Intake

In-cylinder Flow

Exhaust
Intake Stroke
The downward movement of the piston draws the air-fuel mixture into the cylinder through the intake valve on the intake stroke.
Compression Stroke
On the compression stroke, the mixture is compressed by the upward movement of the piston with both valves closed.
Power Stroke
At the end of the compression stroke, ignition occurs and the combustion drives the piston downward to produce power.
Exhaust Stroke

The upward-moving piston forces the burned gases out the open exhaust valve.
In the ideal four-stroke cycle the exhaust process does not consider the actual gas flow and is modeled as constant volume heat extraction.

Exhaust processes are often considered as a constant pressure processes, e.g., 5-6 with the same states 1=5.

Process 5-6 indicates a decrease in specific volume which is incorrect since as the cylinder volume decreases so does the mass and specific volume remains the same.
Ideal Intake and Exhaust Strokes

Valves supposed to operate instantaneously; intake and exhaust processes are supposed adiabatic and constant pressure.

**Unthrottled:** \( P_i = P_e = 1 \text{ atm} \)

**Throttled:** \( P_i < P_e \)

**Supercharged:** \( P_i > P_e \)
Actual Exhaust Stroke (4 \to 6)

The actual exhaust process consists of two phases: 1) Blowdown 2) Displacement

**Blowdown**: At the end of the power stroke when the exhaust valve opens the cylinder pressure is much higher than the exhaust manifold pressure (typically at 1 atm) \((P_4 > P_e)\), so the cylinder gas flows out through the exhaust valve and the pressure drops to \(P_e\).

**Displacement**: Remaining gas is pushed out of the cylinder by the piston moving to TC.
Exhaust Blowdown

During the blowdown the gas remaining in the cylinder undergoes expansion which can be modelled as isentropic process.

State 5 at the end of blowdown does not correspond to actual piston location

\[ P_5 = P_e \quad T_5 = T_4 \left( \frac{P_5}{P_4} \right)^{\frac{\gamma-1}{\gamma}} = T_4 \left( \frac{P_e}{P_4} \right)^{\frac{\gamma-1}{\gamma}} \]
Residual Gas

The gas remaining in the cylinder when the piston reaches TC is called \textbf{residual gas} which mixes with intake gas.

The residual gas temperature $T_6$ is equal to $T_5$.

The \textbf{Residual gas fraction} $f$ is defined as the ratio of the mass of residual gas to the mass of the fuel-air. Assume ideal gas:

$$f = \frac{m_6}{m_1} = \frac{m_6}{m_4} = \frac{V_6/v_6}{V_4/v_4} = \frac{1}{r} \frac{v_4}{v_6} = \frac{1}{r} \frac{T_4}{T_6} \frac{P_6}{P_4} = \frac{1}{r} \frac{T_5}{T_4} \frac{P_6}{P_4}$$

since

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{1}{\gamma}}$$

$$f = \frac{1}{r} \left(\frac{P_5}{P_4}\right)^{\frac{1}{\gamma}} = \frac{1}{r} \left(\frac{P_e}{P_4}\right)^{\frac{1}{\gamma}}$$

Typical values of $f$ are in the range from 3\% to 12\%, lower in Diesels.
Exhaust Manifold Gas Temperature ($T_7$)

The gas leaving the cylinder during blowdown has kinetic energy which is converted into thermal energy when the exhaust gas comes to rest in the exhaust gas manifold so that gas temperature in the exhaust manifold is higher than $T_5$.

Gas leaving cylinder earliest has higher velocity and correspondingly higher temperature when stagnates in the manifold ($T_{7a} > T_{7b} > T_{7c}$).
Intake Stroke: process 6-1

When the intake valve opens the fresh gas with mass $m_i$ mixes with the hotter residual gas with mass $m_R$ so the gas temperature at the end of the intake stroke $T_1$ will be greater than the inlet temperature $T_i$.

Applying conservation of mass

$$m_i = m_1 - m_R = m_1 - m_6$$

And conservation of energy (open system):

$$U_i - U_6 = Q_{6-1} - W_{6-1} + m_i h_i$$

$$m_i u_i - m_6 u_6 = -P_i (V_i - V_6) + m_i h_i$$

$$m_i (h_i - P_i v_i) - m_6 (h_6 - P_6 v_6) = -P_i (V_i - V_6) + (m_i - m_6) h_i$$

$$h_i = \frac{m_6}{m_i} \left[ h_6 + \left( \frac{m_i}{m_6} - 1 \right) h_i + (P_i - P_6) v_6 \right]$$
Intake Gas Temperature ($T_1$)

Assuming ideal gas, $P_6V_6 = RT_6$ and $h = c_pT$ and $m_6 = m_1f$

\[ h_1 = (1 - f)h_i + fh_6 - \left(1 - \frac{P_1}{P_6}\right)fRT_6 \]

\[ T_1 = (1 - f)T_i + fT_6 \left[1 - \left(1 - \frac{P_1}{P_6}\right)\left(\frac{\gamma - 1}{\gamma}\right)\right] \]

In terms of inlet and exhaust conditions $P_1 = P_i$, $P_6 = P_e$, $T_6 = T_e$

\[ T_1 = (1 - f)T_i + fT_e \left[1 - \left(1 - \frac{P_i}{P_e}\right)\left(\frac{\gamma - 1}{\gamma}\right)\right] \]
In real engines valves don’t open and close instantaneously

In order to ensure that the valve is fully open during a stroke for volumetric efficiency, the valves are open for longer than $180^\circ$.

The exhaust valve opens before TC and closes after BC and the intake valve opens before TC and closes after BC.

At TC there is a period of **valve overlap** where both the intake and exhaust valves are open.
When the intake valve opens the cylinder pressure is at $P_e$.

**Part throttle** ($P_i < P_e$): residual gas flows into the intake port. During intake stroke the residual gas is first returned to the cylinder then fresh gas is introduced. Residual gas reduces part load performance.

**WOT** = wide-open-throttle ($P_i = P_e$): some fresh gas can flow out the exhaust valve reducing performance and increasing emissions.

**Supercharged** ($P_i > P_e$): fresh gas can flow out the exhaust valve.
Valve Timing angles

<table>
<thead>
<tr>
<th></th>
<th>Open</th>
<th>Close</th>
<th>Duration</th>
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</thead>
<tbody>
<tr>
<td>Intake</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conventional</td>
<td>5° before tdc</td>
<td>45° after bdc</td>
<td>230°</td>
</tr>
<tr>
<td>High performance</td>
<td>30° before tdc</td>
<td>75° after bdc</td>
<td>285°</td>
</tr>
<tr>
<td>Exhaust</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conventional</td>
<td>45° before bdc</td>
<td>10° after tdc</td>
<td>235°</td>
</tr>
<tr>
<td>High performance</td>
<td>70° before bdc</td>
<td>35° after tdc</td>
<td>285°</td>
</tr>
</tbody>
</table>

Conventional engines operate at low rpms, with idle and part load important.
High performance engines operate at high rpms at WOT, with power and volumetric efficiency important.

At high engine speeds less time available for fresh gas intake which requires more crank angles to get high volumetric efficiency -- large valve overlap.

At low engine speed and part throttle valve overlap is minimized by reducing the angle duration for valves staying open.

Variable Valve Timing used to obtain optimum performance over wide range of engine speed.
Intake and Exhaust Processes in 4-Stroke Cycle

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**WOT**

--- Part throttle
Intake and Exhaust System for Single cylinder engine
The intake manifold is a system designed to deliver air to the engine from a atmosphere to each cylinder through pipes called **runners**.

Exhaust manifold used to duct the exhaust gases from each cylinder to a point of expulsion such as the tail pipe.
The **volumetric efficiency** is defined as: 

\[ \eta_v = \frac{2m_a}{\rho_{a,o} V_d} \]

Volumetric efficiency is affected by:
- Fuel evaporation
- Mixture temperature
- Pressure drop in the intake system
- Gasdynamic effects

Engine speed 

\[ N = \left( S / 2 \right) \cdot \bar{U}_p \]
Factors affecting volumetric efficiency

Fuel evaporation:
In naturally aspirated engines (if there is no supercharging) the volumetric efficiency is always less 100% because fuel is added and the fuel vapour will displace incoming air.

The earlier the fuel is added in the intake system the lower the volumetric efficiency because more of the fuel evaporates before entering the cylinder.

In Diesels fuel is added directly into the cylinder so get a higher efficiency.

Residual gas
As \((P_e/P_i)\) increases, or \(r\) decreases the fraction of cylinder volume occupied by residual gas increases and thus volumetric efficiency decreases.

Opening intake valve before TC (valve overlap):
The longer the valve overlap more exhaust gases are pushed into the intake port.

At lower engine speeds there is more time for exhaust gases to back up – this is a problem for proper operation.
Factors affecting volumetric efficiency

**Heat transfer:**
All intake systems are hotter than ambient air, so the density of the air entering the cylinder is lower than ambient air density.

Injection system and throttle bodies are purposely heated to enhance fuel evaporation.

At lower engine speeds more time is needed for air to be heated – this is a problem.

**Friction:**
The air flows through a duct, an air filter, throttle and intake valve -- a flow restriction undergoes a pressure drop.

The pressure at the cylinder is thus lower than atmospheric pressure – this creates problem at high engine speeds when the air flow velocity is high.
**Closing the intake valve after BC (air backflow):**

When the piston reaches BC there is still a pressure difference across the intake valve and the mixture continues to flow into the cylinder, therefore close the intake valve after TC.

As the piston approaches TC the pressure in the cylinder increases.

As the piston changes direction the air is compressed, when the pressure equals the intake manifold pressure the air flow into the cylinder stops.

Best time to close the intake valve is when the manifold and cylinder pressures are equal, close the valve too early and don’t get full load, too late and air flows back into the intake port.

At high engine speeds larger pressure drop across intake valve because of higher flow velocity, so ideally want to close valve later after BC (60° aBC).

At low engine speeds smaller pressure drop across the intake valve so ideally want to close the intake valve earlier after BC (40° aBC).
The crank angle where the intake valve closes is fixed by the camshaft and cannot change with engine speed. As a result the volumetric efficiency is reduced at the low and high engine speeds.

As the intake valve closes at higher engine speeds, the inertia of the air in the intake system increases the pressure in the intake port -- called the ram effect. This effect becomes more important at higher engine speeds.

To take advantage of ram effect close intake valve later after BC.
Intake and exhaust tuning:

• When the intake valve opens the air suddenly rushes into the cylinder and an expansion wave propagates back to the intake manifold at the local speed of sound relative to the flow velocity.

• When the expansion wave reaches the manifold it reflects back towards to intake valve as a compression wave.

• The time it takes for the round trip depends on the length of the runner and the flow velocity.

• If the timing is appropriate the compression wave arrives at the inlet at the end of the intake process raising the pressure above the nominal inlet pressure allowing more air to be injected.

• For fixed runner length the intake is tuned for one engine speed (flow velocity).

• The exhaust system can be tuned to get a lower pressure at the exhaust valve increasing the exhaust flow velocity.
Factors affecting volumetric efficiency as a function of engine speeds
Flow in the engine cylinder

Large-scale gas motion in the cylinder are characterized by three parameters: **swirl, squish, and tumble.**

**Swirl** is the rotational flow within the cylinder about the axis.

Swirl is used to ensure rapid mixing between fuel and air in direct injection CI engines and for SI engines it is used to promote rapid combustion.

The swirl is generated during air induction into the cylinder by:
- Tangentially directing the flow into the cylinder, or
- Pre-swirling the incoming flow by the use of helical ports.

Some swirl decays due to friction during the engine cycle, but most of it persists through the compression, combustion and expansion processes.
Generation of Cylinder Swirl

- Swirl motion
- Tangential injection
- Helical port
- Contoured valve
Swirl can be modelled as cylinder of gas rotating at an angular velocity, $\omega$.

The swirl ratio, $R_s$, is the ratio of the gas angular velocity and the crank shaft angular velocity:

$$ R_s = \frac{\omega}{2\pi N} $$

$N$ -- the engine speed in revolutions per second; $\omega$ -- air solid-body angular velocity in rad/s.

The mass moment of inertia and angular momentum of the rotating gas cylinder:

$$ I = \frac{mB^2}{8} \quad \Gamma = I\omega $$

$m$ -- mass of the gas; $B$ -- cylinder bore
Angular momentum is conserved if we neglect friction.

Many engines have a wedge shape cylinder head cavity or a bowl in the piston where the gas ends up at TC.

During the compression process as the piston approaches TC more air enters the cavity and the air cylinder moment of inertia decreases and the angular velocity and thus the swirl increase.
**Squish** is the radial flow occurring at the end of the compression stroke in which the compressed gases flow into the cavity in the piston or cylinder head.

![Diagram of Squish](image)

**Tumble** is a secondary flow generated by squish motion rotation about a circumferential axis near the outer edge of the cavity when the piston reaches TC.

![Diagram of Tumble](image)
The gas issues from the valve opening as a conical jet with radial and axial velocities that are about ten times the mean piston velocity.

The jet separates from the valve producing shear layers with large velocity gradients which generate turbulence.

The jet is deflected by the cylinder wall towards the piston and up towards the cylinder head.

Additional turbulence is generated by the velocity gradient at the wall in the boundary layer.
Models often don’t agree with experiments
Most model employ assumptions not satisfied by real flames, e.g.

- Adiabatic (sometimes ok)
- Homogeneous, isotropic turbulence over many LI (never OK)
- Low Ka or high Da (thin fronts) (sometimes OK)
- Lewis number = 1 (sometimes OK, e.g. CH4-air)
- Constant transport properties (never OK -- $\nu$ and $\alpha$ increase in across front)
- Constant density (never OK)
Characteristics of turbulent flames

- Most important property: turbulent flame speed ($S_T$)
- Most models based on physical model of Damköhler
- Turbulent behavior depends on Karlovitz number:
  \[ Ka = \left( \frac{\delta}{L_f} \right)^2 \sim \left( \frac{Re_L}{u'/U_f} \right)^2 \]
  - Kolmogorov turbulence: $u = u'$; $\lambda \sim L/Re^{-3/4}$
- Low $Ka$: “Huygens propagation,” thin fronts that are wrinkled by turbulence but internal structure is unchanged – flamelet.
- High $Ka$: Distributed reaction zones, broad fronts
Low $u'/U_f$: weakly wrinkled flames: $U_T/U_f \approx 1 + (u'/U_f)^2$

This is valid only for periodic flows - for random flows:

$$U_T/U_f - 1 \approx (u'/U_f)^{4/3}$$

Higher $u'/U_f$: strongly wrinkled flames

Schelkin (1947) estimated: $U_T/U_f \approx (u'/U_f)^{1/2}$ for high $u'/U_f$

Other models based on fractals, probability-density functions, etc., mostly predict $U_T/U_f \sim (u'/U_f)$ with the possibility of "bending" or quenching at sufficiently high $Ka$. 
"Thin" flame
(Re \(\gg\) 1, \(K_a \ll\) 1)

"Distributed" flame
(Re \(\gg\) 1, \(K_a \gg\) 1)
Turbulent Flow

Common practice is to define the **turbulent fluctuation intensity** in terms of the root-mean-square of the fluctuations:

\[
    u_t = u'_{\text{rms}} = \sqrt{(u')^2} \quad \Rightarrow \quad \overline{u'^2} = \frac{1}{\Delta t} \int_{t_1}^{t_2} (u'(t))^2 \, dt
\]

- Turbulent flows are always dissipative, viscous shear stresses result in an increase in the internal energy at the expense of its turbulent kinetic energy.
- If no energy is supplied turbulence decays - Energy is required to generate turbulence.
- A source of energy for turbulent velocity fluctuations is shear in the mean flow, e.g., jets and boundary layers.
Turbulence Measurements in Engines

The velocity measurement at a point in the cylinder over time for a two stroke engine (two-stroke cycle has \(2\pi\) crank angles, four-stroke cycle has \(4\pi\) crank angles).

The instantaneous velocity measured at a specific crank angle \(\theta\) in a particular cycle \(i\):

\[
U(\theta, i) = \overline{U}(\theta) + u'(\theta, i)
\]

In engines the flow is periodic, the flow pattern changes with crank angle and the flow is statistically periodic, not steady.
There are both cycle-by-cycle variations in the mean flow at any point in the cycle, as well as turbulent fluctuations about that specific cycle’s mean flow.

One of the most important findings is that at the end of compression when the piston is at TC, the turbulence fluctuating intensity is about one-half the mean piston speed.
The Length-Scales of Turbulence

✓ Turbulent flow is comprised of eddies (vorticies) with a wide range of length scales and vorticities (measure of angular velocity).

✓ The largest eddies in the flow are limited in size by the enclosure with characteristic length-scale of $L$ -- large eddy associated with swirl.

✓ The integral scale $L$ represents the largest turbulent vortex size, determined by the fluctuating velocity frequency.

✓ Most of the kinetic energy of the flow is contained in the large eddies.

✓ The kinetic energy is converted to thermal energy via viscous dissipation.

✓ Viscous forces are only important in the smallest scale where $Re \approx 1$.

✓ The eddy size at which the flow kinetic energy is dissipated by viscous effects is known as the Kolmogorov scale.

✓ Between the integral and Kolmogorov scales there is length-scale -- the Taylor microscale which represents the distance over which viscous effects can be felt, or the mean spacing between dissipative eddies.
The Length-Scales of Turbulence

Integral scale: \( L \)
Taylor micro scale: \( l \)
Kolmogorov scale: \( \lambda \)

\[
\lambda = L \cdot \left( C_\lambda \right)^{-1/4} \text{Re}_t^{-3/4}
\]

\[
\lambda = l \cdot \left( \frac{15}{C_\ell} \right)^{1/2} \text{Re}_t^{-1/2}
\]

\[
\text{Re}_t = \frac{u_L}{\nu}
\]

Numerical coefficients \( C_\lambda \), and \( C_\ell \) are numbers unique to the particular flow.