{angle} = \left(\frac{3000}{60 \text{ rev/sec}}\right) \left(\frac{3600}{\text{rev}}\right) (0.00144 \text{ sec}) = 25.92° \]

Flame propagation starts at 13.92°bTDC, and spark plug firing is at 21.92°bTDC. Ignition timing must be advanced 3.92°.