Chapter 4

Thermodynamics and Engine Cycles

Brief Engine History

- Internal combustion (IC) engines produce power through a combustion process occurring within the piston chambers. In contrast, combustion occurs outside the piston chambers in a steam engine - the combustion heats a boiler to produce steam and the steam is delivered to the piston chambers to produce power.

Brief History con’t

- In 1862, Beau de Rochas set forth his famous four principles of operation for an efficient IC engine:
  1. The combustion chamber should have the smallest possible surface-to-volume ratio
  2. The expansion should take place as rapidly as possible
  3. The piston should have the longest possible stroke
  4. The pressure should be as high as possible at the start of expansion
Brief History con’t

- The first two principles were aimed at reducing the heat loss to a minimum, while the second two were aimed at obtaining the maximum possible work from each power stroke.

Brief History con’t

- Nikolaus Otto built on Beau de Rochas’ principles, patenting his famous Otto-cyle engine in 1876. Two years later he built a successful IC engine. Otto was the first to use the four-stroke cycle (intake, compression, power, and exhaust strokes) that is still used in most IC engines today. With the expiration of the Otto patent in 1890, there was a spurt in development and commercialization of IC engines.

Brief History con’t

- Dr. Rudolph Diesel patented his compression-ignition engine in 1892. The first diesel engine in the United States was built for the Busch brewery in St. Louis under license from Rudolph Diesel.
Four-Cycle Engine Analysis

- The intake, compression, expansion, and exhaust processes can be carried out in two or four piston strokes. Thus, there are two types of engines, those with two-stroke cycles and those with four-stroke cycles.
- Four-cycle engines have become much more popular than two-stroke engines.

Basic Thermodynamic Equations

- Two thermodynamic laws are used in cycle analysis, the ideal gas law and the polytropic compression law. The ideal gas law is:
  \[ pV = MRT \]
  - Where \( p \) = absolute gas pressure, kPa
  - \( V \) = gas volume, m\(^3\)
  - \( M \) = mass of trapped gas, kg
  - \( T \) = absolute temperature of gas, K
  - \( R \) = ideal gas constant = 8.314/mole wt of gas
- Note: the gas constant drops out of the analysis

Basic Thermodynamic Equations con’t

- The mass of gas in a combustion chamber remains essentially constant during the compression process. If the temperature also remained constant, the next equation indicates that the pressure would vary inversely with the volume. That is, the pressure would double if the volume was halved. Such a process is called an isothermal process. In an engine, the temperature increases considerably during the compression stroke; therefore, the pressure increases more than in an isothermal process.
Basic Thermodynamic Equations con’t

- A polytropic process follows:
  - \( pV^n = C_p \)
    - Where \( C_p \) = a constant
    - \( n \) = an exponent between 1 and 1.4
- If the combustion chamber were perfectly insulated, so that there was no energy loss, then \( n = k = 1.4 \) and the process is called an adiabatic process.

Basic Thermodynamic Equations con’t

- Suppose that a gas changed from state 1 at the beginning of compression cycle to state 2 at the end of compression cycle.
  - \( \frac{(p_1V_1)}{T_1} = \frac{(p_2V_2)}{T_2} \)
- The polytropic law can be used to show that:
  - \( \frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^n \)
  - Or \( \frac{p_2}{p_1} = r^n \)

Basic Thermodynamic Equations con’t

- The ratio, \( V_1 / V_2 \), is referred to as the compression ratio: \( r = \frac{V_1}{V_2} \)
  - \( V_1 \) = Volume at Bottom Dead Center
  - \( V_2 \) = Volume at Top Dead Center
- By combining the ideal gas law and the polytropic law, it can be shown that:
  - \( \frac{T_2}{T_1} = r^{n-1} \)
The Otto Cycle

- The theoretical Otto cycle includes:
  1) intake from Point 0 to Point 1,
  2) adiabatic compression from Point 1 to 2,
  3) constant-volume heat input from Point 2 to 3,
  4) adiabatic compression from Point 3 to 4,
  5) blow down from Point 4 to 1, and
  6) exhaust from Point 1 to 0.

The Otto Cycle con’t

- The volume, \( V_2 \), is called the clearance volume; it is the cylinder volume when the piston is at HDC.
- The cylinder displacement is the change in volume from Point 1 to 2:
  \[ D_c = V_1 - V_2 \]

The Otto Cycle con’t

- In analyzing the Otto cycle, it is assumed that the conditions at Point 1 are atmospheric and thus \( p_1 \) and \( T_1 \) are given. It is also assumed that cylinder displacement, \( D_c \), and compression ratio, \( r \), are given:
  \[ V_2 = \frac{D_c}{(r-1)} \] and \[ V_1 = \frac{r^2D_c}{(r-1)} \]
The Otto Cycle con’t

- $T_3$ is the maximum temperature occurring in the cycle. Thus, $T_3$ is also specified as one of the givens for the cycle. The adiabatic flame temperature, $T_3 = 2700 \text{ K}$, is usually specified as the given value for $T_3$.

The work accomplished in the Otto cycle can be calculated by integrating $p*dv$ through the cycle. The results can be simplified to:

$$W = \frac{p_1 D_c \left( (r-\gamma_1) + (T_3/T_1)(r-\gamma_2) \right)}{(k-1)(r-1)}$$

Where $W = \text{work, J/cycle}$
- $p_1 = \text{initial pressure, kPa}$
- $D_c = \text{displacement volume, L}$
- $k = 1.4$

The cycle mean effective pressure, $p_{cme}$ is:

$$p_{cme} = \frac{W}{D_c}$$

Where $p_{cme} = \text{cycle mean effective pressure, kPa}$

The cycle power is:

$$P_c = \frac{(p_{cme} - D_c N_s)}{2(60,000)}$$

Where $P_c = \text{cycle power, kW}$
- $N_s = \text{crankshaft speed of the engine, rpm}$
The Otto Cycle con’t

- The cycle efficiency is: \( e_{\text{Otto}} = 1 - r^{1-k} \)
  - Where \( e_{\text{Otto}} \) = efficiency, decimal
- The energy flows into and out of the cycle are:
  - \( Q_{\text{in}} = W / e_{\text{Otto}} \) and \( Q_{\text{out}} = Q_{\text{in}} - W \)
  - Where \( Q_{\text{in}} \) = energy into cycle, J/cycle
  - \( Q_{\text{out}} \) = energy rejected from cycle, J/cycle

The Otto Cycle con’t

- There are a number of simplifying assumptions that underlie the theoretical Otto cycle. These include zero friction, air as the only working fluid, zero heat transfer, constant-volume heat addition, and constant-volume heat rejection.
- An actual pV diagram includes pumping losses during the intake and exhaust stroke that are not included in the theoretical Otto cycle. All of these assumptions are violated to some extent in a running engine and thus the \( e_{\text{Otto}} \) and \( p_{\text{c}} \) values calculated for the Otto cycle can never be achieved in practice.

The Classic Diesel Cycle

- Diesel realized his engine needed a compression ratio high enough to self-ignite the fuel at the end of the compression stroke.
The Dual Cycle

- The dual cycle is similar to the Otto cycle, except the process between Points 2 and 2a is constant-volume heat input, while 2a to 3 is constant-pressure heat input.

The Dual Cycle

- In analyzing the dual cycle, it is assumed that conditions at Point 1 are atmospheric and thus $p_1$ and $T_1$ are given. As with the Otto cycle, $r$, $D_c$, and $T_3$ are also given.

The Classic Diesel Cycle con’t

- By setting $T_2 = 750$ K (the self-ignition temperature of diesel fuel), taking $T_1 = 300$ K (a reasonable ambient temperature) and assuming $n = 1.33$, the compression ratio would have to be at least 16:1 to self-ignite the fuel. An even higher $r$ would be required for lower values of $T_1$. 
The Dual Cycle con’t

In addition, it is necessary to specify the fraction of energy input that occurs at constant pressure:

- \( \beta_{dc} = \frac{Q_{inp}}{Q_{inp} + Q_{inv}} \)
- Where \( \beta_{dc} \) = fraction of energy input at constant pressure
- \( Q_{inp} \) = heat input at constant pressure, J/cycle
- \( Q_{inv} \) = heat input at constant volume, J/cycle

The Dual Cycle con’t

It can be shown that the dual-cycle efficiency is:

\[ e_{dual} = 1 - r^{1-k} \left[ (\beta_{dc} (r_{co} - 1) + k (r_{co} - 1) (1 - \beta_{dc})) / k (r_{co} - 1) \right] \]

The dual-cycle efficiency increases from that of the theoretical Diesel cycle at \( \beta_{dc} = 1 \) to that of the theoretical Otto cycle at \( \beta_{dc} = 0 \).

The smaller the volume into which a given amount of energy is released, the higher the pressure resulting from that energy input, and, according to de Rochas, the higher the efficiency.
The Dual Cycle con’t

- The total heat into the dual cycle is:
  - \( Q_{\text{inp}} + Q_{\text{inv}} = \frac{(p_{\text{me}} D_c)}{e_{\text{dual}}} \)
  - \( Q_{\text{out}} = (Q_{\text{inp}} + Q_{\text{inv}}) - (p_{\text{me}} D_c) \)

Part-Load Efficiency of SI and CI Engines

- To compare SI and CI engines, we will consider the brake thermal efficiency of engines (the fraction of fuel equivalent power that is converted into power at the flywheel).
- The equation for brake thermal efficiency:
  \( e_{\text{bt}} = e_{\text{t}} (1 + \frac{p_{\text{me}}}{p_{\text{bme}}})^{-1} \)

Part-Load Efficiency of SI and CI Engines con’t

- Both engines have higher mechanical efficiency at full load and lower mechanical efficiency at part load.
- The design of CI fuel-injection systems is such that injection begins at a fixed point before HDC and the rate of injection cannot be changed when the engine load changes. Instead, the duration of injections must change to suit the load on the engine; the heavier the load, the later in the cycle the injections must terminate and the larger the value of \( \beta_{dc} \).
Part-Load Efficiency of SI and CI Engines con’t

- The dual-cycle efficiency equation predicts that CI engines increase their indicated thermal efficiencies as torque load declines. On the other hand, the theoretical Otto cycle predicts no change in indicated thermal efficiency when the load on an SI engine changes.

Part-Load Efficiency of SI and CI Engines con’t

- The cycle efficiency of an SI engine would be higher than that of a CI engine if both were operated at the same compression ratio. However, engine knock considerations limit the compression ratios of SI engines to much lower values than those for CI engines. Therefore, CI engines operate more efficiently than SI engines, both at full load and part load.

Two-Cycle Engines

- Two-cycle engines can be of either the SI or the CI type. There are many more SI than CI engines of the two-cycle type.
- The two-cycle engine shown uses the crankcase as an air pump.
- Upward piston movement creates a vacuum in the crankcase, causing the spring-loaded check valve to open to admit air-fuel mixture into the crankcase.
Two-Cycle Engines

- The piston uncovers the intake and exhaust ports in the cylinder walls during the down stroke. When the ports are uncovered, the compression mixture in the crankcase rushes into the combustion chamber.

Two-Cycle Engines con’t

- The incoming mixture helps to push the spent exhaust exhaust gases from the previous cycle towards the exhaust port.
- These flow processes continue while the piston moves to CDC (Crank Dead Center/Bottom Dead Center (BDC)), reverses direction and starts upward.

Two-Cycle Engines con’t

- After the rising piston covers the intake and exhaust ports, the trapped mixture is compressed as the piston continues to rise.
- Near HDC, a spark plug fires to ignite the mixture and the resulting pressure increase forces the piston downward.
- The cycle is completed when the intake and exhaust ports are again uncovered.
Two-Cycle Engines con’t

- Use of the crankcase as an air pump interferes with its use as a sump for lubricating oil. Lubrication of the moving parts in the two-cycle SI engine is accomplished by mixing oil with the fuel.