DI04019011: Thermal Engineering - I

GTU Course Presentation

July 25, 2026

Course Contents I

Course Contents II

Course Contents III

Course Contents IV

Course Contents V

Course Contents VI

Course Contents VII

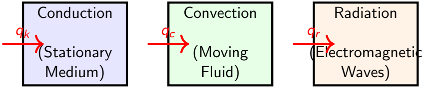
Thermodynamics vs. Heat Transfer I

Thermodynamics determines the heat and work interactions during a transition between equilibrium states. Heat Transfer determines the rate of thermal energy transfer and temperature distribution under non-equilibrium conditions.

Key Points

- Thermodynamics predicts the final state and the feasibility of a process, but does not provide the time rate of heat transfer (Q in Joules, not Watts).
- Heat Transfer is a non-equilibrium phenomenon driven by a temperature gradient (dT/dx). It determines the rate of thermal energy transfer (Q_dot in Watts or J/s).
- Thermal design of engineering systems (e.g., heat exchangers, microchip cooling, turbine blade cooling) requires understanding rate equations and temperature fields.

Modes of Heat Transfer I



Key Points

- Conduction: Transfer of energy from more energetic particles to adjacent less energetic ones through microscopic interactions (collisions, diffusion, free electron mobility).
- Convection: Energy transfer between a solid surface and an adjacent moving fluid, comprising both bulk fluid motion (advection) and conduction.
- Radiation: Energy emitted by matter in the form of electromagnetic waves (or photons) as a result of changes in electronic configurations, requiring no medium.

Fourier's Law of Heat Conduction I

Fourier's Law (1D): q_x = -k * A * (dT / dx) Where: - q_x = Heat transfer rate (W) - k = Thermal conductivity (W/m.K) - A = Cross-sectional area normal to heat flow ($m\hat{2}$) - dT/dx = Temperature gradient (K/m)

Key Points

- The negative sign is a consequence of the Second Law of Thermodynamics, showing that heat flows spontaneously from high temperature to low temperature (dT/dx is negative in the direction of heat flow).
- Thermal conductivity (k) is a transport property of the material. For copper, k $\approx$400 W/m.K; for air, k $\approx$0.026 W/m.K; for polyurethane foam, k $\approx$0.03 W/m.K.
- Heat flux (q"_x = q_x / A = -k * dT/dx) is a directional vector quantity normal to the area.

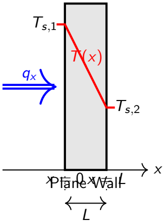
Microscopic Mechanisms of Conduction I

Conduction operates differently across phases. In solids, it is driven by lattice vibrations (phonons) and free electrons. In gases/liquids, it is driven by molecular collisions and diffusion.

Key Points

- Metals have high thermal conductivity (e.g., copper k $\approx$401 W/m.K) due to highly mobile free electrons (Wiedemann-Franz law).
- Gases have low conductivity (e.g., air k $\approx$0.026 W/m.K) because molecules are widely spaced, limiting collision rates.
- Insulators like polyurethane foam or fiberglass trap small pockets of air to minimize both conduction and convection.

Steady-State 1D Plane Wall Conduction I



Key Points

- For a one-dimensional, steady-state system with no internal heat generation, the heat flow rate q_x is constant at any x.
- Integrating Fourier's Law from x=0 (T=T_s1) to x=L (T=T_s2) yields: q_x = (k * A / L) * (T_s1 - T_s2).
- The resulting temperature distribution T(x) is linear across the plane wall thickness.

Thermal Resistance Analogy I

Analogy with Ohm's Law (I = V / R): - Thermal Current: q_x (W) - Driving Potential: Delta T (K) - Conduction Thermal Resistance: R_t,cond = L / (k * A) [K/W]

Key Points

- Allows representation of multi-layer or composite walls as a thermal circuit with resistances in series or parallel.
- For series composite walls, total resistance is the sum of individual resistances: R_tot = R_1 + R_2 + ... + R_n.
- Thermal contact resistance (R_t,c) must be accounted for at joint interfaces due to micro-scale air gaps.

Industrial Application: Furnace Insulation I

Designing a multi-layer kiln wall to limit heat loss and outer skin temperature. Inner wall: Firebrick ($k = 1.4$ W/m.K, $L = 0.22$ m). Outer wall: Rock wool insulation ($k = 0.05$ W/m.K, $L = 0.12$ m).

Key Points

- Firebrick conduction resistance: $R_1 = L_1 / (k_1 * A) = 0.157 / A$ K/W.
- Insulation conduction resistance: $R_2 = L_2 / (k_2 * A) = 2.4 / A$ K/W. The insulation layer accounts for ¿93% of total resistance.
- Optimal insulation thickness minimizes fuel usage, reduces outer surface temperature for safety, and prevents structural thermal fatigue.

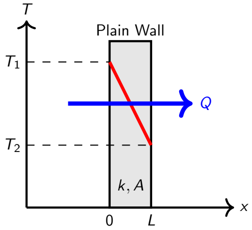
Fourier's Law of Heat Conduction I

Fourier's Law states that the rate of heat transfer through a material is proportional to the negative gradient in the temperature and to the area normal to the direction of flow.

Key Points

- Mathematical formula: Q = -k * A * (dT/dx), where Q is heat transfer rate (W).
- Thermal conductivity (k) is a transport property measured in W/(m·K) indicating heat conduction capability.
- Negative sign enforces the Second Law of Thermodynamics: heat spontaneously flows from high to low temperature.
- Heat flux (q = Q/A = -k * dT/dx) represents heat transfer per unit area (W/m2̂).
- Assumptions: Homogeneous and isotropic material, steady-state condition, and one-dimensional heat flow.

Conduction through a Plain Wall I



Conduction through a Plain Wall II

Key Points

- One-dimensional, steady-state heat conduction with no internal heat generation.
- Integration of Fourier's Law yields a linear temperature profile: T(x) = T_1 - (T_1 - T_2) * (x/L).
- Heat transfer rate: Q = k * A * (T_1 - T_2) / L.
- Thermal resistance analogy (Ohm's Law): R_cond = L / (k * A) in K/W.
- Thermal conductivity k is assumed constant; in reality, it may vary with temperature.

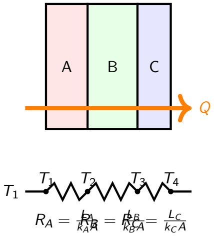
Conduction through a Composite Wall I

A composite wall consists of multiple layers of different materials in series and/or parallel configurations, typical of building envelopes, furnaces, and cryogenic tanks.

Key Points

- Under steady-state conditions, the rate of heat transfer (Q) remains constant through each consecutive layer.
- For a series composite wall of three layers: $Q = (T_1 - T_4) / (R_1 + R_2 + R_3)$ where $R_i = L_i / (k_i * A)$.
- Interface temperatures (T_2, T_3) can be determined by matching heat rates: $Q = (T_1 - T_2)/R_1 = (T_2 - T_3)/R_2$.
- Thermal contact resistance at the layer interfaces is neglected in the simplified analysis but is critical in aerospace applications.
- Industrial application: Refractory firebrick lined with insulating brick and protected by a steel casing.

Thermal Resistance Network for Composite Wall I



Thermal Resistance Network for Composite Wall II

Key Points

- Electrical analogy allows representation of multi-layer conduction as a network of resistors in series.
- Total equivalent thermal resistance: $R\_total = R\_A + R\_B + R\_C$.
- Heat flow is calculated as the total temperature difference divided by the total thermal resistance.
- Allows direct calculation of heat loss without solving for intermediate boundary temperatures first.
- In systems with parallel heat paths, equivalent resistance is calculated using parallel resistor rules: $1/R\_eq = 1/R\_1 + 1/R\_2$.

Conduction through a Cylindrical Wall I

Radial heat transfer occurs in cylindrical geometries such as pipes, pressure vessels, and tubes, where the area normal to heat flow increases with the radius.

Conduction through a Cylindrical Wall II

Key Points

- The cross-sectional area normal to heat flow is a function of radius: A(r) = 2 * pi * r * L.
- Steady-state radial heat transfer rate (no derivation): Q = 2 * pi * k * L * (T_1 - T_2) / ln(r_2/r_1).
- The temperature profile inside a cylindrical wall is logarithmic rather than linear.
- Cylindrical thermal resistance: R_cond,cyl = ln(r_2/r_1) / (2 * pi * k * L) in K/W.
- Critical thickness of insulation: Adding insulation to a cylinder increases conduction resistance but decreases convection resistance. The critical radius is r_cr = k_ins / h.

Convection involves the cumulative effects of conduction (diffusion) and bulk fluid motion (advection) to transfer thermal energy between a solid surface and a fluid.

Key Points

- Governed by Newton's Law of Cooling: $Q = h * A * (T_s - T_{inf})$, where h is the convective heat transfer coefficient.
- Convective resistance: $R_{conv} = 1 / (h * A)$ in K/W.
- The heat transfer coefficient (h) is not a fluid property; it depends on flow velocity, geometry, turbulence, and fluid properties.
- Free Convection: Fluid motion is driven by buoyancy forces arising from density gradients (e.g., hot pipe in quiescent room air).
- Forced Convection: Fluid is driven over the surface by external mechanical devices like fans, blowers, or pumps.

Combined Conduction-Convection Systems I

Practical thermal designs (e.g., heat exchangers, steam pipes) involve heat transferring from a hot fluid, through a solid barrier, to a cold fluid.

Combined Conduction-Convection Systems II

Key Points

- Total heat transfer rate: Q = U * A * dT_overall, where U is the Overall Heat Transfer Coefficient (W/m2̂·K).
- For a plain wall with convection on both sides: 1/U = 1/h_inner + L/k + 1/h_outer.
- For a cylindrical pipe with convection on both sides: 1/(U_o * A_o) = 1/(h_i * A_i) + ln(r_o/r_i)/(2 * pi * k * L) + 1/(h_o * A_o).
- The design goal is typically to minimize U (for thermal insulation) or maximize U (for efficient heat exchangers).
- Fouling factors (additional thermal resistance from scaling or dirt accumulation) are added to the resistance network in industrial heat exchangers.

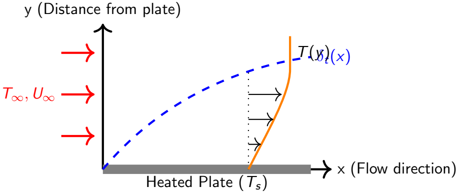
Physical Mechanism of Convection I

Convection is the transfer of thermal energy between a solid surface and an adjacent moving fluid. It is a combined mechanism of heat conduction (molecular diffusion) and advection (bulk fluid motion).

Key Points

- At the solid-fluid interface ($y = 0$), the fluid particles are stationary relative to the surface due to the no-slip condition.
- Consequently, heat transfer from the wall to the immediately adjacent fluid layer occurs purely by conduction.
- As we move away from the wall, bulk motion of the fluid carries the thermal energy downstream (advection), drastically increasing the heat dissipation rate.
- The convective heat transfer rate depends highly on dynamic viscosity (μ), thermal conductivity (), *density*(ρ), *specificheat*(), and velocity profile.

Thermal Boundary Layer Development I



Thermal Boundary Layer Development II

Key Points

- The thermal boundary layer (δ_t) is defined as the region where the temperature gradient - T$iswithin 99\% of - T_\infty$.
- The temperature gradient is steepest at the wall surface ($y = 0$), where heat flux is modeled by Fourier's law: $q'' = -k_f \cdot (\frac{dT}{dy})_{y=0}$.
- As fluid travels along the plate (increasing x), thermal energy penetrates deeper into the free stream, causing $\delta_t(x)$ to grow.
- Thicker boundary layers provide greater thermal resistance, which decreases the local heat transfer coefficient downstream.

Newton's Law of Cooling I

Newton's law of cooling provides a macroscopic parameterization for convection, stating that heat transfer rate is directly proportional to surface-to-fluid temperature difference.

Newton's Law of Cooling II

Key Points

- Mathematical formulation: $Q = h \cdot A_s \cdot (T_s - T_\infty)$ [W], where h is the convective heat transfer coefficient in W/(m$^2\cdot$ K).
- Heat flux version: $q'' = Q/A_s = h \cdot (T_s - T_\infty)$ [W/m^2], where T_s is surface temperature and T_∞ is the undisturbed fluid temperature.
- Unlike thermal conductivity (k), the coefficient h is not a property of the fluid; it is a system parameter determined by boundary layer characteristics.
- Local heat transfer coefficient h_x varies along the flow path, requiring definition of an average coefficient h_{avg} for the entire surface.

Free (Natural) Convection I

Free convection occurs when fluid motion is induced solely by buoyancy forces resulting from density gradients, which in turn are caused by temperature variations.

Key Points

- No external driving force (such as a fan or pump) is applied; gravity acts on density differences within the fluid matrix.
- Warm, less dense fluid adjacent to a hot wall rises, whereas cooler, denser fluid falls, generating a natural circulatory current.
- Typical heat transfer coefficients (h) for free convection are low: 2 to 25 $W/(m^2 \cdot K)$ for gases, and 50 to 1000 $W/(m^2 \cdot K)$ for liquids.
- The flow regime is governed by the Grashof number (Gr), representing the ratio of buoyancy forces to viscous forces, and the Prandtl number (Pr).

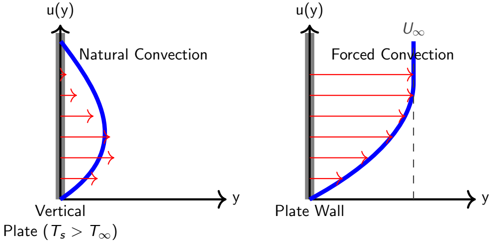
Forced Convection I

Forced convection occurs when fluid flow is driven by external mechanical means such as pumps, fans, blowers, or wind currents.

Key Points

- External velocity increases advective transport, thinning both velocity and thermal boundary layers at the heat exchange surfaces.
- Typical heat transfer coefficients (h) are high: 25 to 250 W/($m^2\cdot$ K) for forced air/gas, and 100 to 20,000 W/($m^2\cdot$ K) for liquids.
- Flow dynamics are characterized by the Reynolds number (Re $=$ ρ* V * D / μ), indicating transition from laminar to turbulent regimes.
- Turbulent forced convection utilizes rapid eddy mixing, greatly enhancing the convective heat transfer coefficient.

Velocity Profiles: Free vs. Forced Convection I



Velocity Profiles: Free vs. Forced Convection II

Key Points

- In forced convection, velocity rises from zero at the wall to the free-stream velocity (U_∞) asymptotically.
- In free convection, velocity is zero at the wall (no-slip) and zero in the ambient fluid, forming a velocity spike close to the wall.
- The location and magnitude of peak velocity in natural convection depend directly on local Grashof numbers.
- Different velocity boundary profiles alter shear stress distributions and local heat transfer rates between forced and free convection.

Industrial Design Case Study I

Problem: A high-power industrial transistor dissipates 15 W of heat. The casing surface area is $A_s = 0.002$ m^2. The maximum allowable casing temperature is $T_s = 80°$C, in an ambient air environment of $T_\infty = 25°$C.

Key Points

- We apply Newton's law of cooling to calculate the required convection coefficient: $Q = h \cdot A_s \cdot (T_s - T_\infty)$.
- Rearranging yields: $h = Q/(A_s \cdot (T_s - T_\infty)) = 15$ W/(0.002 m^2 × (80°C – 25°C)).
- Calculation results in: $h = 15/(0.002 \times 55) = 136.36$ W/(m^2· K).
- Engineering decision: Since natural convection in air yields $= 2to5$ W/(m^2· K), natural air cooling is insufficient. An active cooling fan (forced convection) is mandatory.

Introduction to Thermal Radiation I

Thermal radiation is the electromagnetic radiation emitted by a body as a consequence of its temperature. Unlike conduction and convection, which require a physical medium to transfer energy, radiation can propagate through a perfect vacuum. This mechanism is crucial in space applications, vacuum furnaces, and atmospheric heat balances.

Key Points

- Radiation travels at the speed of light ($c_0 \approx 3 \times 10^8$ m/s in vacuum), meaning there is no thermal lag during propagation.
- The emission of thermal radiation is a volumetric phenomenon for gases and semi-transparent solids, but is treated as a surface phenomenon for most solids.
- The wavelengths of interest for thermal radiation range from 0.1 to 100 μm, which overlaps with the ultraviolet, visible, and infrared bands.
- The maximum rate of radiation emission from a surface at absolute temperature T is governed by the Stefan-Boltzmann law: $E_b = \sigma T^4$.

Concept of Incident Radiation (Irradiation) I

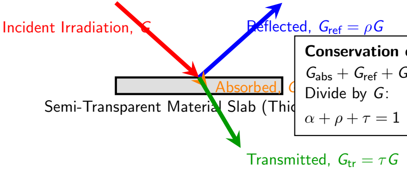
Irradiation (G) is the total rate at which radiation is incident on a surface per unit area from all directions. It can originate from various sources, such as direct solar rays, surrounding walls, or high-temperature combustion gases. When irradiation strikes a surface, the energy is distributed among three paths: absorption, reflection, and transmission.

Concept of Incident Radiation (Irradiation) II

Key Points

- The units of total irradiation G are watts per square meter (W/m^2).
- Spectral irradiation (G_λ) is the radiation incident per unit wavelength interval $d\lambda$ about the wavelength λ.
- The total irradiation is computed by integrating the spectral irradiation over all wavelengths: $G = \int_0^\infty G_\lambda d\lambda$.
- In furnace design, the irradiation from combustion products determines the heat load on the boiler tubes.

Distribution of Incident Energy I



Conservation of Energy:
$G_{abs} + G_{ref} + G_{tr} = G$
Divide by G:
$\alpha + \rho + \tau = 1$

Distribution of Incident Energy II

Key Points

- Graphical representation of a semi-transparent slab dividing incident irradiation (G) into absorbed, reflected, and transmitted parts.
- Conservation of energy dictates that the sum of the rates of absorption, reflection, and transmission must equal the total rate of incident energy.
- For most solid materials in engineering, the transmissivity τ is zero, and they are termed 'opaque' surfaces, which simplifies the balance to $\alpha + \rho = 1$.
- Reflection occurs at the surface boundary, absorption happens within the bulk, and transmission occurs through the boundary into the next medium.

Absorptivity, Reflectivity, and Transmissivity I

These properties characterize how a surface interacts with incident radiation. They are dimensionless ratios between 0 and 1: - Absorptivity (α): The fraction of incident radiation absorbed by the medium. - Reflectivity (ρ): The fraction of incident radiation reflected by the surface. - Transmissivity (τ): The fraction of incident radiation transmitted through the medium.

Absorptivity, Reflectivity, and Transmissivity II

Key Points

- Absorptivity is defined mathematically as $\alpha = G_{\text{abs}}/G$. A high α is desired for solar absorbers.
- Reflectivity is defined as $\rho = G_{\text{ref}}/G$. It is highly dependent on surface finish and roughness.
- Transmissivity is defined as $\tau = G_{\text{tr}}/G$. It is crucial for optical lenses, window glass, and greenhouse gases.
- Each property has a spectral equivalent (α_λ, ρ_λ, τ_λ) which evaluates properties at a specific wavelength.

Reflective Behavior and Surface Characteristics I

Reflectivity is strongly influenced by surface morphology. Two limiting cases of reflection are studied: 1. Specular Reflection: Occurs on highly polished, mirror-like surfaces. The angle of reflection equals the angle of incidence. 2. Diffuse Reflection: Occurs on rough surfaces. The incident rays are scattered equally in all directions, independent of the angle of incidence.

Reflective Behavior and Surface Characteristics II

Key Points

- For specular reflection, the surface roughness is much smaller than the wavelength of the incident radiation ($d \ll \lambda$).
- For diffuse reflection, the surface roughness is comparable to or larger than the wavelength ($d \geq \lambda$).
- Most practical engineering materials (cast iron, oxidized metals, concrete) behave as diffuse surfaces for thermal radiation.
- Assuming diffuse surfaces greatly simplifies radiation analysis because the intensity of reflected radiation is uniform in all directions.

Kirchhoff's Law and Blackbody Concept I

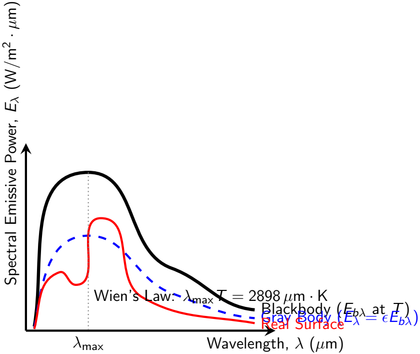
Kirchhoff's Law states that at thermal equilibrium, the spectral emissivity of a surface equals its spectral absorptivity: $\epsilon_\lambda = \alpha_\lambda$. If the surface is diffuse or the incident radiation is blackbody radiation at the same temperature, the total hemispherical properties are equal: $\epsilon = \alpha$.

Kirchhoff's Law and Blackbody Concept II

Key Points

- A blackbody is an idealized physical body that absorbs all incident electromagnetic radiation, regardless of frequency or angle of incidence.
- A gray body is defined as a surface whose spectral properties are independent of wavelength (i.e., ϵ_λ and α_λ are constant).
- For a gray body, the total emissivity is equal to the total absorptivity ($\epsilon = \alpha$), even if the system is not in thermal equilibrium.
- Kirchhoff's law is a powerful tool, allowing engineers to estimate a material's absorption capacity by characterizing its emission behavior.

Spectral Emissive Power Curves I



Spectral Emissive Power Curves II

Key Points

- Comparison of spectral emission curves for a blackbody, a gray body, and a real surface at a constant temperature.
- Planck's distribution describes the blackbody spectral emissive power, peaking at wavelength λ_{max} governed by Wien's Displacement Law.
- A gray body emits a constant fraction ϵ of the blackbody's power at every wavelength, resulting in a smooth scaled curve.
- Real surfaces exhibit irregular curves with peaks and troughs due to wavelength-dependent properties and molecular band structures.

The Ideal Radiation Surface: The Blackbody I

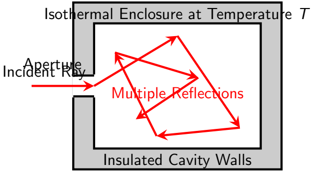
A blackbody is an idealized physical body that absorbs all incident electromagnetic radiation, regardless of wavelength or direction. It serves as a standard of comparison for real surfaces.

The Ideal Radiation Surface: The Blackbody II

Key Points

- Definition: A perfect emitter and perfect absorber. No surface can emit more energy than a blackbody at a given temperature and wavelength.
- Isotropic Emissivity: The radiation emitted by a blackbody is independent of direction (diffuse emitter).
- Cavity Realization: Practically approximated by a small hole (aperture) in an isothermal enclosure (cavity or Hohlraum) where internal reflections ensure absorption.
- Planck's Distribution Law: Describes the spectral blackbody emissive power $E_{b\lambda}$ as a function of absolute temperature and wavelength.

Practical Realization: The Isothermal Cavity I



Practical Realization: The Isothermal Cavity II

Key Points

- A small opening in an isothermal cavity behaves as a blackbody: $\alpha_{aperture} \approx 1$.
- Any ray entering the aperture undergoes multiple internal reflections, with absorption occurring at each bounce.
- By the time the ray could reflect back out of the aperture, its energy is almost completely dissipated and absorbed.
- This concept is used to calibrate industrial pyrometers and design reference calibration furnaces.

Emissive power is the total thermal radiation energy emitted by a surface per unit area per unit time over all wavelengths and in all directions. It is expressed in W/m2̂.

Key Points

- Stefan-Boltzmann Law: For a blackbody, the total emissive power is given by $E_b = \sigma T^4$, where $\sigma = 5.67 \times 10^{-8}$ W/(m^{2}K^4).
- Spectral Emissive Power (E_λ): The rate of radiation energy emitted per unit surface area per unit wavelength interval about λ.
- Integration: The total emissive power is the integral of spectral emissive power over all wavelengths: $E = \int_0^\infty E_\lambda d\lambda$.
- Wien's Displacement Law: The wavelength at which the spectral emissive power is maximum shifts toward shorter wavelengths as temperature increases: $\lambda_{max} T = 2897.8\ \mu\text{m} \cdot \text{K}$.

Emissivity and Real Surfaces I

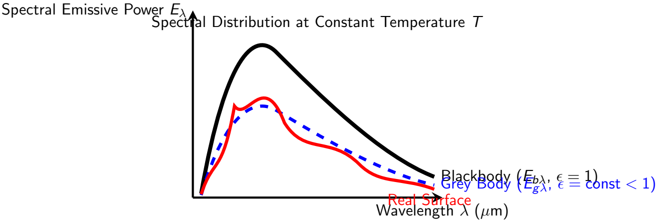
Emissivity (ε) is a surface property defined as the ratio of the radiation emitted by the surface to the radiation emitted by a blackbody at the same temperature.

Emissivity and Real Surfaces II

Key Points

- Spectral Directional Emissivity: $\varepsilon_{\lambda,\theta,\phi}(\lambda, \theta, \phi, T) = E_{\lambda,\theta}(\lambda, \theta, \phi, T)/E_{b,\lambda}(\lambda, T)$. Represents local emission efficiency.
- Hemispherical Emissivity: Integrates over the entire hemisphere. Often simplified to total hemispherical emissivity: $\varepsilon(T) = E(T)/E_b(T)$.
- Range: $0 \leq \varepsilon \leq 1$. For a blackbody, $\varepsilon = 1$. For real surfaces, $\varepsilon < 1$ and varies with surface temperature, roughness, oxidation state, and wavelength.
- Engineering Significance: Low emissivity materials (e.g., polished metals like gold, aluminum: $\varepsilon \approx 0.03 - 0.08$) reflect heat, while oxidized surfaces and paints have high emissivity ($\varepsilon \approx 0.8 - 0.95$).

Black, Grey, and Real Surface Characteristics I



Black, Grey, and Real Surface Characteristics II

Key Points

- A Blackbody represents the upper limit of spectral emission ($E_{b\lambda}$) at all wavelengths.
- A Grey Body has a spectral emissive power that is a constant fraction of the blackbody's emissive power across all wavelengths: $\varepsilon_\lambda =$ constant.
- A Real Surface exhibits highly irregular, wavelength-dependent emissivity ($\varepsilon_\lambda \neq$ constant) due to electronic transitions and surface contamination.
- The grey body simplification is critical in engineering calculations, as it allows total hemispherical radiation heat transfer modeling without complex spectral integration.

The Grey Body Approximation & Kirchhoff's Law I

A grey body is an idealized surface where the spectral emissivity (ε_λ) and spectral absorptivity (α_λ) are independent of wavelength and direction.

The Grey Body Approximation & Kirchhoff's Law II

Key Points

- Kirchhoff's Law of Radiation: Under thermal equilibrium, the total hemispherical emissivity of any surface is equal to its total hemispherical absorptivity: $\varepsilon(T) = \alpha(T)$.
- Grey Body Validation: For a grey body, $\varepsilon = \alpha$ holds even when the incoming radiation spectrum is not in thermal equilibrium with the surface.
- Simplification in Net Radiation Exchange: The net heat exchange between two diffuse, grey surfaces in an enclosure can be calculated using algebraic electrical network analogy.
- Limitations: Real materials deviate from grey behavior when exposed to source temperatures vastly different from their own (e.g., solar radiation ~ 5800 K falling on a surface at 300 K).

Engineering Application and Thermal Calculations I

Applying emissivity and grey/black body models to solve industrial thermal problems like spacecraft thermal control, combustion chamber design, and pyrometry.

Engineering Application and Thermal Calculations II

Key Points

- Spacecraft Thermal Control: Multi-Layer Insulation (MLI) blankets utilize highly polished aluminum foil ($\varepsilon \approx 0.04$) to minimize radiative heat loss to deep space.
- Combustion Furnaces: Internal walls are coated with high-emissivity ceramic refractory coatings ($\varepsilon \approx 0.92$) to maximize radiative heat transfer to process tubes.
- Example Calculation: A grey surface at 800°C (1073 K) has $\varepsilon = 0.6$. Its emissive power is $E = \varepsilon\sigma T^4 = 0.6 \times (5.67 \times 10^{-8}) \times (1073)^4 \approx 45.1\ \mathrm{kW/m^2}$.
- Pyrometer Calibration: Radiation pyrometers measure target emissive power. They must be calibrated using blackbody cavity sources and adjusted for target emissivity ε.

Detailed Concept 1 I

Detailed engineering explanation of Thermal insulation/insulators.

Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

Detailed Concept 2 I

Detailed engineering explanation of Thermal insulation/insulators.

Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

Detailed Concept 3 I

Detailed engineering explanation of Thermal insulation/insulators.

Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

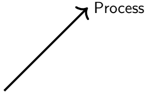
Detailed Concept 4 I

Detailed engineering explanation of Thermal insulation/insulators.

Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

Detailed Concept 5 I



Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

Detailed Concept 6 I

Detailed engineering explanation of Thermal insulation/insulators.

Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

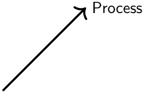
Detailed Concept 7 I

Detailed engineering explanation of Thermal insulation/insulators.

Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

Detailed Concept 8 I



Key Points

- Fundamental principle 1
- Engineering application 2
- Important consideration 3

Concept of Heat Exchangers I

A heat exchanger is a specialized thermal device designed to transfer thermal energy between a hot fluid and a cold fluid without mixing them (in indirect contact systems) or with direct contact. The process is governed by the principles of conduction and convection.

Concept of Heat Exchangers II

Key Points

- Primary objective: Heat recovery, thermal control, phase change (condensation/evaporation), or process optimization.
- Overall Heat Transfer Coefficient (U) accounts for inner convection, wall conduction, outer convection, and fouling resistances.
- Fouling Factor (R_f) represents thermal resistance due to scaling, chemical deposits, and corrosion over operational time.
- Governing design equation: $Q = U \cdot A \cdot \Delta T_m$, where Q is the heat transfer rate, A is the area, and ΔT_m is the mean temperature difference.

Classification of Heat Exchangers I

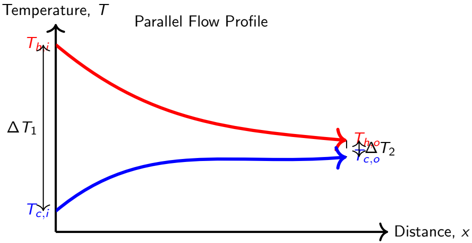
Industrial heat exchangers are classified based on their transfer process, flow configuration, construction geometry, and surface compactness.

Classification of Heat Exchangers II

Key Points

- By Transfer Process: Recuperators (continuous flow through wall) and Regenerators (alternate hot/cold flow through matrix).
- By Construction: Tubular (double-pipe, shell-and-tube), Plate-type (gasketed, brazed, spiral), and Extended Surface (compact/finned configurations).
- By Flow Arrangement: Parallel-flow (co-current), Counter-flow (counter-current), and Cross-flow (perpendicular paths, mixed/unmixed).
- Industrial Applications: Condensers in steam power plants, evaporators in refrigeration, radiators in automobiles, and preheaters in chemical processing.

Parallel Flow Temperature Profile I

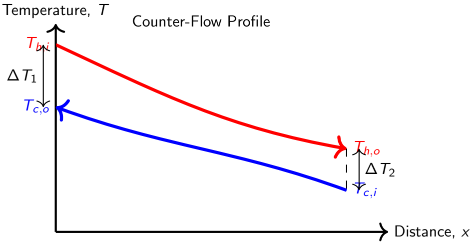


Parallel Flow Temperature Profile II

Key Points

- In a parallel flow arrangement, both hot and cold fluids enter at the same side ($x = 0$) and flow in the same direction.
- The inlet temperature difference is maximum: $\Delta T_1 = T_{h,i} - T_{c,i}$.
- The outlet temperature difference is minimum: $\Delta T_2 = T_{h,o} - T_{c,o}$.
- The temperature of the cold fluid can never reach or exceed the outlet temperature of the hot fluid ($T_{c,o} < T_{h,o}$).

Counter-Flow Temperature Distribution I



Counter-Flow Temperature Distribution II

Key Points

- In a counter-flow arrangement, fluids enter from opposite ends; hot fluid at $x = 0$ and cold fluid at $x = L$.
- Temperature difference definitions: $\Delta T_1 = T_{h,i} - T_{c,o}$ at the hot fluid inlet, and $\Delta T_2 = T_{h,o} - T_{c,i}$ at the hot fluid outlet.
- Allows the outlet temperature of the cold fluid to exceed the outlet temperature of the hot fluid ($T_{c,o} > T_{h,o}$).
- Counter-flow provides a more uniform temperature difference along the path, maximizing thermodynamic efficiency.

Log Mean Temperature Difference (LMTD) I

The Log Mean Temperature Difference (LMTD) is the mathematically rigorous average driving force for heat transfer, assuming constant specific heats, constant overall heat transfer coefficient, and negligible heat loss to surroundings.

Log Mean Temperature Difference (LMTD) II

Key Points

- LMTD Formula: $\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$, where ΔT_1 and ΔT_2 are terminal temperature differences.
- For identical inlet and outlet conditions, the LMTD for counter-flow is always greater than or equal to that of parallel flow.
- A larger LMTD requires a smaller heat transfer surface area (A) to achieve the same rate of heat transfer (Q).
- If $\Delta T_1 = \Delta T_2$ (balanced counter-flow), the formula is indeterminate, and the arithmetic difference is used: $\Delta T_{lm} = \Delta T_1 = \Delta T_2$.

Numerical Example: Parallel Flow Heat Exchanger I

Problem: A parallel flow heat exchanger cools hot engine oil ($c_p = 2.1$ kJ/kg.K) from 150°C to 80°C. Cooling water ($c_p = 4.184$ kJ/kg.K) enters at 25°C and leaves at 60°C. Compute the Log Mean Temperature Difference.

Key Points

- Identify terminal temperatures for parallel flow: $T_{h,i} = 150^\circ$C, $T_{h,o} = 80^\circ$C, $T_{c,i} = 25^\circ$C, $T_{c,o} = 60^\circ$C.
- Calculate inlet temperature difference: $\Delta T_1 = T_{h,i} - T_{c,i} = 150 - 25 = 125^\circ$C.
- Calculate outlet temperature difference: $\Delta T_2 = T_{h,o} - T_{c,o} = 80 - 60 = 20^\circ$C.
- Substitute into LMTD formula: $\Delta T_{lm} = \frac{125-20}{\ln(125/20)} = \frac{105}{\ln(6.25)} = \frac{105}{1.8326} \approx 57.29^\circ$C.

Numerical Example: Counter Flow Comparison I

Problem: Using the same process parameters ($T_{h,i} = 150^\circ$C, $T_{h,o} = 80^\circ$C, $T_{c,i} = 25^\circ$C, $T_{c,o} = 60^\circ$C), determine the LMTD if the heat exchanger is reconfigured to operate in counter-flow.

Numerical Example: Counter Flow Comparison II

Key Points

- Identify terminal temperatures for counter-flow: ΔT_1 is at $x = 0$, ΔT_2 is at $x = L$.
- Calculate hot-end difference:
  $\Delta T_1 = T_{h,i} - T_{c,o} = 150 - 60 = 90^\circ\text{C}$.
- Calculate cold-end difference:
  $\Delta T_2 = T_{h,o} - T_{c,i} = 80 - 25 = 55^\circ\text{C}$.
- Substitute into LMTD formula:
  $\Delta T_{lm} = \frac{90-55}{\ln(90/55)} = \frac{35}{\ln(1.6364)} = \frac{35}{0.4925} \approx 71.07^\circ\text{C}$.
- Engineering Conclusion: Reconfiguring to counter-flow yields a 24% higher LMTD, reducing the required heat transfer area by approximately 19.4% for the same heat duty.

Thermodynamic Concepts of Air Compression I

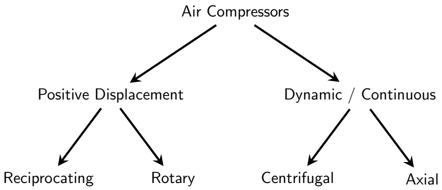
An air compressor is a mechanical device that increases the pressure of air or gas by reducing its volume. This process requires mechanical work input, typically from an electric motor or internal combustion engine. The compression process is governed by fundamental thermodynamic relations, specifically the polytropic process $PV^n = C$.

Thermodynamic Concepts of Air Compression II

Key Points

- Working Medium: Air behaves as a real gas, but at moderate pressures and temperatures, it is analyzed using the ideal gas equation of state: $PV = mRT$.
- Thermodynamic Work: The work required for compression is given by the integral $W = \int_{P_1}^{P_2} VdP$. Minimizing this work is a key objective in compressor design.
- Isothermal vs. Adiabatic: Compression can be isothermal ($PV = C$, minimum work, requires perfect cooling) or adiabatic/isentropic ($PV^\gamma = C$, no heat transfer, maximum work). Actual compression is polytropic ($PV^n = C$, where $1 < n < \gamma$).
- Volumetric Efficiency: The ratio of actual free air delivered to the swept volume of the compressor, affected by clearance volume, pressure ratio, and temperature rise.

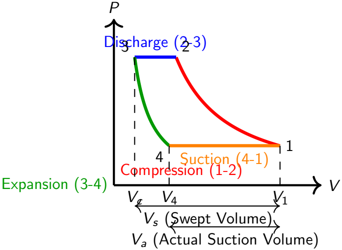
Classification Tree of Air Compressors I



Key Points

- Positive Displacement: Compresses a fixed volume of air in a sealed chamber. Pressure increases as the chamber volume decreases (e.g., reciprocating piston, rotary screw, rotary vane).
- Dynamic Compressors: Accelerates the air using high-speed rotating impellers, converting kinetic energy to static pressure (e.g., centrifugal and axial flow compressors).
- Operating Ranges: Positive displacement is ideal for high pressures at low-to-medium flow rates; dynamic is suitable for low-to-medium pressures at high flow rates.

Reciprocating Compressors: Principles and P-V Diagram I



Reciprocating Compressors: Principles and P-V Diagram II

Key Points

- Working Cycle: Comprises suction, compression, delivery, and expansion of air trapped in the clearance volume (V_c).
- Clearance Volume (V_c): Essential in practical designs to prevent the piston from striking the cylinder head; typically 3% to 10% of swept volume (V_s).
- Volumetric Efficiency (η_{vol}): Defined as $\eta_{vol} = \frac{V_a}{V_s} = 1 - V_c/V_s[(\frac{P_2}{P_1})^{1/n} - 1]$. High pressure ratios drastically reduce volumetric efficiency.
- Work Input: Net work done per cycle is represented by the enclosed area 1-2-3-4 of the P-V diagram: $W = \frac{n}{n-1}P_1 V_a[(\frac{P_2}{P_1})^{\frac{n-1}{n}} - 1]$.

Rotary Compressors: Concepts and Working Principles I

Rotary compressors are positive displacement machines that use rotating elements (screws, vanes, or lobes) to achieve compression. Unlike reciprocating compressors, they do not require valves or piston-crank mechanisms, enabling continuous pulsation-free flow and operation at higher rotational speeds.

Rotary Compressors: Concepts and Working Principles II

Key Points

- Rotary Screw Compressors: Consist of intermeshing male and female helical rotors. As they rotate, the volume between the lobes decreases axially, compressing the trapped air. Widely used for continuous industrial air demand.
- Rotary Vane Compressors: Feature an eccentrically mounted rotor with sliding radial vanes inside a cylindrical casing. Centrifugal force pushes the vanes out, forming chambers of decreasing volume during rotation.
- Key Advantages: Lower vibration, higher reliability (fewer moving parts), compact footprint, and 100% duty cycle compared to reciprocating machines.
- Cooling and Lubrication: Often oil-injected to provide sealing, lubrication, and heat dissipation, keeping discharge temperatures low (around 80-100°C).

Dynamic Compressors: Centrifugal and Axial I

Dynamic compressors increase air pressure by first accelerating it to high velocities using rotating blades (dynamic energy) and then diffusing it to convert kinetic energy into static pressure. They are characterized by continuous flow, large volume capabilities, and oil-free discharge.

Dynamic Compressors: Centrifugal and Axial II

Key Points

- Centrifugal Compressors: Air enters axially and is thrown radially outward by an impeller. A diffuser surrounds the impeller to convert high velocity into static pressure. Used in process industries and turbochargers.
- Axial Flow Compressors: Air flows parallel to the rotating shaft through alternating rows of rotating blades (rotors) and stationary blades (stators). Each stage provides a small pressure rise, but multiple stages achieve high pressure ratios.
- Comparison: Centrifugal compressors are more robust and handle higher pressure ratios per stage (up to 4:1 or 8:1), whereas axial compressors offer higher mass flow rates and superior efficiency at design points (common in gas turbines).
- Operational Limits: Subject to aerodynamic phenomena such as choking (sonic velocity limit) and surging (flow reversal due to high backpressure), which can cause catastrophic mechanical failure.

Industrial Applications of Compressed Air I

Compressed air is often referred to as the 'fourth utility' in industrial plants, alongside electricity, water, and gas. It is vital across multiple sectors, classified by the pressure range and quality (purity) of the air required.

Industrial Applications of Compressed Air II

Key Points

- Pneumatic Tools & Automation: Operating hammers, drills, grinders, assembly line actuators, and pneumatic valves at standard factory pressures (6 to 8 bar).
- Process Air: Air-mixing in chemical reactors, spray painting, powder conveying, and glass blowing, requiring oil-free and dry air to prevent contamination.
- High-Pressure Applications: Bottle blowing (PET), engine starting systems, and breathing air tanks, operating at pressures up to 30 to 40 bar or higher.
- Gas Turbine & Propulsion: Axial compressors are integral components of gas turbine power plants and jet engines, supplying high-pressure air directly to the combustion chamber.

Selection Criteria and Performance Metrics I

Selecting the correct compressor type for an industrial application requires evaluating operating pressure, volume flow rate, air quality, duty cycle, and total lifecycle costs. Matching the compressor characteristic to system demand is critical for energy conservation.

Selection Criteria and Performance Metrics II

Key Points

- Free Air Delivery (FAD): The volume flow rate of compressor discharge converted back to ambient inlet temperature and pressure conditions. Expressed in m^3/min or CFM.
- Pressure vs. Flow Rate Suitability: Reciprocating is selected for very high pressure (> 100 bar) and low flow; Rotary Screw for medium pressure ($7 - 15$ bar) and continuous flow; Centrifugal/Axial for low-to-medium pressure and very high flow rates.
- Power Consumption: Evaluated using Specific Power Consumption (SPC), defined as shaft input power per unit FAD (e.g., $kW/(m^3/min)$). Reducing SPC directly lowers operating costs.
- Air Quality (ISO 8573-1): Classifies compressed air based on contaminants (solid particles, water, and oil). Food, beverage, and pharmaceutical industries require Class 0 oil-free air.

Reciprocating compressors use a piston-cylinder mechanism to compress air. It is a positive displacement machine where the pressure rise is achieved by reducing the volume of the compression chamber.

Introduction & Working Principle II

Key Points

- Positive displacement action: Air is trapped in a cylinder and its volume is physically reduced by a moving piston, which raises the pressure according to thermodynamic gas laws.
- Single-acting vs Double-acting: Compression occurs on only one side of the piston in single-acting, whereas both sides are active in double-acting configurations.
- Single-stage compression: The entire compression process from initial inlet pressure (typically atmospheric) to final delivery pressure is completed in a single cylinder.
- Typical applications: Used in pneumatic tools, garage service stations, paint spraying, and small-scale industrial plants requiring pressures up to 10 bar.

Constructional Details I

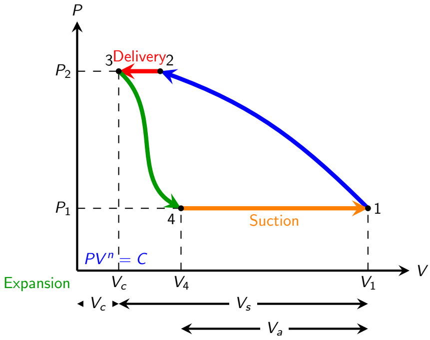
The reciprocating compressor consists of a cylinder, piston, piston rings, connecting rod, crankshaft, and spring-loaded inlet and delivery valves.

Constructional Details II

Key Points

- Cylinder and Piston: The cylinder is provided with water jackets or fins for cooling during compression. Piston rings prevent leakage between piston and cylinder wall.
- Suction and Delivery Valves: Automatic spring-loaded reed or plate valves that operate solely due to the pressure difference across them.
- Connecting Rod and Crank: Converts the rotary motion of the electric motor or engine crankshaft into the reciprocating motion of the piston.
- Clearance Volume: A small space left at the top of the cylinder (typically 3-10% of swept volume) to prevent the piston from striking the cylinder head and to allow valve operation.

PV Diagram (With Clearance) I



PV Diagram (With Clearance) II

Key Points

- State 4 to 1 (Suction Stroke): Piston moves from TDC (Inner Dead Center) to BDC (Outer Dead Center). Intake valve opens at 4 and fresh air is sucked at constant pressure P_1 until State 1 is reached.
- State 1 to 2 (Compression Stroke): Both valves remain closed. Piston moves from BDC towards TDC, reducing the volume and raising pressure from P_1 to P_2 polytropically ($PV^n = C$).
- State 2 to 3 (Delivery Stroke): Delivery valve opens when cylinder pressure exceeds receiver pressure P_2. Compressed air is discharged at constant pressure P_2 until piston reaches TDC at state 3.
- State 3 to 4 (Clearance Expansion): At TDC, some compressed air remains in the clearance volume (V_c). As piston moves back, this air expands polytropically to P_1 before new air can enter.

Step-by-Step Thermodynamic Operations I

The working of a reciprocating compressor operates in a cyclic manner composed of four distinct processes: suction, compression, delivery, and clearance expansion.

Step-by-Step Thermodynamic Operations II

Key Points

- Suction: Initiated as the piston moves down, creating a partial vacuum. The atmospheric pressure exceeds cylinder pressure, forcing the intake valve to open and fill the cylinder.
- Compression: As the piston reverses direction, the intake valve snaps shut due to spring force and rising internal pressure. The air is compressed, increasing its temperature and density.
- Discharge: The delivery valve opens against receiver pressure. The high-pressure air is pushed out. Note that the piston never reaches the cylinder head to prevent mechanical crash.
- Valve Action: Automatic valve opening and closing are governed entirely by differential pressure across the valve plates, controlled with calibrated spring stiffness.

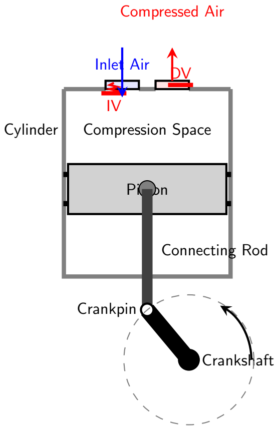
Volumetric Efficiency Analysis I

Volumetric efficiency (η_{vol}) is the ratio of the actual volume of free air sucked at intake conditions to the swept volume of the cylinder.

Key Points

- Mathematical Expression: $\eta_{vol} = \frac{V_a}{V_s} = 1 - C[(\frac{P_2}{P_1})^{\frac{1}{n}} - 1]$, where $C = \frac{V_c}{V_s}$ is the clearance ratio.
- Effect of Clearance: Increasing clearance volume V_c decreases the volumetric efficiency because more space is occupied by re-expanding clearance air.
- Effect of Pressure Ratio: A higher pressure ratio (P_2/P_1) decreases volumetric efficiency drastically since the clearance air expands to a much larger volume (V_4).
- Practical Bounds: Typically, reciprocating compressor volumetric efficiencies range from 70% to 85% in well-designed single-stage units.

Schematic of Reciprocating Compressor I



Schematic of Reciprocating Compressor II

Key Points

- Inlet Valve (IV): Opens inwards when cylinder pressure drops below inlet pressure. Allows fresh air to enter cylinder.
- Delivery Valve (DV): Opens outwards when cylinder pressure exceeds receiver pressure. Discharges compressed air.
- Crank Mechanism: Crank and connecting rod assembly converts rotary drive into linear reciprocating stroke of the piston.
- Piston Rings: Placed in piston grooves to prevent leakage of high-pressure air into the crankcase (blow-by).

Work Done and Power Calculations I

The work done per cycle can be calculated from the area of the indicator diagram. This slide details the work done with clearance volume.

Work Done and Power Calculations II

Key Points

- Polytropic Work formula: $W = \frac{n}{n-1} P_1 (V_1 - V_4)[(\frac{P_2}{P_1})^{\frac{n-1}{n}} - 1]$, where $V_1 - V_4 = V_a$ is the effective swept volume.
- Indicated Power (IP): $IP = \frac{W \cdot N_w}{60}$ Watts, where N_w is the number of working strokes per minute ($N_w = N$ for single-acting, $N_w = 2N$ for double-acting).
- Isothermal Efficiency: The ratio of isothermal work done (ideal scenario) to the actual indicated work. Maximizing cooling improves isothermal efficiency.
- Brake Power (BP): The actual power input to the compressor crankshaft, related to Indicated Power by mechanical efficiency ($BP = IP/\eta_m$).

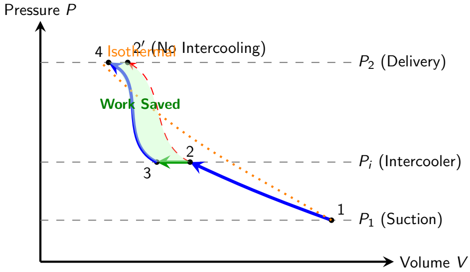
Thermodynamic and Mechanical Limitations of Single-Stage Systems

Need for Multi-Stage Compression II

Key Points

- Volumetric Efficiency Decay: In single-stage systems, high pressure ratios cause clearance volume air to expand significantly during suction, occupying the cylinder volume and dropping volumetric efficiency.
- Thermal Stress and Lubrication Breakdown: Compressing air to high pressure in a single stage raises delivery temperatures past 150°C, which degrades cylinder lubricants, increases wear, and poses combustion risks.
- Excessive Work Input: High pressure ratios cause the polytropic compression curve to deviate drastically from the ideal isothermal curve, requiring excessive mechanical power input.
- Mechanical Unbalance: Higher peak pressures demand heavy structural components (cylinders, pistons, connecting rods) to handle the torque fluctuations, raising capital cost.

Two-Stage P-V Diagram Analysis I



Two-Stage P-V Diagram Analysis II

Key Points

- Indicated Work Comparison: The P-V cycle displays the Low-Pressure (LP) stage (1-2) and the High-Pressure (HP) stage (3-4).
- Intercooling Action: Constant pressure cooling (2-3) reduces the volume of air from V_2 to V_3 before it enters the HP cylinder.
- Work Saved Area: The shaded region bounded by 2-2'-4-3 represents the compression work saved due to the intercooling process.
- Ideal Target: Isothermal compression represents the absolute theoretical minimum work requirement, acting as a boundary reference.

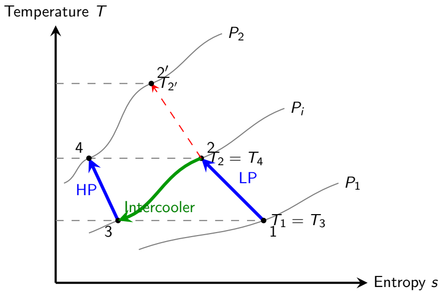
Thermodynamics of Intercooling I

Perfect vs. Imperfect Intercooling Processes

Key Points

- Perfect (Complete) Intercooling: Heat rejection in the intercooler returns the air temperature to its initial suction state ($T_3 = T_1$), placing point 3 on the main isothermal line.
- Imperfect (Incomplete) Intercooling: Occurs when the air is cooled to a temperature higher than suction inlet temperature ($T_3 > T_1$), reducing the overall work-saving benefit.
- Specific Volume reduction: Cooling increases air density ($P = \rho RT$), meaning a smaller cylinder swept volume is required for the HP stage to handle the same mass flow rate.
- Heat Transferred: The heat removed during intercooling is calculated as $Q = mC_p(T_2 - T_3)$, requiring steady external cooling water flow.

Two-Stage T-s Diagram Analysis I



Two-Stage T-s Diagram Analysis II

Key Points

- Temperature Profile: Shows how intercooling controls temperature rise, keeping discharge temperatures T_2 and T_4 well below single-stage level $T_{2'}$.
- Isobaric Paths: Heat rejection in the intercooler follows the intermediate pressure curve P_i from state 2 to state 3.
- Equal Temperature Rise: Under optimum conditions, the temperature increases in LP and HP stages are identical ($T_2 - T_1 = T_4 - T_3$).
- Polytropic Slopes: The compression lines incline to the left if heat is rejected to the cylinder jacket ($1 < n < \gamma$).

Optimization and Mathematical Relations I

Work Formula and Condition for Minimum Total Power Input

Key Points

- Total Indicated Work: For a 2-stage system with complete intercooling: $W = \frac{n}{n-1} mRT_1[(\frac{P_i}{P_1})^{\frac{n-1}{n}} + (\frac{P_2}{P_i})^{\frac{n-1}{n}} - 2]$.
- Optimum Intermediate Pressure: Differentiating W with respect to P_i gives the optimum condition: $P_i = \sqrt{P_1 \cdot P_2}$.
- Equal Stage Pressure Ratios: Under optimum pressure, the pressure ratio is identical in each stage: $r_p = \frac{P_i}{P_1} = \frac{P_2}{P_i} = \sqrt{\frac{P_2}{P_1}}$.
- Equipartition of Work: The optimum condition splits the total compression work equally between the LP and HP cylinders ($W_{LP} = W_{HP}$).
- Minimum Work Equation: The optimized work is given by $W_{min} = \frac{2n}{n-1} mRT_1[(\frac{P_2}{P_1})^{\frac{n-1}{2n}} - 1]$.

Performance of Reciprocating Compressor I

Key Efficiency and Operational Metrics (No Derivation)

Key Points

- Volumetric Efficiency (Clearance Effect): $\eta_{vol} = 1 + C - C(\frac{P_2}{P_1})^{\frac{1}{n}}$, where $C = \frac{V_c}{V_s}$ is the clearance ratio.
- Isothermal Efficiency: $\eta_{iso} = \frac{\text{Isothermal Work}}{\text{Actual Indicated Work}} = \frac{mRT_1 \ln(P_2/P_1)}{W_{\text{indicated}}}$. This is the benchmark for reciprocating compressors.
- Isentropic/Adiabatic Efficiency:

$$\eta_{ad} = \frac{\text{Isentropic Work}}{\text{Actual Indicated Work}} = \frac{\frac{\gamma}{\gamma-1} mRT_1[(\frac{P_2}{P_1})^{\frac{\gamma-1}{\gamma}} - 1]}{W_{\text{indicated}}}.$$

- Mechanical Efficiency: $\eta_{mech} = \frac{\text{Indicated Power (IP)}}{\text{Brake Power (BP)}}$. Accounts for friction losses in bearings and cylinder walls.

Industrial Applications and Design Considerations I

Practical Configurations and Operating Guidelines

Key Points

- Multi-stage Pressure Boundaries: Typically, single-stage is used up to 5 bar, two-stage up to 35 bar, three-stage up to 85 bar, and four-stage for higher pressures.
- Moisture Separator Requirement: Cooling in the intercooler condenses moisture out of the air. Condensate separators are mandatory at the exit of the intercooler to avoid water entry into the HP cylinder.
- Intercooler Construction: Industrial systems use water-jacketed shell-and-tube heat exchangers with counter-flow configuration for maximum heat transfer effectiveness.
- Safety Systems: Pressure relief valves (PRV) must be installed on the intercooler to prevent pressure build-up in case of HP suction valve failure.

Work Required for Single-Stage Compressor (Without Clearance) I

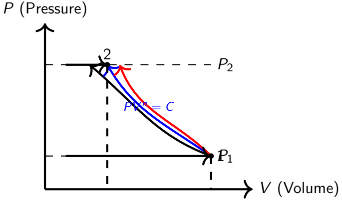
The work required per cycle by a single-stage, single-acting reciprocating compressor without clearance varies depending on the thermodynamic path of compression. The expressions represent work input per cycle.

Work Required for Single-Stage Compressor (Without Clearance) II

Key Points

- Isothermal compression (ideal case, n=1): W = p1 * V1 * ln(p2/p1). This requires the minimum possible work input.
- Polytropic compression (real-world process, 1 ¡ n ¡ gamma): W = (n / (n - 1)) * p1 * V1 * [(p2 / p1)ˆ((n - 1) / n) - 1].
- Isentropic/Adiabatic compression (no heat transfer, n = gamma): W = (gamma / (gamma - 1)) * p1 * V1 * [(p2 / p1)ˆ((gamma - 1) / gamma) - 1].
- Definitions: p1 & V1 are suction conditions, p2 is delivery pressure, and n is the polytropic index (typically 1.2 to 1.35).

P-V Diagram of Air Compressor Cycle I



P-V Diagram of Air Compressor Cycle II

Key Points

- Process 4-1: Induction/suction stroke of air at constant suction pressure p1.
- Process 1-2: Compression stroke from initial pressure p1 to delivery pressure p2.
- Process 2-3: Delivery/discharge stroke at constant delivery pressure p2.
- The area enclosed by 1-2-3-4 represents the net work input required. Isothermal compression requires the smallest area (least work).

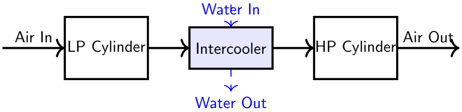
Efficiencies of Reciprocating Compressors I

Efficiencies describe the deviation of actual compressor performance from ideal thermodynamic baselines and mechanical constraints.

Key Points

- Isothermal Efficiency: The ratio of ideal isothermal work to the actual indicated work done. eta_iso = W_isothermal / W_indicated.
- Volumetric Efficiency: The ratio of the actual volume of free air delivered (FAD) at suction conditions to the swept volume of the cylinder. eta_vol = V_actual / V_swept.
- Mechanical Efficiency: The ratio of indicated power (IP) inside the cylinder to the actual brake power (BP) input at the crankshaft. eta_mech = IP / BP.
- Overall Isothermal Efficiency: The ratio of isothermal power to brake power input. eta_overall_iso = eta_iso * eta_mech.

Multistage Compression & Intercooling Schematic I



Key Points

- In multi-stage systems, air is compressed to an intermediate pressure (pi) in the Low Pressure (LP) cylinder, cooled in an intercooler, and then compressed to delivery pressure (p2) in the High Pressure (HP) cylinder.
- Perfect intercooling: The air is cooled back to the initial suction temperature (T3 = T1) before it enters the HP cylinder.
- For minimum total compression work in a 2-stage compressor, the intermediate pressure must be the geometric mean: pi = sqrt(p1 * p2).
- Multistage compression reduces high delivery temperatures, improves volumetric efficiency, and lowers mechanical load on components.

Key Equations for Multi-Stage Compression I

Engineers use standard thermodynamic formulas to compute the work required for multi-stage reciprocating air compressors under optimal conditions.

Key Points

- For N-stage compression with perfect intercooling and equal pressure ratio per stage, total work W = (N * n / (n - 1)) * p1 * V1 * [(p_N+1 / p1)^((n - 1) / (N * n)) - 1].
- For a 2-stage compressor (N=2) with perfect intercooling: W = (2 * n / (n - 1)) * p1 * V1 * [(p2 / p1)^((n - 1) / (2 * n)) - 1].
- If intercooling is imperfect (T3 ¿ T1), total work is computed by summing individual cylinders: W = W_LP + W_HP.
- Condition for minimum work: Work done in the LP cylinder equals work done in the HP cylinder (W_LP = W_HP).

Numerical Example 1: Indicated Work Calculation I

Problem Statement: A single-acting, single-stage reciprocating compressor intakes air at 1 bar and 300 K, compressing it to 6 bar. The compression process is polytropic with n $= 1.3$. Calculate the work required per kg of air. (Take R $= 0.287$ kJ/kg-K).

Numerical Example 1: Indicated Work Calculation II

Key Points

- Given data: p1 = 1 bar, p2 = 6 bar, T1 = 300 K, n = 1.3, R = 0.287 kJ/kg-K.
- Work equation per kg (m = 1 kg): W = (n / (n - 1)) * R * T1 * [(p2 / p1)$\hat{}$((n - 1) / n) - 1].
- Step 1: Compute pressure ratio: p2/p1 = 6 / 1 = 6.
- Step 2: Substitute values: W = (1.3 / (1.3 - 1)) * 0.287 * 300 * [6$\hat{}$((1.3 - 1) / 1.3) - 1].
- Step 3: W = 4.333 * 86.1 * [1.512 - 1] = 373.1 * 0.512 = 191.0 kJ/kg.
- Conclusion: The indicated work required to compress 1 kg of air is approximately 191 kJ.

Numerical Example 2: Efficiency Evaluation I

Problem Statement: For the compressor from Example 1, calculate the isothermal efficiency and mechanical efficiency if the actual shaft work input (Brake Work) is 240 kJ/kg of air. The indicated work is 191 kJ/kg.

Numerical Example 2: Efficiency Evaluation II

Key Points

- Given data: W_indicated = 191 kJ/kg, W_brake = 240 kJ/kg, suction temperature T1 = 300 K, pressure ratio = 6.
- Step 1: Calculate isothermal work: W_iso = R * T1 * ln(p2/p1) = 0.287 * 300 * ln(6) = 86.1 * 1.7918 = 154.3 kJ/kg.
- Step 2: Calculate Isothermal Efficiency: eta_iso = W_iso / W_indicated = 154.3 / 191 = 80.8%.
- Step 3: Calculate Mechanical Efficiency: eta_mech = W_indicated / W_brake = 191 / 240 = 79.6%.
- Conclusion: The system operates at an isothermal efficiency of 80.8% and a mechanical efficiency of 79.6%.

Classification and Overview of Rotary Compressors I

Rotary compressors deliver a continuous flow of high-pressure air through rotational mechanisms rather than reciprocating piston strokes.

Classification and Overview of Rotary Compressors II

Key Points

- Positive Displacement Rotary Compressors: Entrap a fixed volume of air and force it into a smaller space (e.g., Roots blower, screw, vane).
- Dynamic Rotary Compressors: Accelerate the air velocity and convert kinetic energy to static pressure (e.g., centrifugal, axial flow).
- Operating Pressures: Typically lower discharge pressures compared to reciprocating types, but handle significantly higher mass flow rates.
- Mechanical Design: No inlet/outlet valves required; fewer moving parts result in lower maintenance and vibration-free installation.

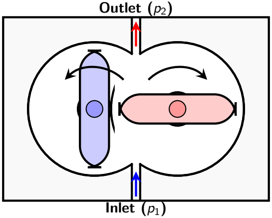
Roots Blower: Constructional Details I

The Roots blower is a rotary positive displacement compressor featuring twin-lobed rotors running inside a close-fitting casing.

Key Points

- Lobed Rotors: Two symmetrical, epitrochoidal or cycloidal lobed rotors (typically 2-lobe or 3-lobe designs) rotating in opposite directions.
- Timing Gears: Key components keyed to rotor shafts that maintain strict synchronization, preventing contact between the lobes.
- Casing Clearances: Tiny clearances (0.1 mm to 0.2 mm) between casing and lobes, and between lobes themselves, act as a labyrinth seal.
- Lubrication & Cleanliness: Lubrication is restricted to external timing gears and bearings; the compression chamber remains oil-free.

Roots Blower Cross-Section Schematic I



Roots Blower Cross-Section Schematic II

Key Points

- Visual representation of the two-lobe rotary configuration showing vertical and horizontal mating positions.
- Inlet Air (p_1) enters from the bottom port and fills the voids created by the moving lobes.
- Lobes rotate in opposite directions (Left: counter-clockwise, Right: clockwise) to carry air packets around the casing periphery.
- Outlet Air (p_2) is forced out through the top port into the receiver.

Roots Blower: Step-by-Step Working Principle I

The compression in a Roots blower is unique because the air is not compressed inside the casing itself; compression occurs at the delivery port due to backflow.

Key Points

- Trapping: As lobes turn, air from the intake is trapped in the pocket volume V_s between the lobe and the casing wall.
- Periphery Transport: This pocket of air is carried around the casing at constant suction pressure p_1 without any volume changes.
- Backflow Compression: When the lobe tip opens to the delivery port, high-pressure air from the receiver (p_2) rushes back into the pocket, compressing the air instantaneously.
- Delivery: The rotation of the lobes then pushes the compressed mixture of air out into the receiver against pressure p_2.

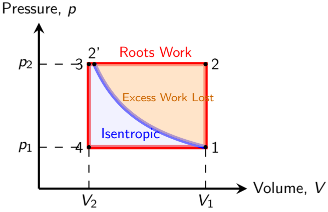
Thermodynamic Analysis & Work Calculations I

Unlike reciprocating compressors that compress air isentropically, the Roots blower uses constant-volume pressure-raise mechanism.

Key Points

- Work Done per Revolution: For a two-lobe blower, four pockets of air are delivered per revolution. Total volume per revolution $V_{rev} = 4V_s$.
- Roots Work Equation: $W_{Roots} = (p_2 - p_1) \cdot V_{rev} = 4V_s(p_2 - p_1)$ Joules/rev.
- Isentropic Comparison: The work for an ideal reciprocating compressor is $W_{isen} = \frac{\gamma}{\gamma-1} p_1 V_{rev}[(\frac{p_2}{p_1})^{\frac{\gamma-1}{\gamma}} - 1]$.
- Power Requirement: Total Power $P = \frac{4V_s(p_2-p_1)N}{60}$ Watts, where N is the rotational speed in RPM.

p-V Diagram & Roots Efficiency I



p-V Diagram & Roots Efficiency II

Key Points

- Comparison of the rectangular Roots Blower indicator diagram (1-2-3-4) and the isentropic compression path (1-2').
- The shaded orange area represents the excess work lost due to irreversible backflow compression.
- Roots Efficiency: $\eta_{Roots} = \frac{W_{isen}}{W_{Roots}} = \frac{\gamma}{\gamma-1} \cdot \frac{p_1[r_p^{(\gamma-1)/\gamma}-1]}{p_2-p_1}$, where $r_p = p_2/p_1$.
- At higher pressure ratios ($r_p > 2$), the excess work lost increases rapidly, making the Roots blower highly inefficient.

Performance Characteristics and Applications I

The Roots blower is a robust and reliable machine suitable for low-pressure, high-volume industrial applications.

Key Points

- Volumetric Efficiency: Generally ranges from 80% to 90%. Decreases as the pressure ratio rises due to high back-leakage through clearance spaces.
- Slip Factor: Accounted for by leakage; slip increases with higher pressure ratios, reducing actual delivered flow rate.
- Advantages: Robust construction, no sliding contact (meaning no wear and tear in compression chamber), and yields fully oil-free air.
- Industrial Applications: Sewage aeration, pneumatic transport of granular material, engine supercharging, vacuum pumping, and metallurgy.

Rotary Vane Compressor: Working Principle I

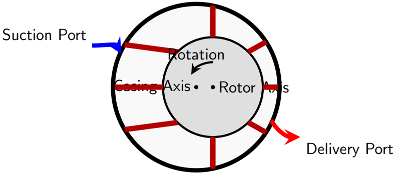
A rotary vane compressor consists of a rotor placed eccentrically inside a cylindrical stator casing. The rotor has longitudinal slots housing radial vanes that are pressed against the casing wall by centrifugal force or spring action.

Rotary Vane Compressor: Working Principle II

Key Points

- As the rotor rotates, the volume between adjacent vanes (vane cells) changes due to the eccentricity between the rotor and stator axes.
- Air is drawn in through the intake port where the cell volume increases, trapped as the cell is sealed, and compressed as the cell volume decreases towards the discharge port.
- Oil injection is commonly used for sealing the clearance gaps, lubricating the sliding vanes, and cooling the air during compression (approximating isothermal compression).
- Typical operating pressures range from 7 to 10 bar, with rotational speeds between 1000 and 3000 RPM.

Rotary Vane Compressor Schematic I



Rotary Vane Compressor Schematic II

Key Points

- Visualizes the stator casing, eccentric rotor, and the radial sliding vanes.
- Clearly shows the expansion of vane cell volume at the suction side and contraction at the delivery side.
- The eccentricity 'e' is the radial distance between the casing axis and the rotor axis.
- Sealing between the vane tips and the casing is crucial to prevent leakage across adjacent cells.

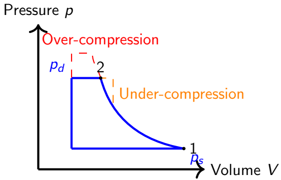
The performance of a vane compressor is characterized by its volumetric efficiency, pressure ratio, and shaft power. The theoretical displacement volume per revolution can be calculated using the rotor and stator geometry.

Thermodynamic and Kinematic Analysis II

Key Points

- Displacement volume: V_d = 2 * pi * b * e * (D_s - e - t_v * n / pi) where b is rotor width, e is eccentricity (R_s - R_r), D_s is stator diameter, t_v is vane thickness, and n is the number of vanes.
- Internal compression ratio is determined by the ratio of the volume of the cell at the moment of suction port closure to its volume at the moment of discharge port opening.
- Under-compression occurs when the discharge port opens too late (internal pressure exceeds receiver pressure), while over-compression occurs when it opens too early, leading to backflow and throttling losses.
- Isentropic efficiency typically ranges from 70% to 85%, heavily influenced by blade-tip friction and clearance-leakage losses.

Indicator p-V Diagram: Ideal vs. Real Compression I



Indicator p-V Diagram: Ideal vs. Real Compression II

Key Points

- Thermodynamic cycle showing the effects of design pressure ratio mismatch.
- Area enclosed by the curves represents the work done per compression cycle.
- Over-compression indicates that the internal pressure exceeded the receiver pressure, resulting in wasted shaft work.
- Under-compression causes a backflow of high-pressure air from the receiver into the cell when the port opens, increasing work required.

Rotary Screw Compressor: Working Principle I

Rotary screw compressors belong to the positive displacement category, consisting of two intermeshing helical rotors (male and female) mounted inside a close-fitting casing. The male rotor typically has 4 lobes and drives the female rotor which has 5 or 6 flutes.

Rotary Screw Compressor: Working Principle II

Key Points

- As the rotors mesh, they create a series of chambers or pockets that move axially from the suction side to the discharge side.
- Air enters the inlet port and is trapped within the helical grooves. As the rotors continue to rotate and intermesh, the pocket volume decreases, compressing the trapped air.
- The compression ratio is built-in based on the rotor geometry and port locations, rather than being determined by the system pressure.
- Oil-flooded screw compressors use oil for sealing, lubrication, and cooling (absorbing heat of compression), while oil-free types use external timing gears to prevent rotor contact.

Key Parameters and Control Strategies in Screw Compressors I

Operational performance is highly dependent on rotation speeds, rotor profile design (e.g., SRM, Sigma profiles), and capacity modulation.

Key Parameters and Control Strategies in Screw Compressors II

Key Points

- Built-in Pressure Ratio (v_i): defined as the ratio of the volume of the trapped air pocket at the start of compression to the volume at the moment the discharge port is uncovered.
- Slide Valve Control: A slide valve located in the casing allows axial movement to bypass a portion of the suction air back to the inlet, enabling stepless capacity control from 10% to 100% load.
- Rotational Speed: Screw compressors can operate at high speeds (up to 6000 RPM or more), allowing very high volumetric flow rates relative to their physical footprint.
- Thermodynamic Efficiencies: Isentropic efficiencies can exceed 85% in large, well-designed oil-flooded industrial units operating near their design pressure ratio.

Comparative Performance and Industrial Applications I

Both vane and screw compressors are popular rotary positive displacement machines, but they serve different operational envelopes and maintenance profiles.

Comparative Performance and Industrial Applications II

Key Points

- Flow Capacity: Screw compressors are suitable for high flow rates (up to 1000 $m\hat{3}$/min), whereas vane compressors are limited to lower to medium flows (typically up to 50 $m\hat{3}$/min).
- Durability and Maintenance: Vane compressors suffer from vane wear due to continuous sliding contact with the casing wall, requiring regular replacement, whereas screw rotors do not contact each other directly.
- Part-Load Efficiency: Screw compressors utilizing slide valves or variable speed drives (VSD) offer excellent part-load efficiency compared to vane compressors.
- Noise and Vibration: Screw compressors produce high-frequency noise that requires silencers, while vane compressors run relatively quietly at lower speeds.

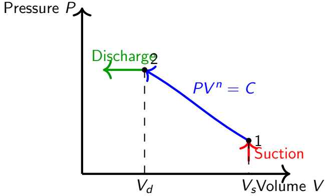
Working Principle of Scroll Compressors I

A scroll compressor uses two co-wound spiral scrolls to ingest, compress, and discharge working fluids.

Key Points

- One scroll is stationary (fixed scroll) while the second orbits eccentrically (orbiting scroll) without rotating around its own axis.
- Suction occurs at the outer periphery where gas is trapped in crescent-shaped pockets formed by the scroll geometries.
- As the orbiting scroll moves, the pocket volume decreases continuously toward the center, raising the gas pressure.
- Discharge takes place via a central port once the minimum volume (maximum pressure) is reached.
- Continuous flow with minimal pulsation, high volumetric efficiency, and fewer moving parts than reciprocating compressors.

P-V Diagram of a Scroll Compressor I



P-V Diagram of a Scroll Compressor II

Key Points

- Suction ends at state 1 when the outer pockets close off at volume V_s and suction pressure P_s.
- Compression phase: The pocket volume decreases continuously according to a polytropic path PVn̂ = C, typically with n ≈1.1 to 1.3.
- Discharge begins at state 2 when the pocket links to the central discharge port at pressure P_d.
- No clearance volume re-expansion losses since there are no suction/discharge valves dynamically cycling in the compression chamber.
- This yields high volumetric efficiency (typically ¿90-95%).

Construction & Mechanical Details I

Key components and geometric parameters of a scroll compressor.

Key Points

- Fixed Scroll: Bolted directly to the compressor housing and contains the central discharge port.
- Orbiting Scroll: Mounted on an eccentric crank mechanism that drives it in an orbital path.
- Oldham Coupling: A key mechanical link that prevents rotation of the orbiting scroll while permitting its orbital translational motion.
- Tip Seals: Located at the tips of the scroll wraps to prevent axial leakage between adjacent pockets.
- Radial and Axial Compliance: Mechanisms allowing slight movement of scrolls to separate under liquid slugging conditions, enhancing reliability.

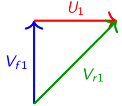
2.4 Dynamic Air Compressors: Introduction I

Dynamic compressors work by transferring kinetic energy to the gas stream, which is then converted into pressure energy.

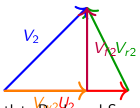
Key Points

- Unlike positive displacement compressors, dynamic compressors operate at constant pressure for a given speed.
- Continuous flow rate is much higher compared to reciprocating or rotary positive displacement types.
- The two primary classes of dynamic compressors are Centrifugal (radial flow) and Axial-flow compressors.
- Energy transfer is governed by Euler's Turbomachinery Equation: W_c = u_2 * V_w2 - u_1 * V_w1.
- Typically utilized in large-scale industrial applications, gas turbines, and chemical processing plants.

Centrifugal Compressor Velocity Triangles I



Inlet: Radial Entry ($V_{w1} = 0$)



Outlet: Backward-Swept

Centrifugal Compressor Velocity Triangles II

Key Points

- At the impeller inlet, air enters axially. If there are no inlet guide vanes, the inlet absolute velocity V_1 is purely axial, meaning velocity of whirl $V_{w1} = 0$.
- The blade velocity U_1 and relative velocity V_{r1} form the inlet velocity triangle.
- At the impeller outlet, air leaves with absolute velocity V_2 at angle α_2, containing both radial flow component V_{f2} and tangential whirl component V_{w2}.
- Work input per unit mass is given by: $w = U_2 * V_{w2}$ (since $V_{w1} = 0$).
- Diffuser channels placed downstream of the impeller convert the high kinetic energy (V_2) of the leaving air into static pressure.

Centrifugal Compressor: Construction & Flow Path I

Detailed overview of the component configuration and thermodynamic changes through the centrifugal compressor.

Key Points

- Casing: Aerodynamically shaped housing to guide the flow and minimize pressure drops.
- Impeller: Rotating component with curved blades that accelerates the air radially outward.
- Diffuser: Stationary ring of passages (vaned or vaneless) surrounding the impeller, converting kinetic head into static pressure.
- Volute (Scroll): Gradual increasing area collector casing that leads the compressed air to the discharge pipe.
- Operating limit phenomena include Surging (complete flow reversal/instability) and Choking (sonic velocity limit at throat).

Comparison: Positive Displacement vs. Dynamic Compressors I

Comparison criteria for selection in industrial thermal systems.

Key Points

- Pressure Ratio: Scroll and other positive displacement types handle high pressure ratios per stage; Dynamic types have lower pressure ratio per stage but support multi-staging.
- Volume Flow Rate: Dynamic compressors handle extremely large volumetric flows (axial ¿ centrifugal), whereas scroll compressors are for low-to-medium flow rates.
- Efficiency characteristics: Positive displacement efficiency is stable across varying pressures, whereas dynamic compressor efficiency drops significantly away from design points.

Comparison: Positive Displacement vs. Dynamic Compressors (Cont.) I

Key Points

- Flow pulsations: Scroll compressors provide smooth continuous flow compared to reciprocating types, but dynamic compressors offer completely pulsation-free delivery.
- Primary application sectors: Scroll for HVAC and heat pumps; Centrifugal for industrial process air, turbochargers, and large-scale gas transport.

Dynamic Compressors: Fundamental Classification I

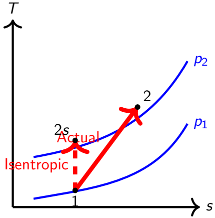
Dynamic compressors continuously impart velocity to the air flowing through them via rotating impellers or blades, subsequently converting this kinetic energy into static pressure in diffusers or stator passages.

Dynamic Compressors: Fundamental Classification II

Key Points

- Unlike positive displacement compressors, dynamic machines operate at constant pressure ratios for a given speed but with variable mass flow rate.
- Centrifugal Compressors: Air enters axially, turns 90 degrees to flow radially outward, resulting in high pressure rise per stage (typically 3.0 to 4.5:1) but lower volumetric capacity.
- Axial Compressors: Air flows parallel to the rotational axis, yielding high mass flow rates and efficiency, but requiring multiple stages due to low pressure rise per stage (1.15 to 1.4:1).
- Industrial Applications: Centrifugal systems are utilized in turbochargers, small gas turbines, and process industries, while axial systems dominate aviation gas turbines and large power plants.

Thermodynamic Analysis: T-s Diagram of Centrifugal Compression I



Thermodynamic Analysis: T-s Diagram of Centrifugal Compression II

Key Points

- The T-s diagram represents the thermodynamic states of compression between inlet pressure p1 and discharge pressure p2.
- State 1 to 2s represents the ideal, isentropic compression process without internal losses, where entropy remains constant.
- State 1 to 2 illustrates the actual irreversible compression process where entropy increases due to friction, turbulence, and shock losses.
- Isentropic Efficiency is defined as the ratio of isentropic work to actual work input: η_isen = (T_2s - T_1) / (T_2 - T_1), which is typically in the range of 75% to 85%.

Centrifugal Compressor Velocity Triangles at Impeller Outlet I

Understanding velocity triangles is critical for calculating the work done on the fluid. Let U2 be the impeller tip speed, V2 the absolute velocity of the leaving air, Vr2 the relative velocity, Vw2 the whirl (tangential) component, and Vf2 the radial flow velocity.

Centrifugal Compressor Velocity Triangles at Impeller Outlet II

Key Points

- Euler Work Equation: The theoretical work input per unit mass flow rate is given by W = U2 * Vw2 - U1 * Vw1. Since inlet flow is usually axial without pre-whirl (Vw1 = 0), the equation simplifies to W = U2 * Vw2.
- Slip Factor (μ): Due to inertia, the fluid does not perfectly follow the blades. The actual whirl velocity Vw2' is less than the theoretical Vw2. Slip factor is defined as μ= Vw2' / Vw2.
- Power Input Factor (ψ): To account for windage, disk friction, and recirculation losses, a power input factor is applied: W_actual = ψ* μ* U2$\hat{2}$.
- Typical blade profiles at outlet are backward-curved (highest efficiency, stable operating range), radial (easy manufacturing, high pressure rise), and forward-curved (rarely used due to instability).

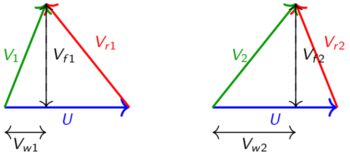
Axial Flow Compressors: Operating Principle I

An axial flow compressor consists of alternating rows of rotating blades (rotors) and stationary blades (stators) arranged in a drum configuration. Fluid is accelerated in the rotor and decelerated in the stator, raising its pressure.

Key Points

- Each stage consists of one rotor row followed by one stator row. The flow remains primarily axial with negligible change in radius.
- The rotor imparts kinetic energy to the fluid while increasing the relative velocity and static pressure due to diverging blade profiles.
- The stator converts the kinetic energy of the fluid leaving the rotor into static pressure and redirects the flow at the correct angle for the next stage.
- Degree of Reaction (R): The ratio of static enthalpy rise in the rotor to the static enthalpy rise in the entire stage. A symmetric stage design typically has a degree of reaction $R = 0.5$.

Velocity Triangles for an Axial Compressor Stage I



Velocity Triangles for an Axial Compressor Stage II

Key Points

- Vector addition shows the transition of fluid speed: Absolute velocity V1 is vector sum of blade speed U and relative velocity Vr1.
- In the rotor, absolute velocity increases from V1 to V2, while relative velocity decreases from Vr1 to Vr2 due to diffuser-like passages.
- The constant flow velocity assumption (Vf1 = Vf2 = Vf) simplifies mass balance and work calculations across a constant annular area.
- Stage Work input per unit mass flow: W = U * (Vw2 - Vw1) = U * Vf * (tanα_2 - tanα_1), where αrepresents absolute flow angles.

Comparative Evaluation: Centrifugal vs. Axial Compressors I

Engineering decisions for selecting a compressor type depend on requirements for flow rate, pressure ratio, efficiency, frontal area, weight, and system complexity.

Comparative Evaluation: Centrifugal vs. Axial Compressors II

Key Points

- Mass Flow Rate: Axial compressors handle much larger mass flows (up to 100+ kg/s) compared to centrifugal units (usually ¡ 15 kg/s).
- Efficiency: Axial compressors achieve higher design efficiencies (85-90%) because flow is linear and undergoes fewer direction changes, minimizing losses.
- Pressure Ratio per Stage: Centrifugal stage yields high ratio (up to 4:1 or more), whereas axial stages produce low ratio (1.2:1), necessitating many stages (e.g., 10-15 stages).
- Operational Flexibility: Centrifugal compressors exhibit a wider operating range and are less prone to sudden instability under variable load conditions.

Compressor Instability: Surging, Choking, and Stalling I

Dynamic compressors are subject to aerodynamic limitations. Operability depends on avoiding flow instability regimes which can cause mechanical failure.

Compressor Instability: Surging, Choking, and Stalling II

Key Points

- Surging: Complete breakdown of flow through the compressor characterized by periodic reversal of flow. Occurs at low mass flow rates and causes high vibrations and thermal stress.
- Stalling: Localized aerodynamic separation of flow from blade surfaces. It can be steady or rotating (propagating around the blade row) and leads to decreased efficiency and blade fatigue.
- Choking: Occurs when mass flow reaches the maximum limit due to sonic velocity (Mach number $= 1.0$) at the throat section of any blade passage. Flow rate cannot increase further.
- Anti-Surge Controls: Implemented via bypass valves (recirculating air back to inlet) or variable stator vanes (VSVs) to maintain flow rates above the surge limit line.

Introduction to Steam Formation I

Water exists in solid, liquid, or vapor phases depending on temperature and pressure. Steam formation is the phase transformation process from liquid water to gaseous water (steam) at constant pressure, which involves sensible heating, latent heat absorption, and superheating.

Key Points

- Phase boundary: The saturation temperature (T_{sat}) is the temperature at which vaporization occurs for a given pressure.
- Saturation pressure (P_{sat}) is the corresponding pressure at which phase change happens at a given temperature.
- Undercooled/Compressed liquid: Water at a temperature lower than T_{sat} for the given pressure.
- Superheated vapor: Steam heated to a temperature higher than T_{sat} at a given pressure, increasing its thermal efficiency for power cycles.

Industrial Applications of Steam I

Steam is the most widely used working fluid and heating medium in industries due to its high specific heat capacity, high latent heat of vaporization (2257 kJ/kg at 1.013 bar), and ease of transport.

Industrial Applications of Steam II

Key Points

- Thermal Power Plants: High-pressure superheated steam (typically > 150 bar, $> 540\ ^{\circ}$C) drives steam turbines in the Rankine cycle to generate electricity.
- Process Industries: Steam is used for direct heating, sterilization, and distillation in chemical, pharmaceutical, paper, and food processing plants.
- Cogeneration Systems: Combined heat and power (CHP) systems use high-pressure steam for electricity and exhaust low-pressure steam for process heating, reaching thermal efficiencies over 80%.
- Mechanical Drives: Steam-driven turbines are used to run heavy industrial equipment like boiler feed pumps, fans, and compressors.

Constant Pressure Steam Formation Stages I

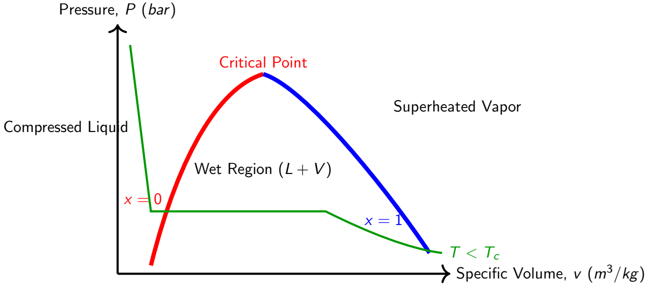
When 1 kg of subcooled water at 0 $^\circ$C is heated at a constant pressure P, it undergoes three distinct thermodynamic phases to transform into superheated steam.

Constant Pressure Steam Formation Stages II

Key Points

- Sensible Heating of Liquid: Water temperature rises from 0 °C to T_{sat}. The heat added is the enthalpy of saturated liquid ($h_f = C_{pw} \cdot T_{sat}$).
- Phase Change (Latent Heat Addition): At T_{sat}, water converts to steam at constant temperature. Enthalpy added is the latent heat of vaporization (h_{fg}).
- Dryness Fraction (x): Defines the quality of wet steam as $x = m_g/(m_f + m_g)$, where $x = 0$ is saturated liquid and $x = 1$ is dry saturated steam.
- Superheating Phase: Saturated steam ($x = 1$) is further heated to T_{sup}. The heat added is sensible heat of vapor ($h_{sup} - h_g = C_{ps}(T_{sup} - T_{sat})$).

Pressure-Volume (P-V) Diagram I

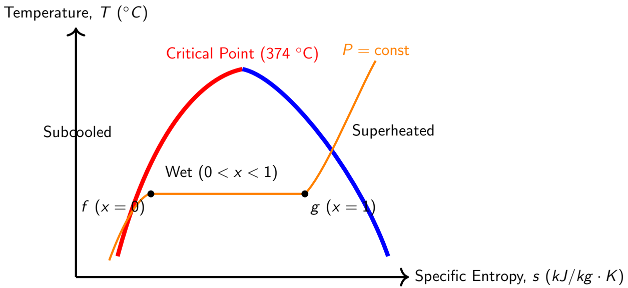


Pressure-Volume (P-V) Diagram II

Key Points

- Liquid and Vapor Dome: The region bounded by the saturated liquid line ($x = 0$) and the saturated vapor line ($x = 1$) represents the two-phase mixture.
- Critical Point: The peak of the dome (for water: $P_c = 221.2$ bar, $T_c = 374.15$ °C), where the phase transition from liquid to vapor occurs instantly without latent heat.
- Isothermal Lines: Isotherms slant downwards in the compressed liquid region, are horizontal in the wet region (constant pressure and temperature), and drop hyperbolically in the superheated region.
- State points: Saturated liquid states lie on the left side of the dome, while dry saturated steam states lie on the right side of the dome.

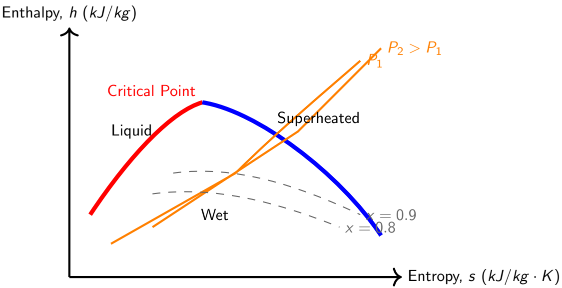
Temperature-Entropy (T-s) Diagram I



Temperature-Entropy (T-s) Diagram II

Key Points

- Entropy (s) change: Entropy increases during sensible heating and increases significantly during constant-temperature latent heat absorption ($s_{fg} = h_{fg}/T_{sat}$).
- Isobaric Lines: Constant pressure lines slope upwards in the liquid region, run horizontal under the dome, and slope upwards again in the superheated region.
- Slope of Liquid Line: Defined by $\frac{dT}{ds} = \frac{T}{C_{pw}}$, representing the temperature-entropy relationship for liquid water heating.
- Work and Heat: The area under the constant-pressure heating curve on the T-s diagram represents the total heat transferred ($Q = \int Tds$).


Enthalpy, h (kJ/kg)
$P_2 > P_1$
P_1
Critical Point
Liquid
Superheated
Wet
$x = 0.9$
$x = 0.8$
Entropy, s ($kJ/kg \cdot K$)

Enthalpy-Entropy (h-s) Diagram / Mollier Chart II

Key Points

- Mollier Chart Utility: Extremely useful for expansion (turbine) and throttling processes, where enthalpy changes are read directly off the vertical axis.
- Slope of Isobars: In the wet region, the slope of isobars is constant and equals the absolute saturation temperature, $(\frac{\partial h}{\partial s})_P = T_{sat}$.
- Diverging Isobars: Isobars diverge in the superheated region, meaning lines of constant pressure spread out at higher enthalpy and entropy values.
- Throttling Process: Represented as a horizontal line (isenthalpic, $h =$ constant), showing how throttling dry or wet steam reduces pressure and increases dryness.

Thermodynamic Property Calculations I

Equations used to calculate steam properties at different states. Subscripts 'f', 'g', and 'fg' represent saturated liquid, dry saturated vapor, and phase change difference respectively.

Key Points

- Wet Steam Dryness Fraction: $x = \frac{m_g}{m_f + m_g}$. Volume: $v_x = (1-x)v_f + xv_g \approx xv_g$ since $v_g \gg v_f$.
- Wet Steam Enthalpy and Entropy: $h_x = h_f + x \cdot h_{fg}$ and $s_x = s_f + x \cdot s_{fg}$.
- Superheated Steam Enthalpy: $h_{sup} = h_g + C_{ps}(T_{sup} - T_{sat})$ where C_{ps} is the specific heat of superheated steam (typically 2.1 to 2.3 kJ/kg K).
- Superheated Steam Entropy: $s_{sup} = s_g + C_{ps} \ln(\frac{T_{sup}}{T_{sat}})$ with absolute temperatures in Kelvin.

Thermodynamic States of Steam I

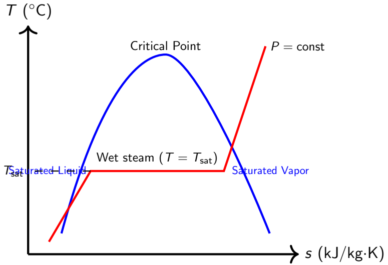
Steam exists in three distinct thermodynamic states: wet steam, dry saturated steam, and superheated steam. Calculating its properties requires identifying the phase state using dryness fraction (x), saturation pressure (P_sat), or saturation temperature (T_sat).

Thermodynamic States of Steam II

Key Points

- Wet Steam State: Defined by dryness fraction $x = m_g / (m_f + m_g)$, representing the ratio of mass of dry vapor to total mixture mass. Properties depend linearly on x.
- Dry Saturated Steam State: The point where phase change is complete ($x = 1.0$) and further heat addition will cause superheating.
- Superheated Steam State: Steam at a temperature higher than its saturation temperature at a given pressure. Its state is defined by two independent properties: pressure (P) and temperature (T).
- Degree of Superheat: Expressed as ($T_{superheated}$ - T_{sat}), representing the temperature buffer preventing premature condensation during expansion.

Temperature-Entropy (T-s) Diagram I



Temperature-Entropy (T-s) Diagram II

Key Points

- Phase boundaries: The saturated liquid line and saturated vapor line meet at the Critical Point (322.06 bar, 373.95 °C for water).
- Constant pressure lines (isobars): Sloped upward in the subcooled liquid zone, horizontal in the wet region (isothermal phase change), and curved steeply upward in the superheat region.
- Wet Region: Located under the dome, where temperature and pressure are coupled variables, meaning T is fixed at T_sat for any given P.

Property Calculation for Saturated & Wet Steam I

For a two-phase mixture (wet steam), thermodynamic properties are calculated by combining saturated liquid (f) and dry saturated vapor (g) properties using the dryness fraction (x).

Key Points

- Specific Volume: $v = (1 - x)*v_f + x*v_g$. Because specific volume of liquid (v_f) is extremely small relative to vapor (v_g), it is often simplified to $v = x*v_g$ at low to moderate pressures.
- Specific Enthalpy: $h = h_f + x*h_{fg}$, where $h_{fg} = h_g - h_f$ is the latent heat of vaporization at the given saturation pressure.
- Specific Entropy: $s = s_f + x*s_{fg}$, representing the degree of molecular disorder. Saturated entropy of vaporization is $s_{fg} = h_{fg} / T_{sat}$ (T_{sat} in Kelvin).
- Specific Internal Energy: $u = u_f + x*u_{fg} = h - P*v$, where P is the absolute pressure in kPa and v is the specific volume in $m\hat{3}/kg$.

Property Calculation for Superheated Steam I

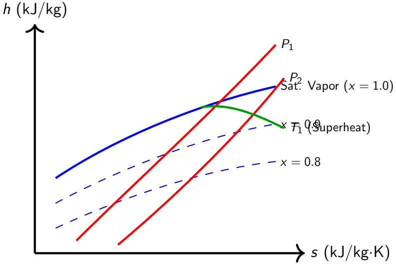
Superheated steam properties cannot be defined by pressure or temperature alone; both values must be specified to locate the state in the superheat tables. Linear interpolation is required for intermediate values.

Property Calculation for Superheated Steam II

Key Points

- Superheated Enthalpy: h_sup = h_g + C_ps * (T_sup - T_sat), where C_ps is the mean specific heat of superheated steam (typically assumed to be 2.1 to 2.3 kJ/kg·K depending on conditions).
- Superheated Entropy: s_sup = s_g + C_ps * ln(T_sup / T_sat), with temperatures expressed strictly in Kelvin.
- Double Interpolation: When both pressure and temperature fall between tabulated grid points, linear interpolation must be performed sequentially on temperature first, then on pressure.
- Physical Significance: Superheated steam contains no liquid moisture, eliminating droplet impingement and erosion on turbine blades while increasing cycle thermal efficiency.

The Mollier (h-s) Diagram I



The Mollier (h-s) Diagram II

Key Points

- Coordinates: The Mollier diagram plots specific enthalpy (h) as ordinate against specific entropy (s) as abscissa, focusing on the high-quality region.
- Slope of Isobars: In the wet region, constant pressure lines are straight lines. Their slope is equal to the absolute temperature: $(dh/ds)_P = T_{sat}$.
- Divergence in Superheat: Above the saturation curve, isobars curve upward and diverge from one another, whereas isotherms curve downward and tend to become horizontal at very low pressures.
- Expansion Visualization: Turbine expansion is represented as a vertical line downward (isentropic) for ideal flow, or slanting to the right for real flow (entropy generation).

A detailed industrial check: Find the enthalpy and specific volume of 3 kg of wet steam at 1.5 MPa (15 bar) with a dryness fraction of 0.92.

Step-by-Step Calculation Exercise II

Key Points

- Step 1: Extract saturation properties from steam tables at P = 1.5 MPa. Values: h_f = 844.89 kJ/kg, h_fg = 1947.3 kJ/kg, v_f = 0.001154 m$\hat{3}$/kg, v_g = 0.13177 m$\hat{3}$/kg.
- Step 2: Calculate specific enthalpy: h = h_f + x*h_fg = 844.89 + 0.92 * 1947.3 = 844.89 + 1791.52 = 2636.41 kJ/kg.
- Step 3: Calculate specific volume: v = (1 - x)*v_f + x*v_g = (0.08 * 0.001154) + (0.92 * 0.13177) = 0.000092 + 0.121228 = 0.12132 m$\hat{3}$/kg.
- Step 4: Compute total properties for mass m = 3 kg: Total Enthalpy H = m*h = 3 * 2636.41 = 7909.23 kJ. Total Volume V = m*v = 3 * 0.12132 = 0.36396 m$\hat{3}$.

In actual thermal power plants and chemical processing facilities, steam quality must be precisely measured and controlled to protect mechanical equipment and optimize thermodynamic efficiency.

Industrial Applications & Measurement Techniques II

Key Points

- Calorimetry: Dryness fraction is experimentally measured using separating calorimeters (for very wet steam), throttling calorimeters (for dry-ish steam), or combined calorimeters.
- Throttling Process: Wet steam is expanded through a narrow orifice. Because throttling is a constant enthalpy (isentropic-like but highly irreversible) process, the steam becomes superheated, allowing its state to be measured using T and P.
- Rankine Cycle Analysis: Mollier charts allow engineers to graphically determine turbine output, condenser heat rejection, and pump work without tedious multi-variable interpolation.
- Digital Steam Properties: Modern plant automation systems use standard digital algorithms (ASME / IAPWS-IF97 formulations) to compute properties dynamically for real-time mass flow and energy balance calculations.

Concept and Significance of Dryness Fraction I

Dryness fraction (x) is the ratio of mass of actual dry steam to the total mass of the wet steam mixture: $x = \frac{m_g}{m_f + m_g}$.

Key Points

- It defines the thermodynamic state of a liquid-vapor mixture, ranging from 0 (saturated liquid) to 1 (dry saturated vapor).
- Required to determine properties like specific enthalpy: $h = h_f + x \cdot h_{fg}$, and entropy: $s = s_f + x \cdot s_{fg}$.
- High moisture content (low x) causes mechanical erosion of turbine blades due to water droplet impingement.
- Accurate measurement is critical for calculating thermal efficiency in industrial boiler plants and steam networks.

Overview of Steam Calorimeters I

Classification of measurement devices based on steam state and quality range.

Key Points

- Barrel (Bucket) Calorimeter: Simple water-equivalent vessel method, suitable for medium quality steam ($x \approx 0.9$ to 0.95).
- Separating Calorimeter: Mechanical separation method using inertia, suitable for very wet steam ($x < 0.9$).
- Throttling Calorimeter: Constant enthalpy expansion process, suitable for dry steam ($x > 0.95$).
- Combined Separating and Throttling Calorimeter: Dual-stage system for precise measurement across the entire wet steam range.

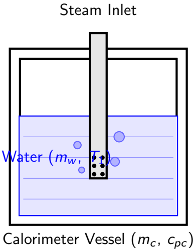
Barrel Calorimeter: Principle and Energy Balance I

Operates on the principle of heat exchange between a known mass of cold water and a condensed sample of wet steam.

Key Points

- A copper calorimeter contains a known mass of water (m_w) at an initial temperature (T_1).
- Steam of mass m_s is injected and fully condensed, raising the system temperature to a final equilibrium (T_2).
- Energy Balance: Heat lost by condensing and cooling steam equals heat gained by water and calorimeter vessel.
- Governing Equation: $m_s[x \cdot h_{fg} + c_{pw}(T_s - T_2)] = (m_w \cdot c_{pw} + m_c \cdot c_{pc})(T_2 - T_1)$, where m_c is calorimeter mass.

Schematic of a Barrel Calorimeter I



Schematic of a Barrel Calorimeter II

Key Points

- The perforated nozzle ensures complete steam distribution and condensation within the water volume.
- An air gap or wood lining (represented by double lines) reduces heat transfer to the environment.
- Continuous stirring is required to ensure a uniform final mixture temperature (T_2).
- Heat loss by radiation and water droplet carryover represent the primary sources of experimental error.

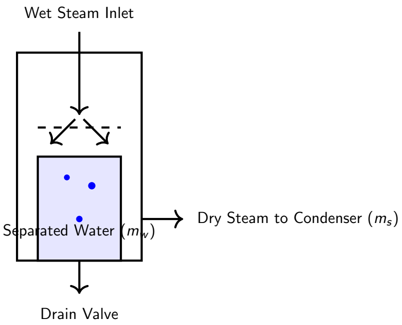
Separating Calorimeter: Working Principle I

A mechanical separator that extracts liquid droplets from wet steam using inertia and sudden changes in flow direction.

Key Points

- Mainly designed for very wet steam ($x < 0.9$) where thermodynamic throttling methods are not directly applicable.
- Wet steam enters the inner chamber through a nozzle, forcing a sharp 180° turn.
- Due to density differences, heavier water droplets fail to turn and are captured in an inner collecting chamber.
- The dry steam is routed through an outer annular space to a surface condenser where its mass (m_s) is measured.
- The dryness fraction is calculated as: $x = \frac{m_s}{m_s + m_w}$, where m_w is the mass of collected liquid water.

Separating Calorimeter Schematic Diagram I



Separating Calorimeter Schematic Diagram II

Key Points

- Steam direction reversal at the perforated baffle plate separates liquid water by centrifugal force.
- Separated liquid accumulates at the bottom and its level is monitored using a calibrated glass gauge.
- This device measures an 'apparent' dryness fraction because some fine water mist remains entrained.
- Normally combined with a throttling calorimeter to measure the quality of the remaining steam.

Industrial Applications and Selection Criteria I

Selecting the correct calorimetry technique based on steam quality and application constraints.

Key Points

- Bucket calorimeters are suitable for quick, low-precision, field-based steam quality checks.
- Separating calorimeters are preferred for high-moisture steam lines preceding turbine extraction stages.
- Throttling calorimeters require low-moisture steam to guarantee superheat at low exit pressures.
- Combined systems provide a multiplicative dryness calculation: $x_{total} = x_1 \cdot x_2$, where x_1 is separating quality and x_2 is throttling quality.

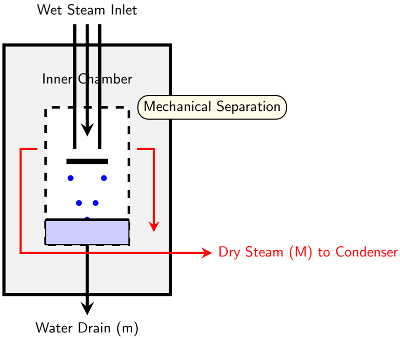
Separating Calorimeter: Working Principle I

Mechanical separation of water droplets from wet steam using inertia and centrifugal force.

Key Points

- Wet steam is introduced into the inner chamber where it undergoes a sudden change of direction.
- Heavy water droplets cannot change direction rapidly due to inertia, separating from the steam.
- Dry steam flows upward through perforated walls and is led to a condenser to measure its mass (M).
- Separated water is collected at the bottom of the separator and measured as mass (m).
- Apparent dryness fraction: $x1 = M / (M + m)$. It is an approximation as complete separation is not possible.

Schematic of Separating Calorimeter I



Schematic of Separating Calorimeter II

Key Points

- The schematic represents the mechanical separation chamber where centrifugal force acts on droplets.
- Inner perforated cylinder allows dry steam to exit to the outer jacket, acting as an insulating layer.
- Mass of separated water (m) is drained and weighed, while dry steam (M) is condensed and weighed.

Throttling Calorimeter: Theory & Principle I

Isenthalpic expansion of wet steam to superheated state across a throttling orifice.

Key Points

- Throttling is a constant enthalpy process (h1 = h2) under steady flow, adiabatic conditions.
- Wet steam at high pressure P1 is expanded to a lower pressure P2 (often atmospheric) through a narrow orifice.
- If the initial dryness fraction is high (x ¿ 0.95), the steam becomes superheated after expansion.
- Enthalpy at State 1: h1 = hf1 + x2 * hfg1 (where x2 is the dryness fraction determined by throttling).
- Enthalpy at State 2: h2 = hg2 + Cp * (T_superheated - Tsat2), where Cp is specific heat of superheated steam (2.1 kJ/kg.K).
- Equating h1 and h2 allows direct computation of x2.

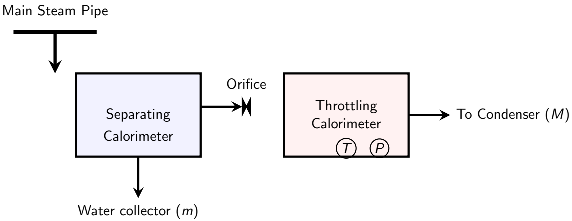
Combined Separating & Throttling Calorimeter I

Combining mechanical separation and throttling to measure a wide range of dryness fractions.

Key Points

- A throttling calorimeter cannot measure dryness fractions below 0.95 because the steam will not become superheated.
- A separating calorimeter alone is inaccurate because it fails to capture fine water droplets.
- In the combined setup, wet steam first enters the separating calorimeter where mass 'm' of water is separated.
- The remaining dryer steam (with mass 'M' and dryness x2) then undergoes throttling to become superheated.
- Total dryness fraction of the initial steam: $x = x1 * x2 = [M / (M + m)] * x2$.

Combined Separating & Throttling Calorimeter Layout I



Combined Separating & Throttling Calorimeter Layout II

Key Points

- Block diagram showing the series arrangement: steam flows through the separator first, then the throttling orifice.
- Thermometer (T) and pressure gauge (P) are placed after the orifice to confirm superheating.
- The exhaust steam is condensed to measure the mass flow rate of dry steam (M).

Electrical Calorimeter: Principles I

Determining dryness fraction by adding a known amount of electrical heat to superheat the steam.

Electrical Calorimeter: Principles II

Key Points

- Wet steam is passed through a well-insulated chamber containing an electrical heating element.
- Electric power input Q = V * I (where V is voltage and I is current) is adjusted until the steam becomes superheated.
- Energy conservation: Mass flow rate (m_dot) * h1 + Electrical Power = Mass flow rate (m_dot) * h2.
- Enthalpy equation: h1 + (V * I / m_dot) = hg2 + Cp * (T_superheated - Tsat2).
- Solving for x: x = [hg2 + Cp * (T_superheated - Tsat2) - (V * I / m_dot) - hf1] / hfg1.
- Advantage: Does not rely on pressure drop alone; heating rate can be adjusted dynamically.

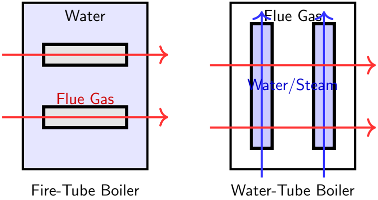
Step-by-step calculation of steam dryness fraction from combined calorimeter data.

Numerical Application of Combined Calorimeter II

Key Points

- Given: m = 2.0 kg, M = 20.5 kg. Inlet pressure P1 = 12 bar. Exit pressure P2 = 1.013 bar, T2 = 120 C.
- Properties at 12 bar: hf1 = 798.4 kJ/kg, hfg1 = 1984.3 kJ/kg. Properties at 1.013 bar: hg2 = 2676 kJ/kg, Tsat2 = 100 C, Cp = 2.1 kJ/kg.K.
- Step 1 (Separation dryness): x1 = M / (M + m) = 20.5 / (20.5 + 2) = 0.9111.
- Step 2 (Throttling dryness): h2 = 2676 + 2.1 * (120 - 100) = 2718 kJ/kg. Since h1 = h2: x2 = (2718 - 798.4) / 1984.3 = 0.9674.
- Step 3 (Total dryness): x = x1 * x2 = 0.9111 * 0.9674 = 0.8814 (or 88.14 percent).

Classification: Fire-Tube vs. Water-Tube Boilers I



Classification: Fire-Tube vs. Water-Tube Boilers II

Key Points

- Fire-Tube Boilers: Hot combustion gases pass through tubes; water surrounds the tubes. Examples: Cochran, Lancashire, Locomotive, Scotch Marine boilers.
- Water-Tube Boilers: Water circulates inside the tubes; combustion gases flow around the outside. Examples: Babcock & Wilcox, Stirling, Yarrow boilers.
- Operating Range: Fire-tube boilers are typically limited to low-to-medium pressures (¡ 25 bar) and smaller capacities (¡ 20 tons/hr) due to shell thickness constraints.
- Water-tube boilers are suitable for high-pressure power generation (¿ 100 bar, up to supercritical pressures) and very large steam generation capacities.

Cochran Boiler: Multi-Tubular Vertical Fire-Tube I

Key design parameters and engineering features of Cochran boiler: a vertical, multi-tubular, internally fired, natural circulation fire-tube boiler.

Key Points

- Vertical shell minimizes floor space requirement, making it highly compact for small-scale industries.
- Hemispherical crown of the shell provides maximum strength to withstand internal steam pressure with minimal bracing.
- Grate is located at the bottom of the firebox, and coal/fuel is fired internally. Flue gases go from firebox to combustion chamber, then through horizontal fire tubes.
- Combustion chamber is lined with firebricks to prevent heat loss and ensure complete combustion before gases enter the fire tubes.
- Typical technical parameters: Operating pressure up to 15 bar, steam capacity up to 3500 kg/hr, and thermal efficiency around 70-75%.

Babcock & Wilcox Boiler: Longitudinal Drum Water-Tube I

A horizontal, externally fired, natural circulation water-tube boiler designed for high pressure and power production applications.

Key Points

- Consists of a horizontal steel steam-and-water drum connected to front and rear headers via inclined water tubes (inclined at 15 degrees to horizontal).
- Inclination of tubes promotes rapid natural circulation of water due to density differences (thermosyphon effect) between hot water in tubes and cooler water in headers.
- Baffles force the hot flue gases to pass over the water tubes three times (three-pass design) to maximize convective heat transfer efficiency.

Babcock and Wilcox Boiler (Cont.) I

Key Points

- Equipped with a superheater placed above the water tubes to heat the saturated steam to superheated steam, increasing Rankine cycle efficiency.
- Performance parameters: Can operate at pressures up to 40 bar and generate steam up to 40,000 kg/hr, with thermal efficiencies exceeding 80%.

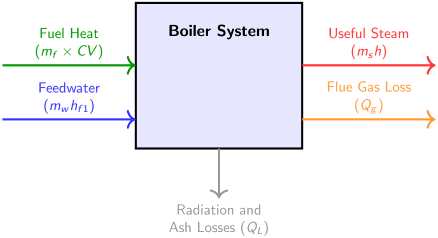
Boiler Mountings and Accessories I

Distinction between safety/operational devices (mountings) and efficiency-boosting devices (accessories).

Key Points

- Boiler Mountings: Essential fittings mounted directly on the boiler shell for safety and proper operation. Required by boiler regulations (e.g., Indian Boiler Regulations - IBR).
- Examples of Mountings: Safety valves (lever, spring-loaded), Water level indicator, Pressure gauge, Steam stop valve, Feed check valve, Blow-off cock, Fusible plug.
- Boiler Accessories: Auxiliary devices installed to increase the overall thermal and combustion efficiency of the boiler system.
- Examples of Accessories: Feed pump, Injector, Economizer (heats feed water using flue gas), Air preheater (heats combustion air), Superheater (increases steam temperature).

Boiler Performance Parameters & Governing Equations I



Boiler Performance Parameters & Governing Equations II

Key Points

- Equivalent Evaporation (E_e): Amount of water evaporated from and at 100°C into dry saturated steam. Formula: $E_e = \frac{m_s(h - h_{f1})}{2257}$ kg/kg of fuel, where 2257 kJ/kg is the latent heat of vaporization.
- Boiler Efficiency (Thermal Efficiency, η_{th}): Ratio of heat utilized in steam generation to the heat liberated by combustion of fuel. Formula: $\eta_{th} = \frac{m_s(h - h_{f1})}{m_f \times CV} \times 100\%$.
- Factor of Evaporation (F_e): Ratio of heat received by 1 kg of water under boiler conditions to the heat required to evaporate 1 kg of water from and at 100°C. $F_e = \frac{h - h_{f1}}{2257}$.
- Heat Balance Sheet: A detailed accounting of thermal energy input vs. output. Typically lists steam generation (useful), dry flue gas heat loss, moisture in fuel loss, unburnt carbon in ash, radiation, and unaccounted losses.

Boiler Draught (Draft): Natural vs. Artificial I

Analysis of pressure differentials required to maintain continuous flow of combustion air and flue gases through the boiler.

Key Points

- Definition: Draught is the small pressure difference required to force air through the fuel bed and to discharge the products of combustion through the flue, chimney, and accessories.
- Natural Draught: Produced by the chimney alone due to the difference in density between hot flue gases inside the chimney and cold ambient air outside.

Key Points

- Height of Chimney (H) calculation for draught pressure (h_w in mm of water): $h_w = 353H\left(\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right)$, where T_a and T_g are temperatures in Kelvin, and m is the air-fuel ratio.
- Artificial Draught: Mechanical draft using fans. Forced Draught (FD fan pushes air in), Induced Draught (ID fan draws flue gas out, placed near chimney base), and Balanced Draught (combination of FD and ID).
- Steam Jet Draught: Uses kinetic energy of high-velocity steam jets. Can be induced (placed in chimney) or forced (placed in ash pit).

Modern High-Pressure Boilers: Supercritical & Once-Through I

Advanced steam generators operating above critical pressure (221.2 bar, 374.15°C) to maximize modern steam turbine plant efficiencies.

Modern High-Pressure Boilers: Supercritical & Once-Through II

Key Points

- LaMont Boiler: Forced circulation boiler using a centrifugal pump to circulate water through headers and tubes at high velocity, preventing overheating.
- Benson Boiler: Once-through water-tube boiler that eliminates the steam separator drum. Operates at supercritical pressure where water converts directly to steam without boiling.
- Loeffler Boiler: Uses superheated steam to evaporate water. Steam is circulated through tubes, and only a portion is sent to the turbine; the rest goes to the evaporator drum.
- Supercritical Operation: Eliminates two-phase flow instabilities, increases plant efficiency up to 45%, and reduces CO_2 emissions per megawatt-hour generated.

History and Purpose of IBR I

The Indian Boilers Act was enacted in 1923 to regulate the manufacture, installation, and maintenance of boilers to ensure safety and prevent explosions. The Indian Boilers Regulations (IBR) were formed under Section 28 of this Act to provide standard technical codes.

History and Purpose of IBR II

Key Points

- Enacted primarily as safety legislation following numerous catastrophic boiler explosions in industrial areas during the late 19th and early 20th centuries.
- Created a unified set of regulations to standardize the design, construction, inspection, and testing of boilers throughout India.
- IBR 1950 is the comprehensive code containing detailed guidelines on material specifications, design calculations, piping layouts, and testing procedures.
- Legally mandates that no boiler can be operated without a valid certificate issued by the State Boiler Directorate.

Technical Definition of 'Boiler' under IBR I

Under Section 2(b) of the Indian Boilers Act, 1923, a 'boiler' is defined strictly based on volumetric capacity and pressure generation capabilities.

Key Points

- Defines a 'boiler' as any closed vessel exceeding 22.75 liters (5 gallons) in capacity, which is used expressively for generating steam under pressure.
- The capacity calculation includes the volume of all fittings and mountings attached to the vessel that are under pressure.
- Includes any mounting or other fitting attached to such vessel, which is wholly or partly under pressure when steam is shut off.
- Excludes vessels that do not meet the minimum capacity or are operated entirely at atmospheric pressure without steam containment.

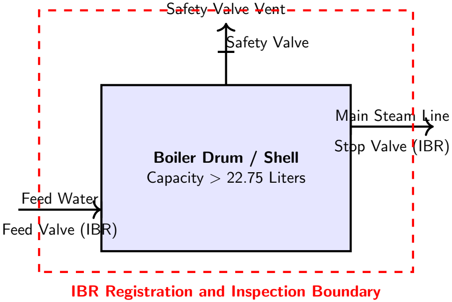
Definitions of Steam Pipe and Feed Pipe I

The IBR extends its regulation to critical associated piping systems to ensure overall system integrity and safety during high-pressure operation.

Key Points

- Steam Pipe: Any pipe through which steam passes from a boiler to a user plant if the pressure exceeds 3.5 kg/cm$\hat{2}$ (gauge) or the design temperature exceeds 220°C.
- Also applies to steam pipes where the internal diameter exceeds 254 mm (10 inches) even if operating under lower pressure limits.
- Feed Pipe: Any pipe through which feed water is delivered to the boiler under pressure, typically terminating at the feed check valve.
- Boiler Component: Any part of a boiler, including superheaters, economizers, headers, and safety valves, which is subject to internal steam/water pressure.

IBR Boundary and Control Volume I



IBR Boundary and Control Volume II

Key Points

- The diagram defines the physical scope of IBR regulations, spanning from the feed check valve to the main steam stop valve.
- Any fitting, mounting, or piping within this dashed boundary is legally part of the boiler and requires official certification.
- The Feed Water inlet must have a non-return valve and a feed stop valve, both conforming to IBR material standards.
- Steam Outlet: The main steam stop valve acts as the terminal point of the boiler proper, beyond which steam pipe regulations apply.

IBR Design Standards & Material Certification I

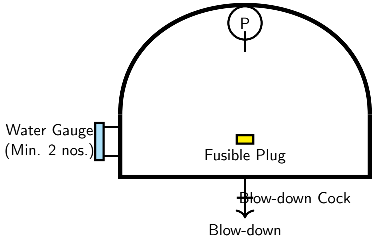
IBR specifies strict regulations for design calculations, allowable stresses, and material certificates (Form II-A, Form III-A, etc.).

IBR Design Standards & Material Certification II

Key Points

- Equation for Shell Thickness (Regulation 270): $t = \frac{P \cdot D}{2 \cdot f \cdot \eta - P} + C$, where P is design pressure, D is internal diameter, f is allowable stress, η is joint efficiency, and C is corrosion allowance.
- Allowable Stress (f): Determined based on the minimum yield strength or ultimate tensile strength of the material at design temperature, divided by a factor of safety.
- Material Traceability: All plates, tubes, and castings must be tested at a recognized laboratory and certified by an IBR Inspector using Form IV.
- Fabrication: Welding must be performed by certified IBR high-pressure welders, and non-destructive testing (NDT) such as radiography is mandatory.

IBR Mandatory Boiler Mountings I



Mandatory Mountings as per Regulation 281

IBR Mandatory Boiler Mountings II

Key Points

- IBR Regulation 281 mandates specific mountings for safe operation of steam boilers.
- Two safety valves (minimum) must be fitted, each capable of discharging the total peak evaporative capacity of the boiler without pressure rise exceeding 10%.
- Two independent water level indicators (gauges) are required to prevent dry-running and tube overheating.
- A pressure gauge, blow-down valve, feed check valve, and fusible plug (for fire-tube boilers) are strictly required for safe operation.

IBR Registration & Inspection Procedure I

A boiler cannot be operated legally in India unless it is registered and certified annually by the State Boiler Inspectorate.

IBR Registration & Inspection Procedure II

Key Points

- Submission of Drawings: Detailed design drawings and calculations must be approved by the Chief Inspector of Boilers (CIB) before fabrication begins.
- Stage-wise Inspection: Inspections are conducted during fabrication (material verification, weld inspection, radiography) and erection at the site.
- Hydraulic Test: The completed boiler must withstand a hydraulic test at 1.5 times the maximum allowable working pressure (MAWP) for IBR certification.
- Annual Renewal: The boiler inspector conducts an open inspection and steam test annually to renew the operating certificate (Form VI).

Classification Criteria of Steam Generators I

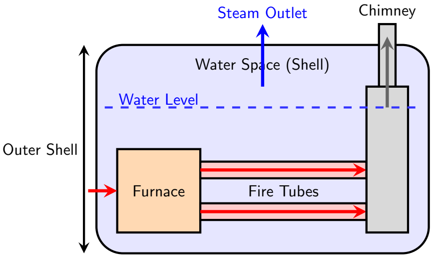
Steam generators (boilers) are classified based on several operating and design parameters to suit specific industrial applications.

Classification Criteria of Steam Generators II

Key Points

- Relative position of hot gases and water: Fire-tube boilers (gases inside tubes, water outside) vs. Water-tube boilers (water inside tubes, gases outside).
- Method of water circulation: Natural circulation (density-difference driven, subcritical pressures ¡ 180 bar) vs. Forced circulation (pump-assisted flow, used in high-pressure and supercritical boilers).
- Operating pressure: Low pressure (¡ 20 bar), Medium pressure (20 to 80 bar), High pressure (¿ 80 bar), and Supercritical (above thermodynamic critical point of water, 221.2 bar, 374.15°C).
- Axis of shell: Horizontal, Vertical, or Inclined design, influencing the footprint, floor space requirements, and gas flow path.

Schematic of a Fire-Tube Boiler I



Schematic of a Fire-Tube Boiler II

Key Points

- Characterized by flue gases passing through tubes surrounded by water in a closed shell, limiting maximum operating pressure to 20 bar due to shell thickness constraints ($t = P \cdot D/(2\sigma \cdot \eta)$).
- Common examples include Cochran (vertical multi-tubular), Lancashire (horizontal, internal fire, double flues), and Cornish (single flue) boilers.
- Highly suited for low-pressure steam applications, heating systems, small-scale chemical industries, and mobile applications (historical steam locomotives).
- Boasts high steam storage capacity (large water volume) which helps absorb sudden load fluctuations, but has slow starting characteristics and is prone to explosion risks of the main shell.

Water-Tube Boilers: High-Pressure Steam Generation I

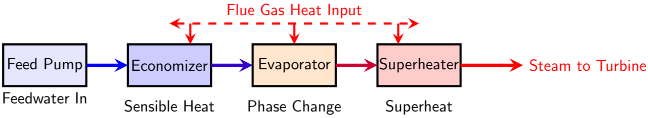
Water-tube boilers circulate water inside tubes while hot combustion gases flow over the external surfaces, facilitating high-pressure and high-temperature steam generation.

Water-Tube Boilers: High-Pressure Steam Generation II

Key Points

- Operating pressures can exceed critical levels (up to 300 bar) and temperatures up to 600°C, significantly improving Rankine cycle efficiency ($\eta = 1 - T_{cold}/T_{hot}$).
- Examples include Babcock & Wilcox (longitudinal/cross drum, natural circulation), Stirling (multi-drum bent-tube), and Benson (forced circulation, once-through supercritical boiler).
- Smaller water holdup volume results in rapid response to steam demand changes and decreases catastrophic failure risks (explosions are localized to individual tubes).
- Used extensively in utility thermal power plants, cogeneration facilities, and heavy industrial steam supply systems.

Benson Once-Through Boiler Flow Circuit I



Benson Once-Through Boiler Flow Circuit II

Key Points

- A supercritical, once-through water-tube boiler operating above 221.2 bar, eliminating the need for a steam-separating drum.
- Water is pumped directly through the economizer, evaporator tubes, and superheater in a single continuous path, converting entirely to steam without bubbling.
- Avoids the bubble-formation issue which reduces heat transfer coefficient under subcritical conditions; operates with high heat flux ($q' > 500$ kW/m^2).
- Key components shown in the diagram: Feed pump, Economizer (sensible heating), Evaporator (phase change / transition zone), and Superheater (superheating to high temperature).

The circulation ratio (CR) is defined as the ratio of the mass flow rate of water circulating through the evaporator tubes to the mass flow rate of steam produced: $CR = \dot{m}_{water}/\dot{m}_{steam}$.

Natural vs. Forced Circulation Mechanics II

Key Points

- Natural Circulation: Relies on density differences ($\rho_{water} - \rho_{steam-water}$) between the cold downcomer and hot riser. CR typically ranges from 5 to 25. Self-regulating but limited to pressures ¡ 180 bar.
- Forced Circulation: Uses a boiler circulating pump (BCP) to overcome flow resistance. CR is lower (typically 3 to 10), enabling smaller tube diameters, flexible tube routing, and higher heat absorption rates per unit area.
- Critical Pressure Limit: As pressure approaches critical pressure ($P_{crit} = 221.2$ bar), the density difference between saturated water and saturated steam approaches zero ($\Delta \rho arrow 0$), rendering natural circulation impossible.
- Once-through systems (e.g., Benson, Sulzer) have a circulation ratio of exactly $CR = 1$, where feedwater entering is completely evaporated and superheated in a single pass.

Industrial Applications and Selection Criteria I

Steam generators are selected based on the specific industrial process requirements, considering steam parameters (pressure, temperature, flow rate) and fuel availability.

Industrial Applications and Selection Criteria II

Key Points

- Power Generation: High-capacity water-tube boilers (pulverized coal, circulating fluidized bed, or natural gas-fired) generating superheated steam for steam turbines (Rankine cycle).
- Process Industries: Food processing, paper mills, textiles, and refineries use low-to-medium pressure steam (often saturated) for process heating, sterilization, and distillation.
- Cogeneration (CHP): Combined Heat and Power plants generate electricity and utilize exhaust steam/heat for district heating or industrial processes, achieving thermal efficiencies ¿ 80%.
- Waste Heat Recovery Steam Generators (HRSGs): Installed at the exhaust of gas turbines in Combined Cycle Power Plants (CCPP) to capture energy from hot exhaust gases (500-600°C) without burning additional fuel.

Boiler Selection Matrix I

Key engineering parameters compared between major steam generator types.

Boiler Selection Matrix II

Key Points

- Parameter — Fire-Tube — Water-Tube (Subcritical) — Once-Through (Supercritical)
- Operating Pressure Range: Low (¡ 20 bar) — Medium-High (20 to 180 bar) — Very High (Supercritical, ¿ 221.2 bar)
- Rate of Steam Generation: Low (up to 15 tons/hr) — High (up to 300 tons/hr) — Very High (exceeding 1000 tons/hr)
- Fuel Flexibility: High (can run on cheap fuels easily) — High (pulverized fuel, oil, gas, biomass) — Strict water purity requirements to prevent tube hot-spots
- Load Response: Slow (high water inventory acts as thermal buffer) — Fast (moderate water inventory) — Extremely Fast (once-through, responsive to fuel-feedwater ratio adjustments)

Introduction to Packaged Boilers I

A packaged boiler is a shop-assembled, self-contained unit complete with burner, controls, pump, and auxiliary equipment, shipped fully built to the site on a skid. This design paradigm shifts fabrication from the field to a controlled factory environment.

Introduction to Packaged Boilers II

Key Points

- Factory Assembly: Fabricated, piped, wired, and tested in a controlled factory environment to ensure adherence to strict quality control codes (ASME Section I or BS EN 12953).
- Reduced Installation Time: Arrives on-site ready for installation, requiring only external piping, electrical hookups, fuel supply line, and flue gas stack connection.
- Forced Draft Design: Utilizes forced or induced draft fans to maintain high combustion rates within a small furnace volume, increasing heat transfer efficiency.
- Space & Capacity Limits: Compact dimensions are constrained by transport limitations (road/rail widths); capacities generally range up to 40 tonnes/hour for shell/fire-tube and 100 tonnes/hour for water-tube packages.

Packaged Fire-Tube Boilers I

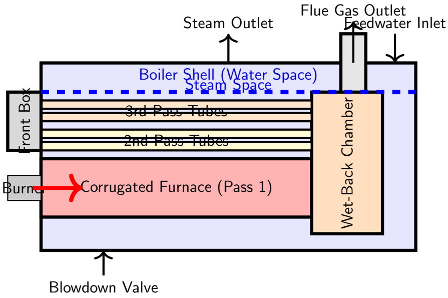
In fire-tube packaged boilers, high-temperature combustion gases pass inside the tubes while water surrounds the outside of the tubes within the main pressure vessel shell. Modern industrial packages are designed as multi-pass wet-back or dry-back boilers.

Packaged Fire-Tube Boilers II

Key Points

- Wet-Back Design: The combustion reversal chamber at the back of the furnace is completely surrounded by boiler water. This maximizes heating surface area and eliminates the thermal cracking issues associated with dry-back refractory walls.
- Three-Pass Gas Flow: Pass 1 is through the primary furnace tube (combustion chamber); Pass 2 is through the lower set of convection tubes; Pass 3 is through the upper set of convection tubes before exiting to the stack.
- Corrugated Furnace: Utilizes a corrugated structure (e.g. Morison furnace) to accommodate longitudinal thermal expansion without creating high localized stresses on the tube sheets.
- Applications: Ideal for heating loads and moderate industrial steam requirements up to 25 bar pressure, offering a high thermal inertia due to large water storage capacity.

Three-Pass Wet-Back Fire-Tube Boiler Layout I



Three-Pass Wet-Back Fire-Tube Boiler Layout II

Key Points

- Radiant Heat Extraction: Radiant energy from the burner flame is primarily absorbed by the water-cooled furnace walls (Pass 1).
- Convective Heat Transfer: Convective heat transfer dominates in Pass 2 and Pass 3 tubes, where gas velocities are optimized for thermal exchange.
- Wet-Back Thermal Efficiency: By eliminating the dry-back rear refractory dome, heat loss is minimized, and maintenance costs are significantly lowered.
- Safety and Control Ports: Equipped with multiple safety valves, water level gauge glasses, feed check valves, and a bottom blowdown valve to purge dissolved solids.

Packaged Water-Tube Boilers I

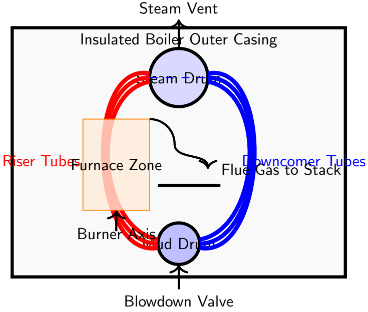
In water-tube packaged boilers, high-pressure water and steam circulate inside the tubes while the combustion furnace and hot flue gases flow around the outer surface of these tubes. This configuration is essential for high-capacity, high-pressure utility and process systems.

Packaged Water-Tube Boilers II

Key Points

- Pressure Capability: Designed to handle high-pressure operation (up to 70+ bar) because the smaller tube diameters can withstand higher internal pressures with thinner walls compared to a large-diameter fire-tube shell.
- D-Type Boiler Architecture: Comprises a larger steam drum at the top and a smaller water (or mud) drum at the bottom. Tubes form a 'D' shape enclosing the furnace, providing a compact, self-shielding radiant combustion chamber.
- Rapid Load Response: Small water volume in the tubes means a much lower thermal inertia, permitting fast startup sequences and agile response to steam demand changes.
- Critical Water Quality: Requires strict deaeration and chemical dosing. Even thin scale deposits (less than 1 mm) of calcium carbonate or silica can lead to localized tube overheating, bulging, and catastrophic tube rupture.

D-Type Packaged Water-Tube Boiler Layout I



D-Type Packaged Water-Tube Boiler Layout II

Key Points

- Natural Circulation Loop: Driven by density difference between the hot steam-water mixture in the radiant risers and the cooler, denser subcooled water in the convective downcomers.
- Steam Separation: Mechanical cyclones and chevron-type baffles in the steam drum extract entrained water droplets from steam, ensuring a steam dryness fraction above 99.5%.
- Mud Drum Sedimentation: Non-volatile solids and sludge accumulate in the lower drum and are blown down periodically to maintain purity.
- Refractory Construction: Internal baffles direct gas flow across the downcomer bank, extracting convective heat before exit.

Combined Tube / Hybrid Packaged Boilers I

Combined-tube or hybrid packaged boilers merge fire-tube and water-tube heat transfer technologies into a single plant design. They optimize efficiency, cost, and load handling, particularly for solid fuel or waste-heat applications.

Combined Tube / Hybrid Packaged Boilers II

Key Points

- Hybrid Geometry: Consists of an external water-tube radiant furnace section linked to a shell-and-tube fire-tube convection block.
- Combustion Adaptability: Water-tube membrane walls define a large furnace chamber ideal for slow-burning, high-moisture fuels (biomass, wood chips, bagasse) without thermal shock to tube sheets.
- Convective Section Optimization: The fire-tube shell acts as a compact convective pass, capturing residual heat from flue gases with low draft-loss while maintaining a large steam storage volume.
- Thermal Expansion Management: Separating the high-temperature radiant zone (water-tube) from the convection zone (fire-tube) minimizes differential thermal expansion stress on critical joints.

Boiler Calculations & Operational Parameters I

Thermodynamic evaluation of steam boilers requires analyzing energy input versus energy absorption. Standard performance indicators include equivalent evaporation, factor of evaporation, and direct thermal efficiency.

Boiler Calculations & Operational Parameters II

Key Points

- Equivalent Evaporation (E_e): Defined as the amount of water evaporated from and at 100°C per hour. Calculated as: $E_e = \frac{m_s(h-h_f)}{2257}$ kg/h, where m_s is steam flow rate (kg/h), h is steam enthalpy (kJ/kg), and h_f is feed water enthalpy (kJ/kg).
- Factor of Evaporation (FE): Represents the ratio of actual heat absorbed to the heat required to evaporate water at 100°C: $FE = \frac{h-h_f}{2257}$. Actual evaporation multiplied by FE equals equivalent evaporation.
- Direct Efficiency (Input-Output Method): Calculated as: $\eta = \frac{m_s(h-h_f)}{m_f \times LCV} \times 100$, where m_f is fuel flow rate (kg/h) and LCV is the lower calorific value of the fuel (kJ/kg).
- Heat Loss Distribution (Indirect Method): Accounted for by: $\eta_{ind} = 100 - (L_G + L_H + L_M + L_R + L_U)$ where L_G is dry flue gas loss, L_H is moisture loss from combustion of hydrogen, L_M is fuel moisture loss, L_R is radiation loss, and L_U is unaccounted losses.

Thermodynamic Fundamentals of WHRB I

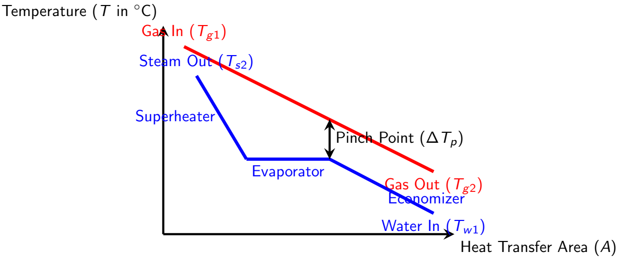
A Waste Heat Recovery Boiler (WHRB) recovers sensible heat from hot exhaust gases (400°C - 650°C) of gas turbines, internal combustion engines, or industrial kilns to generate high-quality steam.

Thermodynamic Fundamentals of WHRB II

Key Points

- Efficiency Optimization: Simple cycle gas turbine thermal efficiency is limited to 35-40%; integrating a WHRB raises combined cycle power plant (CCPP) thermal efficiency above 60%.
- Convective Heat Transfer Dominance: Unlike direct-fired boilers that rely heavily on radiative heat transfer from a high-temperature flame, WHRBs are dominated by convective heat transfer.
- Gas-Side Pressure Drop: Operating WHRBs imposes a backpressure on upstream equipment (e.g., gas turbines). Allowable gas-side pressure drop is typically restricted to 250-400 mm of water column to prevent turbine output degradation.

WHRB Temperature Profile & Heat Transfer I



WHRB Temperature Profile & Heat Transfer II

Key Points

- Pinch Point (ΔT_p): The minimum temperature difference between the cooling gas and the evaporating water/steam. Typically designed between 8°C and 15°C; smaller values maximize recovery but require exponentially larger heat exchanger surface area.
- Approach Point (ΔT_a): The difference between the saturation temperature and the temperature of the water leaving the economizer ($T_{sat} - T_{eco,out}$). Usually kept at 5°C - 10°C to prevent steaming/boiling inside the economizer tubes under transient loads.
- Exergy Optimization: The heat recovery curve must match the gas cooling curve as closely as possible to minimize irreversibilities and maximize exergy transfer.

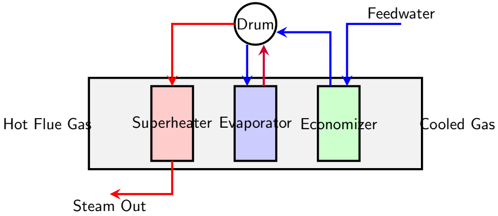
WHRBs are engineered in various configurations depending on exhaust gas cleanliness, pressure levels, and structural spatial constraints.

Classification and Structural Designs II

Key Points

- Water-Tube vs. Fire-Tube: Water-tube designs are standard for high-capacity, high-pressure utility applications like gas turbine exhausts. Fire-tube boilers are selected for lower pressure, particulate-laden exhaust streams (e.g., diesel engines, chemical waste gas).
- Natural vs. Forced Circulation: Natural circulation utilizes density differentials between hot water/steam risers and downcomers. Forced circulation uses circulation pumps to permit horizontal tube configurations and compact layouts.
- Pressure Level Configurations: Single-pressure systems are cost-effective but leave significant heat in stack gas. Multi-pressure (dual or triple-pressure) systems extract heat at different saturation levels to maximize heat recovery.

Schematic of a Single-Pressure WHRB Cycle I



Schematic of a Single-Pressure WHRB Cycle II

Key Points

- Economizer (ECO): Located at the coolest zone of the gas duct. Preheats feed water to a temperature just below saturation before entering the steam drum.
- Evaporator (EVAP): Performs phase change (boiling) at constant saturation temperature. High heat transfer rates are maintained via boiling heat transfer mechanism.
- Superheater (SH): Located at the hottest gas zone (gas inlet). Raises saturated steam temperature to superheated levels to prevent condensation during turbine expansion.
- Steam Drum: Functions as a separator separating dry steam from liquid water. Supplies saturated steam to the superheater and recirculates saturated water to the evaporator.

Designing a WHRB requires calculating heat exchanger duties, log-mean temperature differences (LMTD), and necessary heat transfer surface areas.

Thermodynamic Calculations & Design Equations II

Key Points

- Overall Thermal Balance:
  $Q = \dot{m}_g C_{pg}(T_{g,in} - T_{g,out})\eta_{th} = \dot{m}_s(h_{steam} - h_{feedwater})$, where η_{th} represents radiation and convection casing loss factors (typically 0.985 - 0.995).
- Heat Transfer Rate: $Q = UA\Delta T_{lm}$, where U is the overall heat transfer coefficient, A is the heat transfer area, and ΔT_{lm} is the Logarithmic Mean Temperature Difference.
- LMTD Calculation: $\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$, computed independently for the Economizer, Evaporator, and Superheater sections.
- Finned Tubes: Because the gas-side heat transfer coefficient (h_g) is low, spiral or solid-finned tubes are extensively used to increase gas-side surface area (A_{eff}) by 10 to 15 times.

Operational Challenges: Corrosion & Fouling I

Industrial WHRB operations face structural durability issues due to high-temperature gradients and chemical composition of flue gases.

Operational Challenges: Corrosion & Fouling II

Key Points

- Cold-End Acid Corrosion: If fuel contains sulfur, flue gases contain SO_x which reacts with water vapor to form sulfuric acid (H_2SO_4). If economizer tube wall temperatures drop below the acid dew point (110°C - 130°C), severe corrosion occurs. Prevented by preheating feedwater.
- Flue Gas Fouling: Particulate matter (soot, fly ash) deposits on heat exchanger tubes, increasing thermal resistance. Addressed by installing soot blowers (steam/sonic) and maintaining gas velocity between 15 - 25 m/s.
- Thermal Stress Management: Fast startup times in gas turbine systems create transient thermal gradients in thick-walled drums and headers, requiring fatigue analysis and drum bypass stack systems during startup.

WHRBs are critical components across diverse sectors, yielding high economic and environmental returns.

Industrial Applications & Performance Analysis II

Key Points

- Combined Cycle Power Plants (CCPP): Multi-pressure HRSGs generate high, medium, and low-pressure steam, driving a steam turbine cycle that extracts maximum work from gas turbine exhaust.
- Steel and Metallurgical Plants: Recovers massive energy from Basic Oxygen Furnaces (BOF), blast furnaces, and coke dry quenching (CDQ) processes.
- Cement Industry: Integrates WHRBs into the preheater and clinker cooler exhaust systems to generate up to 30% of the plant's internal electrical power requirements.
- Carbon Abatement: Displacing fossil-fuel-fired auxiliary boilers directly translates into significant reductions in plant greenhouse gas emissions.

The Physics of Fluidization I

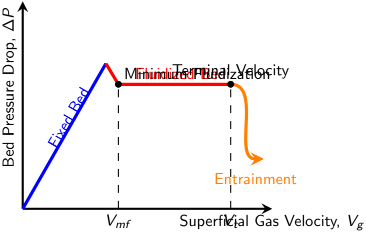
Fluidization occurs when a gas flows upward through a bed of solid particles, imparting drag forces that counteract the gravitational force.

The Physics of Fluidization II

Key Points

- At low gas velocities, the bed remains static (fixed bed) and pressure drop increases linearly with superficial velocity (V_0).
- Minimum Fluidization Velocity (V_{mf}) is reached when the drag force of upward-flowing gas equals the buoyant weight of the particles: $\Delta P = H(1 - \epsilon_0)(\rho_s - \rho_g)g$.
- Beyond V_{mf}, the bed behaves like a boiling liquid, exhibiting high heat transfer rates (200-450 W/m$\hat{2}$K) and uniform temperature distribution.
- If the superficial velocity exceeds the terminal velocity (V_t) of the particles, entrainment occurs, which is the foundational principle of Circulating Fluidized Beds (CFBC).

Fluidization Regimes & Pressure Drop Profile I



Fluidization Regimes & Pressure Drop Profile II

Key Points

- Fixed Bed Regime: Pressure drop increases linearly with velocity according to the Ergun Equation.
- Minimum Fluidization Velocity (V_{mf}): The point where the pressure drop becomes independent of velocity and equals the effective weight of the bed.
- Fluidized Bed Regime: Characterized by a constant pressure drop and intense solids mixing.
- Pneumatic Transport: Beyond terminal velocity (V_t), particles are carried out of the bed, requiring cyclonic separation and recirculation.

FBC Classification: Bubbling vs. Circulating Beds I

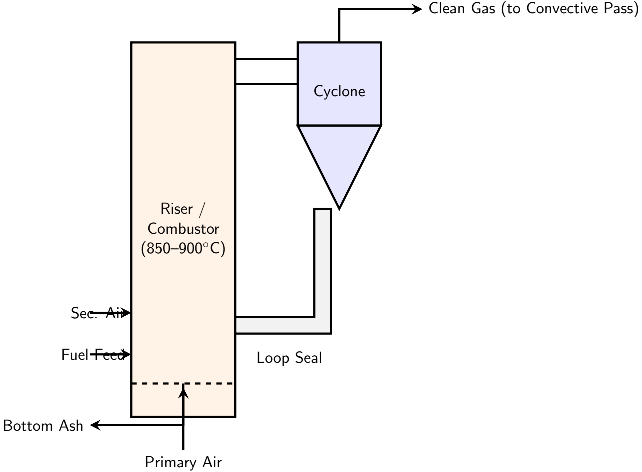
FBC boilers are classified based on their operating velocity and particle recycling mechanisms into Atmospheric Bubbling (AFBC), Pressurized Bubbling (PFBC), and Circulating Fluidized Bed (CFBC) boilers.

FBC Classification: Bubbling vs. Circulating Beds II

Key Points

- Bubbling FBC (BFBC) operates at low velocities (1 to 3 m/s) with a distinct bed surface and is suitable for small to medium scale applications (up to 150 MW).
- Circulating FBC (CFBC) operates at high velocities (4 to 10 m/s) with no distinct bed interface; particles are constantly entrained, captured by cyclones, and returned to the furnace.
- Pressurized FBC (PFBC) operates at pressures of 1 to 2 MPa, enabling combined cycle operations by running combustion gases directly through a gas turbine.
- CFBC is highly favored for large scale plants due to higher combustion efficiency (98-99%), fuel flexibility, and lower NOx emissions.

Circulating Fluidized Bed Combustion (CFBC) Layout I



Circulating Fluidized Bed Combustion (CFBC) Layout II

Key Points

- Riser / Combustor: The primary reaction chamber where fuel is burned in a suspended state at 800-900 °C.
- Air Distribution: Primary air fluidizes the bed via a nozzle distributor plate; secondary air is injected higher up to ensure staged, complete combustion.
- Cyclone Separator: Centrifugally separates unburned carbon and bed materials from the hot flue gas with ¿99% efficiency.
- Loop Seal (L-valve): Returns captured solids to the furnace bottom, maintaining a high solid concentration profile and preventing gas bypass.

In-situ SOx and NOx Emission Control I

One of the defining advantages of FBC boilers is the ability to capture sulfur dioxide (SO_2) directly within the combustion zone and limit thermal NO_x formation.

In-situ SOx and NOx Emission Control II

Key Points

- Limestone ($CaCO_3$) is fed directly into the bed where it calcines at furnace temperatures: $CaCO_3 arrow CaO + CO_2$.
- The generated Calcium Oxide reacts with sulfur dioxide and oxygen to form stable calcium sulfate: $CaO + SO_2 + 0.5 O_2 arrow CaSO_4$.
- Optimal desulfurization temperature is in the range of 800 °C to 850 °C; higher temperatures cause sintering of CaO pores, reducing capture efficiency.
- Low combustion temperatures (800-900 °C) are well below the threshold for thermal NO_x formation (¿1400 °C), limiting NO_x emissions primarily to fuel-bound nitrogen.

Heat Transfer Mechanisms & Performance Calculations I

Heat transfer in FBC boilers is exceptionally high due to the combination of convective gas flow, radiative transfer, and conductive particle contact.

Heat Transfer Mechanisms & Performance Calculations II

Key Points

- Overall bed heat transfer coefficient (h) typically ranges from 200 to 450 W/m2̂K, which is 5 to 10 times higher than in conventional pulverized coal boilers.
- The high heat transfer rate allows for compact furnace designs and reduced tube surface area requirements: $Q = UA\Delta T_{lm}$.
- Heat is extracted via in-bed tubes in bubbling beds, or via membrane water walls and external fluid bed heat exchangers (FBHE) in circulating beds.
- Fluidized bed density affects the heat transfer coefficient; higher suspension densities in the lower combustor lead to higher local heat transfer rates.

Fuel Flexibility and Operational Benefits I

FBC boilers represent a key technology for burning low-grade, high-ash fuels, agricultural biomass, and industrial waste that are difficult to burn in other systems.

Fuel Flexibility and Operational Benefits II

Key Points

- High Fuel Flexibility: Can burn high-ash coals (up to 70% ash), lignite, petroleum coke, municipal solid waste, and biomass without severe slagging.
- Reduced Slagging and Fouling: Lower combustion temperature (850 °C) keeps ash below its softening temperature, preventing fusion and deposit formation.
- Elimination of Pulverizers: Fuel only needs to be crushed to a size of 1-10 mm, significantly reducing capital and auxiliary power costs compared to pulverized coal mills.
- High Turndown Ratio: Can operate efficiently down to 30-40% of rated capacity by adjusting fluidization velocity and bed height.

The Concept and Necessity of Draft I

Draft is the small pressure difference required to maintain the flow of air and combustion products through the boiler system.

Key Points

- Necessity: To supply the required amount of oxygen/air for combustion of fuel at a controlled rate.
- Resistance overcoming: Combustion gases must overcome frictional resistance in fuel bed, grates, flues, tubes, economizer, air preheater, and chimney.
- Pressure differential: ΔP = P_atm - P_furnace. Typically measured in millimeters of water gauge (mm of wg) or Pascals (Pa), ranging from 10 to 100 mm of wg.
- Disposal: To discharge the burnt flue gases into the upper atmosphere to reduce local pollution.

Classification of Boiler Draft Systems I

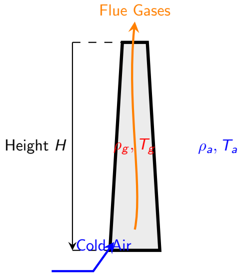
Boiler draft is broadly classified into Natural Draft (using a chimney) and Artificial Draft (using mechanical fans or steam jets).

Classification of Boiler Draft Systems II

Key Points

- Natural Draft: Produced by a chimney alone due to the density difference between hot flue gases inside the chimney and cold atmospheric air outside.
- Mechanical (Artificial) Draft: Uses centrifugal fans. Subdivided into Forced Draft (positive pressure in furnace), Induced Draft (negative pressure at exit), and Balanced Draft (combination of both).
- Steam Jet Draft: Uses high-velocity steam jets. Can be Induced Steam Jet (placed in chimney/smoke box) or Forced Steam Jet (placed in ash pit).
- Choice factors: Rate of combustion, fuel type, boiler height, thermal efficiency, and environmental regulations.

Natural Draft: Temperature and Pressure Relations I



Natural Draft: Temperature and Pressure Relations II

Key Points

- Draft Pressure Equation: h_w = 353 H ($\frac{1}{T\_a}$ - $\frac{m+1}{m}$ $\frac{1}{T\_g}$) mm of water column, where H is the chimney height in meters.
- Temperatures T_a (ambient air) and T_g (flue gas) must be in Kelvin. m is the mass of air supplied per kg of fuel.
- Density difference: Natural draft relies on ρ_a ¿ ρ_g where ρ_a $\approx$353/T_a and ρ_g $\approx$353(m+1)/(m T_g).
- Limitation: Natural draft is highly dependent on atmospheric temperature (T_a) and stack gas temperature (T_g). It cannot produce high draft (rarely exceeds 15-20 mm of wg).

Forced Draft vs. Induced Draft I

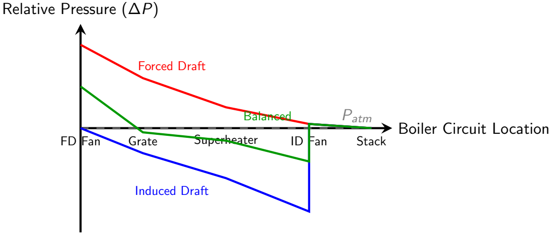
Mechanical draft systems use centrifugal fans to overcome high system resistances, allowing higher combustion rates.

Forced Draft vs. Induced Draft II

Key Points

- Forced Draft (FD) System: FD fan is located at the air inlet. It forces fresh air under positive pressure through the fuel bed and flues. Prevents air leakage into the furnace but can cause gas leakage out.
- Induced Draft (ID) System: ID fan is located at the base of the chimney. It sucks out flue gases, creating a negative pressure (sub-atmospheric) in the furnace. Causes air infiltration if there are leaks.
- Balanced Draft System: Utilizes both an FD fan (to push air through the grate) and an ID fan (to draw flue gases and maintain furnace pressure slightly below atmospheric, e.g., -2 to -5 mm of wg).
- Power Consumption: Power required by fan is proportional to the volume of gas handled. ID fans require more power than FD fans because hot flue gases have a larger specific volume.

Pressure Variation Across Draft Systems I



Pressure Variation Across Draft Systems II

Key Points

- Balanced Draft Pressure Profile: FD fan raises pressure above atmospheric to overcome grate resistance. ID fan maintains suction to keep furnace at slightly negative pressure.
- Forced Draft Profile: Pressure remains positive throughout the boiler flue paths, dropping to zero (atmospheric) only at the chimney exit.
- Induced Draft Profile: Pressure is negative (suction) throughout the system, starting from atmospheric at the grate inlet and reaching maximum suction at the ID fan inlet.
- Safety & Efficiency: Balanced draft prevents hot gas blowout (safety of forced draft) and minimizes air infiltration (efficiency of induced draft).

Chimney Height & Efficiency of Natural Draft I

The height of the chimney is determined by draft requirements and pollution dispersion guidelines, while its thermal efficiency is extremely low.

Key Points

- Chimney Height Calculation: $H = \frac{h\_d}{353(\frac{1}{T\_a} - \frac{m+1}{mT\_g})}$ where h_d is the required draft in mm of water column.
- Environmental regulation: H = 14 (Q)$\hat{0}$.3 meters (for particulate matter emissions) or H = 74 (Q_SO2)$\hat{0}$.33 meters, where Q is the emission rate in kg/hr.
- Efficiency of Chimney: η_chimney $= \frac{H \cdot g \cdot (T\_g - T\_a)}{c\_p \cdot T\_g \cdot T\_a} \cdot 100\%$. Usually less than 1% because most energy is discharged as heat.
- Heat Recovery: Artificial draft allows the addition of Economizers and Air Preheaters, which cool down the flue gases to recover heat, increasing overall boiler efficiency.

Industrial Boiler Draft Design Parameters I

Summary of typical industrial values, operational challenges (like slagging, fouling, and dew point corrosion), and fan controls.

Industrial Boiler Draft Design Parameters II

Key Points

- Typical Fan Selection: Backward-curved centrifugal fan blades for FD fans (high efficiency, self-limiting power); Radial or forward-curved for ID fans (dust handling, erosion resistance).
- Acid Dew Point Corrosion: Flue gas temperature must not drop below the acid dew point (approx 120-140°C depending on sulfur content) to prevent H_2SO_4 condensation in the ID fan and chimney.
- Fan Control Methods: Inlet guide vanes (IGVs), variable speed drives (VFDs), and damper controls to match fluctuating boiler loads.
- Key Takeaway: Mechanical draft is essential for modern, high-capacity utility boilers to achieve high heat transfer rates and meet strict emission controls.

Boiler Mountings: Safety & Operational Control Devices I

Boiler mountings are mandatory physical attachments installed directly on the boiler shell for safe operation and pressure boundary control. They must conform to strict standards like ASME Boiler and Pressure Vessel Code or Indian Boiler Regulations (IBR).

Boiler Mountings: Safety & Operational Control Devices II

Key Points

- Safety Valves: Positioned at the highest point of the steam space to automatically vent steam and prevent overpressure. Must be sized to discharge the maximum steam capacity of the boiler.
- Water Level Indicators: Installed on the front of the shell (typically two for redundancy) to allow operators to visually monitor liquid level. Low water levels lead to catastrophic thermal stress and dry firing.
- Pressure Gauge: Dial indicator connected to the steam space via a U-tube siphon (which fills with condensate to protect the gauge from hot steam) to show the operational pressure.
- Fusible Plug: Threaded into the furnace crown plate. It has a gunmetal body with a core of low-melting-point alloy that melts if water levels drop, releasing steam to extinguish the fire.

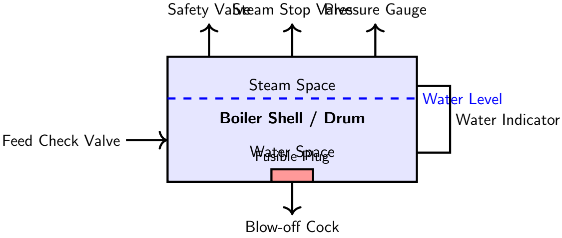
Boiler Mountings: Control and Feed Valves I

Beyond primary safety sensors, mountings include robust valve systems designed to direct high-pressure steam, introduce feedwater, and maintain clean internal surfaces by purging scaling impurities.

Key Points

- Steam Stop Valve: Regulates the steam output rate from the boiler to the main steam header. Designed as a high-temperature globe valve to provide tight shutoff and throttle control.
- Feed Check Valve: A combination of a non-return valve and a regulating valve. It prevents hot pressurized boiler water from backflowing into the feed lines if pump pressure drops.
- Blow-off Cock: Installed at the lowest point of the mud drum or shell to purge high-concentration dissolved solids, mud, and precipitated scale during routine blowdown cycles.

Schematic Diagram of Boiler Drum and Mountings I



Schematic Diagram of Boiler Drum and Mountings II

Key Points

- The diagram shows typical locations of critical safety and flow control mountings relative to the steam/water interface.
- Operational feed check valve is located in the water space, while safety valves and steam stop valves are situated at the highest points of the steam space.
- The blow-off cock is strategically located at the absolute bottom of the boiler drum to maximize scale and sediment removal.
- The fusible plug sits directly above the firebox (furnace crown) to ensure it is exposed to combustion heat if water levels fall below the plug.

Boiler Accessories: Thermodynamic Efficiency Enhancers I

Accessories are auxiliary components that improve the thermal efficiency of the steam generating plant by recovering waste heat from combustion products or pre-treating working fluid inputs.

Boiler Accessories: Thermodynamic Efficiency Enhancers II

Key Points

- Superheater: Increases steam temperature above saturation without increasing pressure, increasing Rankine cycle efficiency and avoiding condensation in the steam turbine stages.
- Economizer: Preheats incoming boiler feedwater using hot exhaust flue gases. The rule of thumb is that every 6°C rise in feedwater temperature saves approximately 1% in fuel consumption.
- Air Preheater: Extracts residual heat from flue gases to preheat combustion air. This accelerates fuel ignition, stabilizes the flame, and raises the adiabatic flame temperature in the furnace.
- Feed Pump: An auxiliary high-pressure pump (centrifugal or reciprocating) required to overcome internal steam pressure and force water continuously into the boiler drum.

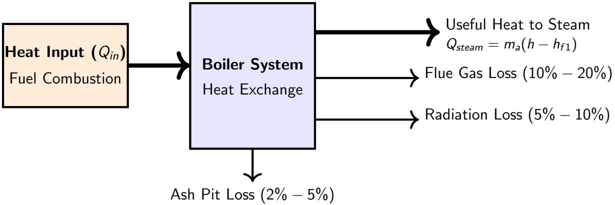
Boiler Performance: Equivalent Evaporation I

Boilers operate under highly variable steam pressures, steam temperatures, and feedwater inlet temperatures. Performance must be standardized to equivalent conditions (water evaporated from and at 100°C) to enable fair thermodynamic comparison.

Boiler Performance: Equivalent Evaporation II

Key Points

- Equivalent Evaporation (me): Defined as the amount of water evaporated from water at 100°C into dry saturated steam at 100°C per hour.
- Mathematical Model: me = [ma * (h - hf1)] / 2257 (kg/hr), where ma is the actual mass of steam generated per hour (kg/hr), h is the enthalpy of steam generated (kJ/kg), hf1 is the enthalpy of feed water (kJ/kg), and 2257 is the latent heat of vaporization of water at atmospheric pressure (kJ/kg).
- Factor of Evaporation (Fe): The ratio of heat absorbed by 1 kg of water under actual operating conditions to the latent heat of vaporization at 100°C. Fe = (h - hf1) / 2257.
- Boiler Power Rating: Often expressed in Boiler Horsepower (BHP), which represents the evaporation of 15.65 kg of water per hour from and at 100°C.



Heat Input (Q_{in})
Fuel Combustion
Boiler System
Heat Exchange
Useful Heat to Steam
$Q_{steam} = m_a(h - h_{f1})$
Flue Gas Loss (10% – 20%)
Radiation Loss (5% – 10%)
Ash Pit Loss (2% – 5%)

Boiler Heat Balance and Energy Flow Diagram II

Key Points

- The Sankey-style block diagram maps the allocation of energy from the primary fuel input to the output streams.
- Total Heat Input is $Q\_in = m\_f * CV$, where m_f is the mass flow rate of fuel and CV is the lower calorific value.
- Useful heat transfer occurs to the water and steam, accounting for 60% to 85% of total input energy in modern configurations.
- Sensible heat carried away by the dry flue gases constitutes the largest continuous energy loss, which economizers and air preheaters actively minimize.

Boiler Thermal Efficiency & Heat Balance Sheets I

A Heat Balance Sheet is an energy audit tool listing heat inputs and all paths of heat rejection. It is formatted as an energy ledger per kg of fuel burned or per unit time.

Boiler Thermal Efficiency & Heat Balance Sheets II

Key Points

- Boiler Thermal Efficiency (η): $\eta =$ [ma * (h - hf1)] / (mf * CV). This measures the percentage of fuel energy successfully converted into useful steam enthalpy.
- Flue Gas Loss Calculation: Qg $=$ mg * Cpg * (Tg - Ta), where mg is mass of dry flue gas per kg of fuel, Cpg is specific heat of flue gas, Tg is flue gas exit temperature, and Ta is boiler room ambient temperature.
- Moisture Loss Calculation: Qm $=$ mm * (h_sup - h_f), accounting for water vapor generated from fuel hydrogen combustion and raw fuel moisture content.
- Unaccounted Losses: Typically calculated by subtracting all measured outputs and known losses from the total heat input. This includes radiation, convection, and unburnt combustibles in ash.

Performance Comparison Challenges I

Boilers operate under highly varying pressure, feed water temperature, and steam quality conditions, making direct comparison of actual evaporation rates (m_a) impossible without standardizing the base of comparison.

Performance Comparison Challenges II

Key Points

- Actual evaporation rate (m_a) depends heavily on the inlet feed water temperature (t_f) and the exit steam pressure and quality.
- A boiler operating with hot feed water and producing wet steam will evaporate more water per kg of fuel than one with cold feed water producing superheated steam, even if their heat transfer effectiveness is the same.
- To compare boiler performance fairly, actual evaporation must be reduced to a standard reference state.
- Reference State: Feed water temperature of 100 $^\circ$C and steam generation at 100 $^\circ$C under standard atmospheric pressure (1.01325 bar).

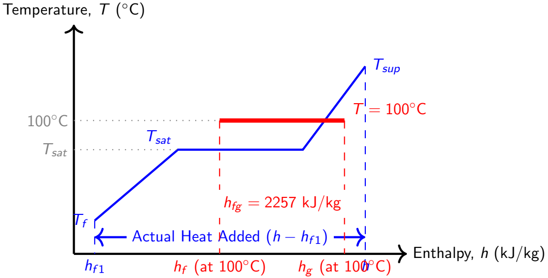
Thermodynamic Reference Base I

$$m\_e = \frac{m\_a \times (h - h\_f1)}{2257}$$

Key Points

- Here, h is the enthalpy of steam generated per kg (kJ/kg), which depends on whether the steam is wet $(h_f + xh_{fg})$, dry saturated (h_g), or superheated $(h_g + c_{ps}(T_{sup} - T_{sat}))$.
- h_{f1} is the specific enthalpy of feed water corresponding to the actual feed water temperature t_f $(h_{f1} \approx 4.187 \times t_f$ kJ/kg).
- The value 2257 kJ/kg is the latent heat of vaporization of water at standard atmospheric pressure (100 °C).
- Equivalent evaporation (m_e) represents the amount of water that would have been evaporated from and at 100 °C by the same heat energy absorbed by water/steam in the actual boiler.

Temperature-Enthalpy (T-h) Comparison I



Temperature-Enthalpy (T-h) Comparison II

Key Points

- The diagram contrasts the actual multi-phase heat absorption path (blue) against the standard isothermal reference path (red).
- Actual process: liquid heating from T_f to T_{sat}, constant pressure evaporation, and superheating to T_{sup}.
- Standard process: pure phase change at 100°C where $h_{fg} = 2257$ kJ/kg.
- Equating the heat rates: $Q = m_a(h - h_{f1}) = m_e \times 2257$, yielding the conversion equation.

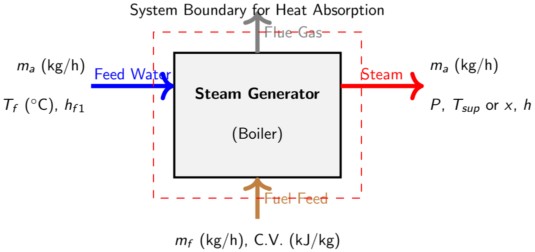
Factor of Evaporation (F_e) I

$$F\_e = \frac{h - h\_f1}{2257}$$

Key Points

- The Factor of Evaporation (F_e) is a dimensionless ratio that compares the actual heat added per kg of steam to the standard latent heat of steam.
- Mathematically, $m_e = m_a \times F_e$.
- Since $h > h_{f1}$ and usually $(h - h_{f1}) > 2257$ for modern high-pressure or superheated boilers, F_e is typically greater than 1.0 (ranging from 1.05 to 1.3).
- For low-pressure boilers or systems with very high feed water temperature and low-quality wet steam, F_e can occasionally be less than 1.0.

Boiler Energy Balance & Mass Flow I



Boiler Energy Balance & Mass Flow II

Key Points

- Mass conservation dictates that the actual steam flow rate exactly equals the feed water flow rate m_a (neglecting blowdown).
- Energy input to water/steam: $Q_a = m_a(h - h_{f1})$ kJ/h.
- The energy absorbed is normalized to the equivalent steam generation rate: $m_e = Q_a/2257$ kg/h.
- Boiler Thermal Efficiency: $\eta_{boiler} = \frac{m_a(h-h_{f1})}{m_f \times \text{C.V.}} = \frac{m_e \times 2257}{m_f \times \text{C.V.}}$, where m_f is fuel mass flow rate and C.V. is Calorific Value.

Practical Boiler Calculation Example I

A boiler generates 5000 kg/h of steam at a pressure of 10 bar and temperature of 200 °C from feed water at 40 °C. Let's calculate the Equivalent of Evaporation.

Practical Boiler Calculation Example II

Key Points

- Step 1: Identify steam state. At 10 bar, $T_{sat} = 179.9^\circ$C. Since $T_{steam} = 200^\circ$C, the steam is superheated.
- Step 2: Obtain enthalpies from Steam Tables. At 10 bar: $h_g = 2778.1$ kJ/kg, $c_{ps} \approx 2.3$ kJ/kg·K. Thus, $h = h_g + c_{ps}(T_{sup} - T_{sat}) = 2778.1 + 2.3(200 - 179.9) = 2824.3$ kJ/kg.
- Step 3: Feed water enthalpy. At 40 °C, $h_{f1} = 4.187 \times 40 = 167.5$ kJ/kg.
- Step 4: Factor of evaporation $F_e = (2824.3 - 167.5)/2257 = 1.177$.
- Step 5: Equivalent of evaporation $m_e = m_a \times F_e = 5000 \times 1.177 = 5885$ kg/h. Thus, standard evaporation capacity is 5885 kg/h.

Engineering Significance of m_e I

In commercial utility plant specifications and acceptance trials, boiler capacity is always rated and guaranteed in terms of Equivalent Evaporation 'from and at 100 °C' to ensure contract compliance.

Engineering Significance of m_e II

Key Points

- Allows direct comparison of boilers operating under vastly different conditions (e.g., waste heat recovery boiler vs high-pressure utility boiler).
- Enables calculation of specific boiler rating: Equivalent evaporation per unit of heating surface area ($kg/m^2 \cdot h$).
- Crucial for calculating the exact thermal efficiency (η_{boiler}) when testing with alternative fuels or varying feed water heating configurations.
- Acts as a standard benchmark for commercial boiler data sheets, allowing procurement engineers to perform apples-to-apples comparisons.

Boiler Power Concept & Evaporation Capacity I

Boiler power is traditionally measured by its capacity to evaporate water into steam under specified operating conditions.

Key Points

- Evaporation Capacity: The mass of steam generated per unit time, typically expressed in kg/hr or lb/hr.
- Standard comparison is difficult due to varying feed water temperatures and steam working pressures across different boiler installations.
- Boiler load varies continuously; maximum continuous rating (MCR) is the peak load a boiler can sustain safely for 24 hours.
- To standardize comparison, equivalent evaporation and factor of evaporation are utilized.

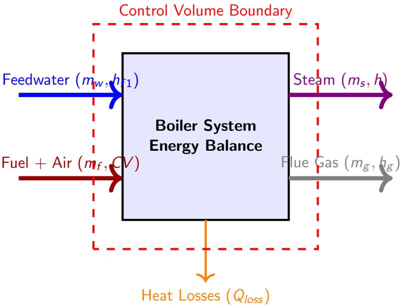
Equivalent Evaporation from and at 100°C I

Equivalent evaporation (m_e) is the quantity of water that would be evaporated from water at 100°C to dry saturated steam at 100°C per hour, using the same amount of heat.

Key Points

- Mathematical formula: $m_e = \frac{m_a \times (h - h_{f1})}{h_{fg}}$ where m_a is the actual evaporation rate (kg/hr).
- h is the specific enthalpy of steam generated (kJ/kg); h_{f1} is the specific enthalpy of feedwater at inlet temperature T_1 (kJ/kg).
- h_{fg} is the latent heat of vaporization of water at atmospheric pressure (100°C), taken as 2257 kJ/kg (or 539 kcal/kg).
- This metric removes the influence of inlet feedwater temperature and operating pressure, providing a uniform base for comparing different boilers.

Boiler Energy Balance and Flow Schematic I



Boiler Energy Balance and Flow Schematic II

Key Points

- The energy input is primarily the chemical energy of the fuel: $Q_{in} = m_f \times CV$ (Calorific Value).
- Useful heat output is absorbed by the water/steam: $Q_{useful} = m_s \times (h - h_{f1})$.
- Major energy losses include heat carried away by dry flue gases, incomplete combustion, moisture in fuel, and radiation/convection from the shell.
- A detailed energy balance (heat balance sheet) is essential to identify and mitigate thermal inefficiencies in steam plants.

Factor of Evaporation I

The Factor of Evaporation (F_e) represents the ratio of heat absorbed per kg of steam under actual conditions to the latent heat of steam at 100°C.

Key Points

- Formula: $F_e = \frac{h - h_{f1}}{2257}$, where h is steam enthalpy and h_{f1} is feed water enthalpy.
- Equivalent evaporation can be rewritten as: $m_e = m_a \times F_e$.
- Since modern boilers operate at high pressures and generate superheated steam, F_e is usually greater than 1.0, often ranging from 1.1 to 1.35.
- It measures the thermal effort required to generate steam compared to the standard reference state.

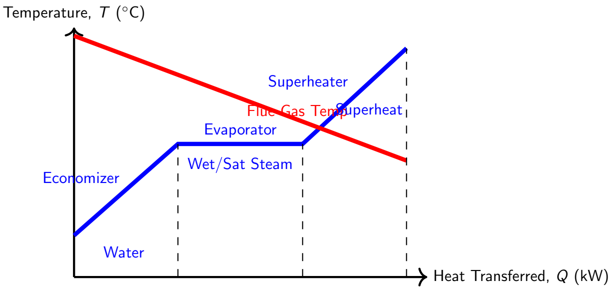
Boiler Horsepower (BHP) I

Boiler Horsepower is a traditional American unit defined as the evaporation of 34.5 pounds (15.65 kg) of water per hour from and at 212°F (100°C).

Key Points

- One BHP equals approximately 33,475 BTU/hr, 9,809.5 Watts (9.81 kW), or 8,436 kcal/hr.
- Historically, 1 BHP was the amount of steam needed to drive a one-horsepower steam engine, but engine efficiency has since improved drastically.
- Modern rating: A boiler's capacity is often rated in terms of heating surface area (e.g., 10 square feet of heating surface per BHP for fire-tube boilers).
- BHP remains widely used in industrial HVAC and process steam applications in North America to specify boiler output size.

Heat Transfer Temperature Profile (T-Q Diagram) I



Heat Transfer Temperature Profile (T-Q Diagram) II

Key Points

- Economizer: Raises the feedwater temperature to near saturation point using waste heat from flue gases.
- Evaporator (Boiler tubes): Phase change occurs at constant saturation temperature corresponding to boiler pressure.
- Superheater: Heat is added to dry saturated steam, raising its temperature above saturation to prevent condensation in turbines.
- Flue gas temperature decreases continuously as it flows counter-current or cross-current relative to the steam/water circuit.

Thermal Efficiency & Performance Metrics I

Boiler efficiency (η_{boiler}) is the ratio of heat energy utilized in steam generation to the chemical energy liberated by fuel combustion.

Key Points

- Direct Efficiency Formula: $\eta_{boiler} = \frac{m_s(h - h_{f1})}{m_f \times CV} \times 100\%$, where m_s and m_f are flow rates of steam and fuel.
- Indirect Efficiency (Heat Loss Method): Calculated by subtracting all fractional heat losses from 100% (ASME PTC 4.1 standard).
- Primary losses: Dry flue gas loss (typically 4-8%), moisture in fuel and combustion air (3-10%), and surface radiation (1-3%).
- Key efficiency improvements include installing air preheaters, economizers, blowdown heat recovery systems, and optimizing air-to-fuel ratio.

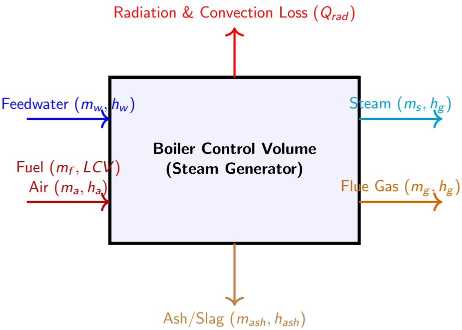
Thermodynamic Definition and Assessment Methods I

Boiler efficiency is the percentage of total heat exported by the outlet steam compared to the total heat supplied by the fuel. There are two primary standards for testing: BS 845 and ASME PTC 4.1.

Thermodynamic Definition and Assessment Methods II

Key Points

- Direct Method (Input-Output Method): Compares energy gain of working fluid (water/steam) directly to energy input of fuel.
- Indirect Method (Heat Loss Method): Computes efficiency by subtracting the sum of all individual heat losses from 100%.
- Direct Method Advantage: Simple and quick calculation using primary flow rates and temperatures.
- Indirect Method Advantage: Highlights specific areas of energy degradation, enabling target-oriented maintenance and optimization.



Radiation & Convection Loss (Q_{rad})
Feedwater (m_w, h_w)
Steam (m_s, h_g)
Boiler Control Volume
(Steam Generator)
Fuel (m_f, LCV)
Air (m_a, h_a)
Flue Gas (m_g, h_g)
Ash/Slag (m_{ash}, h_{ash})

Key Points

- The control volume establishes mass conservation: $m_{water} = m_{steam}$, and $m_{fuel} + m_{air} = m_{flue\_gas} + m_{ash}$.
- First Law of Thermodynamics dictates: Energy Input (Fuel + Air + Feedwater) = Useful Heat Output + Sum of Losses.
- Accurate boundary definitions are critical; reference temperature for ambient air is typically 25 degrees C.
- Enthalpy changes represent energy transitions: water to steam, chemical combustion to sensible heat of gas.

Direct Method: Formulation and Analysis I

The Direct Method calculates efficiency (η) by dividing the heat energy absorbed by the water/steam by the heat energy released by the combustion of fuel. Formula: $\eta = \frac{m_s \times (h - h_{f1})}{m_f \times GCV} \times 100\%$.

Direct Method: Formulation and Analysis II

Key Points

- m_s = Quantity of steam generated per hour (kg/hr).
- m_f = Quantity of fuel burnt per hour (kg/hr).
- h = Enthalpy of steam at boiler outlet pressure and temperature (kJ/kg).
- h_{f1} = Enthalpy of feed water at boiler inlet temperature (kJ/kg).
- GCV = Gross Calorific Value of the fuel (kJ/kg). If fuel contains moisture and hydrogen, Net Calorific Value (NCV) is sometimes used based on standard conventions.
- Limitations: Does not identify why the efficiency is low or which specific component (economizer, superheater, combustor) is underperforming.

Indirect Method: Detailed Heat Loss Formulations I

The Indirect Method, also known as the Heat Loss Method, evaluates individual heat losses in the boiler. Efficiency is calculated as: η= 100 - (L1 + L2 + L3 + L4 + L5 + L6 + L7 + L8), where each L represents a specific loss percentage.

Key Points

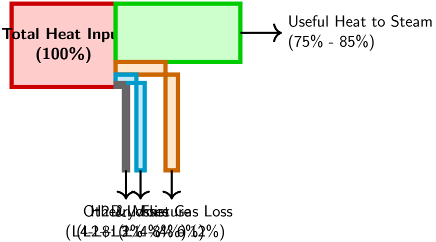
- L1: Dry flue gas loss. L1 $= [m \times C_p \times (T_f - T_a)]/GCV \times 100$. This is typically the largest loss in a boiler.
- L2: Loss due to hydrogen in fuel. L2 $= [9 \times H_2 \times (584 + C_{ps} \times (T_f - T_a))]/GCV \times 100$. Accounts for latent heat of water vapor formed.
- L3: Loss due to moisture in fuel. L3 $= [M \times (584 + C_{ps} \times (T_f - T_a))]/GCV \times 100$.
- L4: Loss due to moisture in combustion air. L4 $= [AAS \times \text{humidity} \times C_{ps} \times (T_f - T_a)]/GCV \times 100$.

Heat Loss Method (Indirect Method) - Continued I

Key Points

- L5: Loss due to incomplete combustion (CO formation). L5 $=$ $[\%CO \times C]/[\%CO + \%CO2] \times [5744/GCV] \times 100$.
- L6: Loss due to unburnt carbon in ash (fly ash and bottom ash).
- L7: Loss due to radiation and convection from the outer boiler casing.
- L8: Unaccounted losses (blowdown heat loss, soot blower steam consumption, etc.).

Sankey Diagram of Boiler Heat Losses I



Sankey Diagram of Boiler Heat Losses II

Key Points

- Sankey diagrams visually represent energy flow paths, where the width of each branch is proportional to the heat quantity.
- The primary stream goes horizontally, representing the thermodynamic efficiency (useful enthalpy transfer to water/steam).
- Loss branches divert downwards, visually emphasizing that dry flue gas loss (L1) represents the dominant thermal drain.
- This graphical energy balance sheet is a vital tool for plant engineers to instantly diagnose boiler performance and prioritize recovery options.

The Boiler Heat Balance Sheet I

A Heat Balance Sheet is an account book of heat energy supplied and heat energy distributed. It is structured on a 'per kg of fuel burnt' basis (kJ/kg of fuel) or a percentage basis.

The Boiler Heat Balance Sheet II

Key Points

- Left Column (Heat Input): Shows heat supplied by 1 kg of fuel (equal to Gross/Net Calorific Value).
- Right Column (Heat Output/Distribution): Lists useful heat absorbed by feedwater/steam and all individual heat losses.
- The sum of the heat distribution side must mathematically equal the heat input side (100% total balance).
- A typical heat balance table lists: Heat in steam (e.g. 78%), Dry flue gas loss (10%), Moisture/Hydrogen loss (5%), CO loss (1%), Ash unburnt loss (2.5%), Radiation/Unaccounted (3.5%).
- Standardized conditions: Inlet temperatures and pressure measurements must be taken during stable, steady-state boiler operation.

Engineers apply targeted thermodynamic adjustments to minimize the losses identified in the heat balance sheet.

Industrial Efficiency Optimization and Remediation II

Key Points

- Reduction of Dry Flue Gas Loss (L1): Installing/cleaning air preheaters (APH) and economizers to lower stack exhaust temperature.
- Control of Excess Air: Optimizing air-to-fuel ratio to reduce dry gas mass flow without inducing incomplete combustion (CO formation/L5).
- Soot Blowing: Periodic removal of soot deposits on heat transfer tubes to maintain high overall heat transfer coefficients (U).
- Water Treatment & Blowdown Control: Minimizing scale formation to prevent thermal resistance, and optimizing continuous blowdown with heat recovery.
- Condensing Economizers: Recovering the latent heat of vaporization of water vapor (L2/L3) in flue gas by cooling below the dew point.

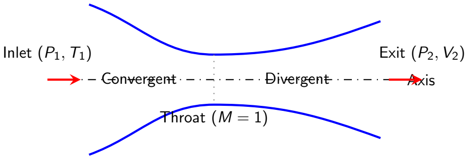
A steam nozzle is a duct or passage of varying cross-sectional area through which steam flows. During this flow, the pressure energy of steam is converted into kinetic energy. The primary function is to produce a high-velocity steam jet.

Concept of a Steam Nozzle II

Key Points

- Nozzles are adiabatic expanders where shaft work is zero ($W = 0$) and potential energy changes are negligible.
- According to the Steady Flow Energy Equation (SFEE), the enthalpy drop is converted to kinetic energy: $h_1 + V_1^2/2 = h_2 + V_2^2/2$.
- Assuming negligible inlet velocity ($V_1 \approx 0$), the exit velocity is given by $V_2 = \sqrt{2(h_1 - h_2)}$ m/s. In terms of enthalpy drop in kJ/kg, $V_2 = 44.72\sqrt{h_1 - h_2}$ m/s.
- Steam nozzles are designed to minimize flow separation, turbulence, and frictional losses to achieve maximum expansion efficiency.

Convergent-Divergent Nozzle Geometry I



Convergent-Divergent Nozzle Geometry II

Key Points

- Convergent section: Area decreases, velocity increases (subsonic flow, $M < 1$).
- Throat: Section of minimum cross-sectional area where velocity reaches sonic speed ($M = 1$).
- Divergent section: Area increases, velocity increases further to supersonic speed ($M > 1$) because steam density drops rapidly at lower pressures.
- The geometry depends on the relationship between area, velocity, and Mach number: $dA/A = (M^2 - 1)dV/V$.

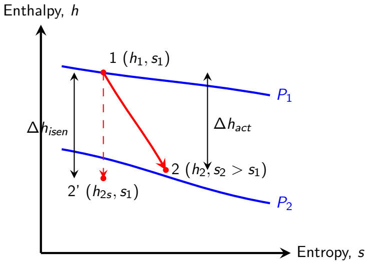
Classification of Steam Nozzles I

Steam nozzles are categorized based on their cross-sectional profile variation along the flow axis. The type of nozzle selected depends on the back pressure ratio and required exit velocity.

Key Points

- Convergent Nozzle: Cross-section area decreases continuously from inlet to exit. Used when exit pressure (back pressure) is equal to or higher than the critical pressure ($P_2 \geq P_c$).
- Divergent Nozzle: Cross-section area increases continuously from inlet to exit. Rarely used alone due to low velocity at the entry.
- Convergent-Divergent (CD) Nozzle: First converges to a throat, then diverges. Used when exit pressure is less than critical pressure ($P_2 < P_c$), generating supersonic velocities.
- Classification can also be based on flow orientation: radial flow nozzles (as in some reaction turbines) vs. axial flow nozzles.

Thermodynamic Expansion Process I



Thermodynamic Expansion Process II

Key Points

- Isentropic expansion (1 to 2'): Reversible adiabatic expansion with constant entropy ($s_1 = s_2'$). Ideal case without friction.
- Actual expansion (1 to 2): Irreversible adiabatic expansion with increasing entropy ($s_2 > s_1$) due to friction between steam molecules and nozzle walls.
- Nozzle Efficiency (Co-efficient of velocity squared): $\eta_n = (h_1 - h_2)/(h_1 - h_{2s}) = \Delta h_{act}/\Delta h_{isen}$.
- Frictional losses occur mainly in the divergent section due to high velocity and shock waves, decreasing the final kinetic energy.

Steam nozzles play a vital role in several industries, converting thermal energy of high-pressure steam into directed kinetic energy for mechanical work or fluid transport.

Industrial Applications of Steam Nozzles II

Key Points

- Steam Turbines: Directs high-velocity jets of steam onto the blades of impulse turbines (e.g., De Laval turbine) to rotate the rotor.
- Steam Jet Injectors: Used for pumping water into boilers without mechanical pumps, utilizing the kinetic energy of steam to create suction.
- Ejectors and Condensers: Creates vacuum in steam surface condensers by entraining air and non-condensable gases.
- Steam Jet Refrigeration Systems: Drives the thermocompressor to draw water vapor from the flash evaporator, creating low temperature conditions.

Critical Pressure Ratio & Choked Flow I

The critical pressure ratio (P_c/P_1) is the ratio of throat pressure (P_c) to inlet pressure (P_1) at which the discharge through the nozzle is maximum. This condition is called choked flow.

Key Points

- Mathematical derivation gives: $P_c/P_1 = (2/(n+1))^{(n/(n-1))}$, where n is the polytropic index of expansion.
- For superheated steam, $n = 1.3$, giving $P_c/P_1 \approx 0.546$. For saturated steam, $n = 1.135$, giving $P_c/P_1 \approx 0.578$. For wet steam, $n = 1.035 + 0.1x$, where x is dryness fraction.
- At critical pressure, steam velocity at the throat equals the speed of sound ($V_c = a = \sqrt{nP_c v_c}$), corresponding to Mach number $M = 1$.
- Further reduction in back pressure P_2 below P_c does not increase the mass flow rate through the nozzle.

Summary and Key Design Formulae I

Designing a steam nozzle requires balancing mass conservation, thermodynamic state relations, and geometric constraints.

Key Points

- Mass flow rate continuity: $m = A_1 V_1 / v_1 = A_t V_t / v_t = A_2 V_2 / v_2$, where v is the specific volume.
- Ideal exit velocity: $V_2 = \sqrt{2000(h_1 - h_{2s})}$ m/s (when h is in kJ/kg). Actual exit velocity: $V_{2,act} = C_v \cdot V_{2,ideal}$, where C_v is velocity coefficient.
- Throat Area calculation: $A_t = m \cdot v_c / V_c$. Requires determining specific volume v_c at throat pressure P_c.
- Effects of Supersaturation: Rapid expansion prevents steam from condensing instantly, leading to meta-stable dry saturated state, increasing mass flow rate but reducing exit velocity.

Thermodynamic Concept of a Nozzle I

A steam nozzle is a duct of varying cross-sectional area that converts the thermal energy (enthalpy) of steam into kinetic energy. The process is ideally modeled as a steady-flow, adiabatic (isentropic) expansion where $h_1 + V_1^2/2 = h_2 + V_2^2/2$.

Key Points

- Exit velocity formula derived from Steady Flow Energy Equation (SFEE): $V_2 = \sqrt{2(h_1 - h_2) + V_1^2}$. Neglecting inlet velocity V_1, $V_2 = 44.72\sqrt{\Delta h_d}$ m/s, where Δh_d is the enthalpy drop in kJ/kg.
- The primary function in steam turbines is to generate a high-velocity jet to drive the turbine blades.
- Isentropic efficiency of the nozzle: $\eta_n = (h_1 - h_2)/(h_1 - h_{2s})$, accounting for frictional losses in the boundary layer.

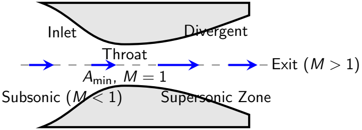
Nozzles are classified based on the variation of their cross-sectional area from inlet to exit: Convergent, Divergent, and Convergent-Divergent (CD) nozzles.

Classification of Nozzles by Area Variation II

Key Points

- Convergent Nozzle: Cross-sectional area decreases continuously from entrance to exit. Used when the back pressure is equal to or greater than the critical pressure ratio ($P_2/P_1 = 0.577$ for superheated steam, 0.582 for wet steam). Max exit velocity is sonic ($M = 1$).
- Divergent Nozzle: Cross-sectional area increases continuously from inlet to exit. Rarely used alone; requires supersonic inlet conditions to accelerate flow further.
- Convergent-Divergent Nozzle: Area first decreases to a minimum (throat) and then increases. Used to obtain supersonic exit velocities ($M > 1$) when the back pressure is lower than the critical pressure.

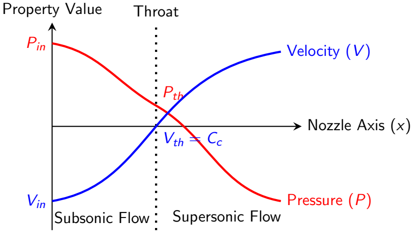
Convergent-Divergent Nozzle Geometry I



Key Points

- Inlet Section: Converging profile accelerates subsonic flow up to Mach 1 at the throat.
- Throat Section: Minimum cross-sectional area where the flow achieves sonic velocity ($M = 1$) under choked conditions.
- Divergent Section: Diverging profile expands the fluid, decreasing pressure and accelerating the flow to supersonic speeds ($M > 1$).

Pressure and Velocity Distribution I



Key Points

- Pressure drop: Continuously decreases from inlet pressure to exit pressure, driving the expansion.
- Velocity increase: Rises monotonically. The rate of velocity increase is highest in the divergent section due to rapid expansion.
- Choked Flow condition: Occurs when mass flow rate reaches its theoretical maximum, and further lowering back pressure does not increase mass flow rate.

A steam turbine is a rotary prime mover that converts the kinetic energy of expanding steam into mechanical work. The expansion of steam occurs in the nozzles (stationary blades) and the kinetic energy is absorbed by the moving blades attached to the rotor.

Key Points

- Working Principle: High-velocity steam jets impinge on curved blades, changing the steam momentum and producing a force that rotates the shaft.
- Key Thermodynamic Advantage: High thermal efficiency and power-to-weight ratio compared to reciprocating steam engines.
- Industrial Application: Centralized power generation stations (coal, nuclear, concentrated solar) and marine propulsion.

Steam turbines are classified primarily by their action of steam on the blades, direction of steam flow, and staging method.

Key Points

- By Action of Steam: Impulse Turbines (pressure drop occurs only in stationary nozzles; e.g., De Laval, Rateau, Curtis) and Reaction Turbines (pressure drop occurs in both stationary guide blades and moving blades; e.g., Parsons).
- By Flow Direction: Axial Flow (steam flows parallel to the rotor axis - most common) and Radial Flow (steam flows perpendicular to the rotor axis).
- By Exhaust Condition: Condensing Turbines (exhaust steam is condensed below atmospheric pressure) and Non-Condensing/Back-Pressure Turbines (exhaust steam is used for industrial process heating).

Comparison: Impulse vs. Reaction Principles I

Understanding the operational distinctions between Impulse and Reaction stages is fundamental to steam turbine design.

Key Points

- Impulse Turbine: Profile blades are symmetric. Velocity of steam decreases across the moving blade while pressure remains constant. High pressure drop per stage allows fewer stages.
- Reaction Turbine: Aerofoil-shaped blades forming nozzle-like passages. Steam pressure drops across both fixed and moving blades, creating a reaction force. Relative velocity increases across the moving blade.
- Compounding: To reduce high rotor speeds (up to 30,000 RPM in single-stage impulse), turbines are compounded by velocity (Curtis), pressure (Rateau), or both.

Major Components of Steam Turbines I

A steam turbine is a rotary prime mover that converts thermal energy from pressurized steam into mechanical work. Understanding its core anatomy is essential for troubleshooting structural integrity, blade erosion, and thermal expansion matching in industrial power plants.

Major Components of Steam Turbines II

Key Points

- Casing/Shell: The pressure vessel containing the rotor and blades, typically split horizontally for ease of maintenance and cast from alloy steels (like Cr-Mo-V) to withstand high temperatures and pressures without creep.
- Rotor: The central shaft that transmits torque to the generator, machined from a single forging of high-strength alloy steel (e.g., Ni-Cr-Mo) to resist centrifugal forces.
- Blades (Vanes): Aerodynamic profiles divided into fixed blades (stationary, redirect steam) and moving blades (attached to rotor, extract work); requires high corrosion and erosion resistance (e.g., 12% chromium stainless steel).

Key Points

- Nozzles & Diaphragms: Stationary elements that expand steam to increase its velocity. In impulse turbines, nozzles are mounted in diaphragms held by the casing.
- Shaft Gland Seals: Carbon ring or labyrinth seals designed to prevent high-pressure steam leakage out of the casing or air ingress into the low-pressure condenser section.
- Bearings: Journal bearings (typically fluid-film tilting pad type) support rotor weight and dynamic loads; thrust bearings (like Kingsbury type) absorb axial thrust from steam pressure differentials.

Working Principle of Impulse Turbines I

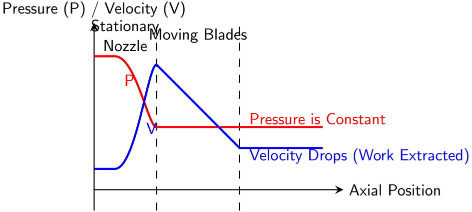
In an impulse turbine (e.g., De Laval turbine), steam expands entirely within the stationary nozzles. The resulting high-velocity steam jet strikes the moving blades, changing momentum and exerting an impulsive force to rotate the wheel.

Working Principle of Impulse Turbines II

Key Points

- Nozzle Expansion: Enthalpy drop (dh) is completely converted to kinetic energy in stationary nozzles. Theoretical exit velocity is $V_1 = \sqrt{2 \cdot (h_1 - h_2)}$ where enthalpy is in J/kg.
- Constant Pressure Across Blades: Since no expansion occurs within the moving blade channels, steam pressure (P) remains constant from the inlet to the outlet of the rotor blade.
- Symmetrical Blade Design: The blades are typically symmetrical (inlet angle β_1 equals outlet angle β_2) and have constant passage area to prevent pressure drop.
- Kinetic Energy Transfer: The steam velocity relative to the blade (V_r) remains constant (neglecting friction) or decreases slightly due to blade friction ($V_{r2} = k \cdot V_{r1}$ where $k < 1$).
- Force and Work: The tangential force on the blades is given by $F_x = \dot{m} \cdot (V_{w1} + V_{w2})$, where $\dot{m}$ is mass flow rate and V_w represents whirl velocities.

Impulse Stage Pressure-Velocity Profile I



Impulse Stage Pressure-Velocity Profile II

Key Points

- The diagram shows the axial variation of pressure (red curve) and absolute velocity (blue curve) across a single-stage impulse turbine.
- Pressure Drop: Occurs strictly within the nozzle stage. The blade passage serves only to deflect the steam flow, meaning pressure remains constant across the moving blades.
- Velocity Peak: Absolute velocity rises to its maximum at the nozzle exit, converting thermal energy to kinetic energy.
- Kinetic Energy Extraction: In the moving blade channel, the absolute velocity drops as mechanical work is extracted, transferring kinetic energy to the rotor.
- Optimal Blade Speed: Blade efficiency is maximized when the blade speed $U \approx V_1 \cdot \cos(\alpha_1)/2$.

Working Principle of Reaction Turbines I

Unlike impulse turbines, reaction turbines (such as Parsons turbines) feature continuous steam expansion across both the stationary (fixed) blades and the moving blades.

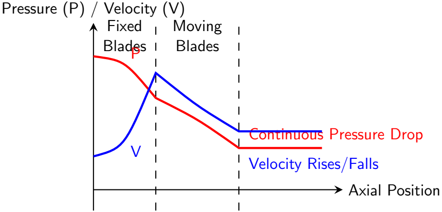
Key Points

- Nozzle Effect in Blades: The moving blades themselves have convergent channels that act as nozzles, causing pressure to drop and steam to expand while passing through them.
- Reaction Force: The acceleration of steam relative to the moving blades creates a reaction force (based on Newton's third law) which, combined with the impulse force, drives the rotor.
- Degree of Reaction (R): Defined as the ratio of enthalpy drop in the moving blades to the total enthalpy drop in the stage: $R = \frac{\Delta h_{moving}}{\Delta h_{stage}}$. For a symmetric Parsons stage, $R = 0.5$ (50% reaction).

Key Points

- Blade Profiles: Reaction blades are aerodynamically shaped like aerofoils, maintaining a pressure differential across the blade thickness to lift and rotate the rotor.
- High Volume Flows: Reaction turbines operate at lower velocities and pressure drops per stage compared to impulse turbines, requiring more stages for the same overall pressure ratio.

Reaction Stage Pressure-Velocity Profile I



Reaction Stage Pressure-Velocity Profile II

Key Points

- The diagram illustrates the continuous expansion and pressure drop across both fixed and moving blades of a reaction stage.
- Pressure Drop: Unlike the impulse stage, pressure drops steadily through both the stationary and moving blade rows, indicating work is done by reaction.
- Velocity Profile: Velocity increases in the fixed blades (acting as nozzles) and then decreases in the moving blades as mechanical energy is extracted, despite expansion.
- Relative Velocity Increase: In the moving blade, although absolute velocity drops, the relative velocity of the steam increases due to the nozzle-like convergence.
- Optimal Blade Speed: Peak efficiency occurs at a higher blade speed-to-steam speed ratio than impulse stages, typically $U/V_1 \approx \cos(\alpha_1)$.

Necessity of Compounding I

Compounding is the process of extracting energy from steam in multiple stages rather than a single wheel. Without compounding, single-stage turbines would face catastrophic mechanical and thermodynamic limits.

Necessity of Compounding II

Key Points

- Excessive Rotor Speeds: Expanding steam from 15 bar (boiler pressure) to 0.05 bar (condenser pressure) yields a steam velocity (V_1) of over 1200 m/s. A single-stage impulse wheel requires a blade velocity of $U \approx V_1/2 \approx 600$ m/s.
- Centrifugal Stress Limit: Rotor centrifugal stress is proportional to the square of blade speed ($\sigma \propto \rho \cdot U^2$). At 600 m/s, stresses exceed the yield strength of modern structural steels, leading to blade detachment.
- Extreme Rotational Speeds: For a typical rotor diameter of 1m, $U = 600$ m/s corresponds to a rotational speed of over 11,000 RPM. This is impractical for directly driving standard 50Hz/60Hz AC generators (3000/3600 RPM).

Necessity of Compounding (Cont.) I

Key Points

- High Leaving Loss: A single-stage turbine would discharge steam at extremely high residual velocity, leading to enormous kinetic energy losses (leaving losses) and low thermal efficiency.
- Blade Erosion: High steam velocities generate water droplets that cause severe impingement erosion on the low-pressure stage blades.

Types of Compounding I

To reduce the rotor speed to practical limits and maximize stage efficiency, steam turbines utilize different compounding methods: velocity, pressure, or a combination of both.

Key Points

- Velocity Compounding (Curtis Turbine): Steam expands completely in a set of nozzles, obtaining a very high velocity. This kinetic energy is then absorbed in stages by passing through alternating rows of moving blades and stationary guide blades without further pressure drops.
- Pressure Compounding (Rateau Turbine): The total pressure drop from boiler to condenser is divided into several smaller pressure drops across successive stages. Each stage has its own nozzle diaphragm and moving blade wheel, maintaining moderate velocities throughout.

Types of Compounding (Cont.) I

Key Points

- Pressure-Velocity Compounding: A hybrid system where the turbine is divided into multiple pressure stages (like Rateau), and each pressure stage is velocity compounded (like Curtis) with multiple moving blade rows. This optimizes axial length and efficiency.
- Comparison: Curtis stages are compact and cheap but have higher friction losses; Rateau stages are more efficient but require a longer casing, more diaphragms, and complex shaft sealing.

The thermodynamic efficiency of the Rankine cycle is directly enhanced by lowering the turbine exhaust pressure. Operating the condenser under sub-atmospheric pressure maximizes the turbine enthalpy drop.

Thermodynamic Benefits of Condensation II

Key Points

- According to the Carnot limit, reducing the temperature of heat rejection (T_cold) is the most effective way to increase thermal efficiency.
- Lowering condenser pressure from 1.0 bar (atmospheric) to 0.06 bar increases the enthalpy drop across the turbine, boosting power output by 25% to 30%.
- At 0.06 bar, the saturation temperature of water is roughly 36.2 degrees Celsius, enabling heat rejection to ambient water sources.
- Condensation reduces the specific volume of steam by a factor of over 20,000, which drastically lowers the power required for the condensate extraction pump.

Classification of Condensers I

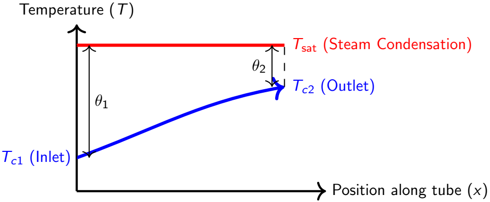
Condensers are broadly classified into direct-contact (jet) and non-contact (surface) condensers based on the mechanism of heat exchange between steam and cooling water.

Classification of Condensers II

Key Points

- Jet Condensers: Exhaust steam and cooling water mix directly. They feature high heat transfer rates, simple designs, and lower initial capital costs.
- Limitation of Jet Condensers: The condensate is contaminated by the cooling water, making it unsuitable for direct reuse in high-pressure boilers without extensive treatment.
- Surface Condensers: Steam and cooling water are separated by a physical boundary (usually brass or cupro-nickel tubes).
- Surface condensers allow the recovery of pure condensate (distilled water) for boiler feedwater, which is essential for preventing boiler tube scaling.

Temperature Profile in a Surface Condenser I



Temperature Profile in a Surface Condenser II

Key Points

- Logarithmic Mean Temperature Difference (LMTD) governs the heat transfer rate: $LMTD = (\theta_1 - \theta_2)/\ln(\theta_1/\theta_2)$.
- Here, $\theta_1 = T_{\text{sat}} - T_{c1}$ (temperature difference at inlet) and $\theta_2 = T_{\text{sat}} - T_{c2}$ (difference at outlet).
- The steam temperature remains constant at T_{sat} as it undergoes phase change, while the cooling water temperature rises non-linearly.
- Total heat transfer is computed as $Q = U \cdot A \cdot LMTD$, where U is the overall heat transfer coefficient and A is the total surface area.

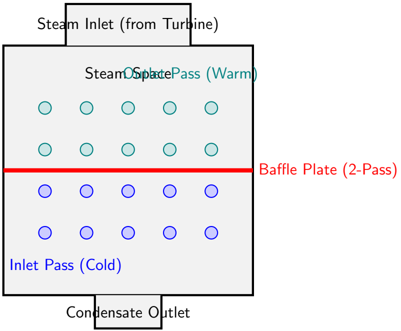
Air Inleakage and Its Detrimental Effects I

Since the condenser operates at sub-atmospheric pressure, atmospheric air continuously leaks into the shell. Additionally, non-condensable gases enter via the exhaust steam from feedwater treatment.

Key Points

- According to Dalton's Law of Partial Pressures, the total pressure is the sum of steam pressure and air pressure: $P_{\text{total}} = P_{\text{steam}} + P_{\text{air}}$.
- Air accumulation increases the backpressure in the condenser, which directly reduces turbine thermal efficiency and work output.
- Air forms a stagnant insulating layer around the cooling tubes, drastically reducing the overall heat transfer coefficient (U).
- Dissolved oxygen in the condensate promotes corrosive reactions in the condensate piping, feed pumps, and boiler tubes.

Cross-Section of a Two-Pass Surface Condenser I



Cross-Section of a Two-Pass Surface Condenser II

Key Points

- A horizontal baffle plate divides the water box, forcing cooling water to complete a double pass through the tube bundle.
- Cooling water enters the lower bank of tubes (blue), flows to the far water box, reverses direction, and exits via the upper bank (teal).
- The exhaust steam flows downwards over the outer surfaces of the tubes, condensing and draining into the hotwell.
- The two-pass arrangement increases velocity and turbulence of cooling water, enhancing the convective heat transfer coefficient.

Condenser Performance Parameters I

Condenser performance is evaluated using vacuum efficiency and condenser efficiency, assessing thermodynamic deviation from the ideal state.

Key Points

- Vacuum Efficiency $= \frac{\text{Actual Vacuum}}{\text{Ideal Vacuum}} = \frac{P_{\text{atm}} - P_{\text{actual}}}{P_{\text{atm}} - P_{\text{sat}}}$, reflecting the effectiveness of air extraction systems.
- Ideal vacuum is the saturation pressure of steam corresponding to the condensate temperature in the hotwell.
- Condenser Efficiency $= \frac{\text{Actual Rise in Cooling Water Temp}}{\text{Ideal Rise}} = \frac{T_{c2} - T_{c1}}{T_{\text{sat}} - T_{c1}}$, assessing the thermal effectiveness of the heat exchanger.
- High air leakage or scale formation inside the tubes significantly degrades both vacuum and condenser efficiencies.

Mass and Energy Balance Equations I

Designing a condenser requires sizing the cooling water system to absorb the latent heat rejected by the steam. A control-volume energy balance is applied.

Key Points

- Energy Balance: $m_s \cdot [x \cdot h_{fg} + (h_f - h_c)] = m_w \cdot c_{p,w} \cdot (T_{c2} - T_{c1})$, assuming adiabatic outer shell boundary.
- Here, m_s is the steam mass flow rate, x is dryness fraction, h_{fg} is latent heat, and m_w is the cooling water mass flow rate.
- The cooling water ratio (m_w/m_s) typically ranges from 30 to 60 under design conditions, depending on the ambient wet-bulb temperature.
- For a standard 500 MW coal-fired unit, the required cooling water circulation can exceed 50,000 cubic meters per hour.

Concept of Steam Condensers I

A steam condenser is a closed vessel in which steam is condensed by abstracting heat using cooling water, creating a vacuum.

Key Points

- Maintains a pressure below atmospheric level (vacuum) at the exhaust of the steam turbine to maximize enthalpy drop and thermal efficiency.
- Condensation process reduces the specific volume of steam by a factor of over 20,000, creating a localized low-pressure zone.
- Converts exhaust steam back into high-purity condensate, which is recycled back to the boiler as feed water, minimizing make-up water requirements.
- Enables the thermodynamic cycle to operate at a lower sink temperature (T_2), thereby increasing the Carnot and Rankine cycle efficiency.

Primary and Secondary Functions I

The dual role of steam condensers in maximizing cycle work output and conserving working fluid quality.

Key Points

- Primary Function: To decrease the turbine backpressure to sub-atmospheric levels (typically 0.04 to 0.1 bar absolute pressure).
- Thermodynamic Benefit: Lowering backpressure from 1 bar to 0.08 bar increases the available turbine enthalpy drop by approximately 20-30%.
- Secondary Function: To recover high-purity condensate, avoiding the thermal stresses and scale formation associated with raw untreated feed water.
- Deaeration: Acts as a deaerator to remove non-condensable gases (O_2, CO_2) that cause boiler corrosion and impair heat transfer.

Classification of Condensers I

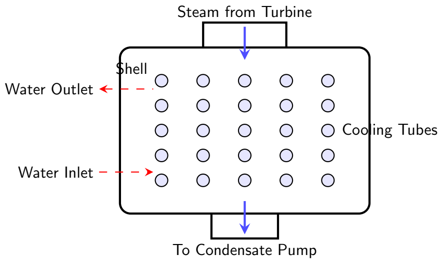
Categorization based on the mechanism of heat transfer between the exhaust steam and cooling medium.

Classification of Condensers II

Key Points

- Jet Condensers (Direct Contact): Cooling water is sprayed directly into the exhaust steam, causing rapid condensation through direct mixing.
- Surface Condensers (Indirect Contact): Steam and cooling water are physically separated by a thermal barrier (typically tube walls), preventing contamination.
- Jet Condensers classification: Parallel-flow (steam and water flow in the same direction), Counter-flow or Low-level (opposite directions), High-level or Barometric (uses a barometric leg of 10.3m).
- Surface Condensers classification: Down-flow (steam enters at top), Central-flow (steam flows radially towards center), Regenerative (steam reheats condensate), Evaporative (uses evaporative cooling of water film).

Schematic of a Two-Pass Surface Condenser I



Schematic of a Two-Pass Surface Condenser II

Key Points

- Shows the typical two-pass configuration where cooling water enters the bottom tubes, reverses direction in the water box, and exits from top tubes.
- Steam enters from the top inlet flange under vacuum and flows downwards over the tube bundle.
- Heat transfer occurs across the copper-alloy or titanium tube walls, maintaining separate paths for condensate and cooling water.
- Baffles are strategically placed to direct steam flow and optimize the overall heat transfer coefficient (U).

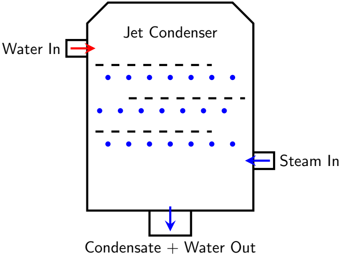
Condenser Performance Calculations I

Evaluation of condenser efficiency, vacuum efficiency, and cooling water requirements.

Key Points

- Vacuum Efficiency: $\eta_{vac} = \frac{\text{Actual Vacuum}}{\text{Ideal Vacuum}} = \frac{P_{atm} - P_{cond}}{P_{atm} - P_{sat}}$, where P_{sat} is the saturation pressure corresponding to the temperature of condensate.
- Condenser Efficiency: $\eta_{cond} = \frac{T_{out} - T_{in}}{T_{sat} - T_{in}}$, measuring the actual temperature rise of cooling water relative to maximum possible rise.
- Cooling Water Requirement: $m_w = \frac{m_s(x_s h_{fg} + c_{pc}(T_{sat} - T_c))}{c_{pw}(T_{out} - T_{in})}$, derived from energy balance between steam and water.
- Logarithmic Mean Temperature Difference (LMTD): $\theta_m = \frac{\theta_1 - \theta_2}{\ln(\theta_1/\theta_2)}$, where $\theta_1 = T_{sat} - T_{in}$ and $\theta_2 = T_{sat} - T_{out}$.

Counter-Flow Low-Level Jet Condenser I



Counter-Flow Low-Level Jet Condenser II

Key Points

- Illustrates direct contact heat transfer where cooling water is sprayed into the steam path.
- Steam enters from the lower section and flows upwards, while cooling water enters at the top and falls downward through perforated trays.
- Ensures intimate mixing, yielding a very high thermal efficiency and small size compared to surface types.
- Requires high auxiliary power for pumping out the mixture of condensate and cooling water against the vacuum pressure.

Engineering guidelines for selecting the optimal condenser type for power plants and process industries.

Comparison & Industrial Selection Criteria II

Key Points

- Surface Condensers are selected for high-capacity power plants because they prevent mixing of cooling water (which may be sea or river water) with pure condensate.
- Jet Condensers are chosen for low-capacity plants or process applications where water purity is not critical and space/capital costs are limited.
- Vacuum levels in surface condensers are higher (up to 95-98% vacuum efficiency) compared to jet condensers due to better air separation and cooling design.
- Maintenance: Surface condensers suffer from tube fouling, scaling, and tube leakage, requiring regular chemical cleaning and eddy current testing.

Steam Condensers: Overview and Working Principle I

A steam condenser is a closed vessel in which exhaust steam from a turbine or engine is condensed by cooling water, maintaining a vacuum to maximize turbine enthalpy drop.

Key Points

- Primary function is to maintain a low back pressure (vacuum) on the exhaust side of the turbine, thereby increasing thermodynamic efficiency.
- Secondary function is to supply pure condensate as hot feed-water back to the steam boiler, reducing makeup water requirements.
- Classified broadly into Jet Condensers (Direct Contact type) and Surface Condensers (Non-Contact / Heat Exchanger type).
- Heat balance equation: Heat rejected by steam is equal to heat absorbed by cooling water: $m_s * (h_s - h_c) = m_w * c_p * (T_{w2} - T_{w1})$.

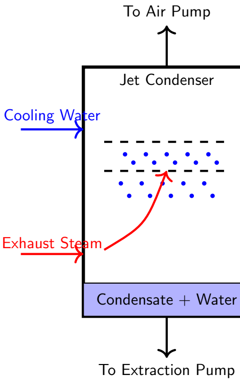
Low Level Counter-Flow Jet Condenser I

In low-level counter-flow jet condensers, the condenser shell is placed at a low level. Steam flows upwards while cooling water flows downwards.

Key Points

- Cooling water enters near the top of the shell and is sprayed through perforated trays or nozzles to create fine droplets.
- Exhaust steam enters from the bottom of the shell and travels upwards, coming into direct contact with the falling water.
- Condensation occurs rapidly due to direct contact, and the mixture of condensate and cooling water collects at the bottom.
- A condensate extraction pump is required to discharge the mixture from the condenser shell into the hotwell.

Schematic Diagram of a Counter-Flow Jet Condenser I



Schematic Diagram of a Counter-Flow Jet Condenser II

Key Points

- The schematic represents a Low-Level Counter-Flow Jet Condenser showing the counter-current path of steam and cooling water.
- Water sprays downwards through perforated trays to maximize the contact surface area and heat transfer coefficient.
- Steam enters from the lower side and travels upwards, transferring its latent heat of condensation directly to the water droplets.
- Air extraction is positioned at the top (coldest region) to reduce the volume of air to be handled by the vacuum pump.

High-Level (Barometric) and Ejector Jet Condensers I

High-level jet condensers use a barometric tail pipe to drain condensate by gravity, eliminating the need for a condensate extraction pump.

Key Points

- The condenser shell is mounted at a height of about 10.36 meters (barometric head) above the hotwell.
- The water column inside the tail pipe balances atmospheric pressure, allowing automatic gravity drainage.
- In Ejector Jet Condensers, cooling water under a high head (approx. 5-6 meters) is discharged through converging nozzles, creating a high-velocity jet.
- The high-velocity jet creates a localized vacuum that draws in exhaust steam, mixing and condensing it without requiring an external air pump.

Construction and Working of Surface Condensers I

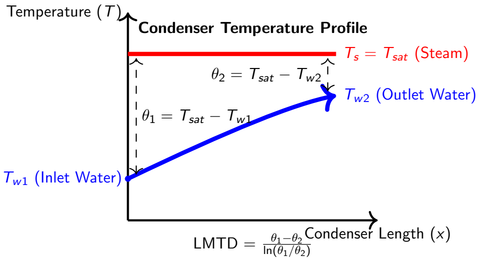
Surface Condensers are shell-and-tube heat exchangers where cooling water flows inside the tubes and exhaust steam flows over the outer surfaces of the tubes.

Construction and Working of Surface Condensers II

Key Points

- Consists of a horizontal cylindrical shell fitted with tube plates at both ends, containing thousands of brass or Cu-Ni tubes.
- Baffles are installed inside the shell to guide the steam flow, ensure uniform distribution, and prevent direct steam impingement on tubes.
- Typically designed as two-pass or multi-pass: water enters the lower half of the tubes, travels to the other end, and returns through the upper half.
- Maintains high condensate purity since the condensate does not mix with cooling water, making it ideal for high-pressure marine and utility boilers.

Heat Transfer Temperature Profiles in Steam Condensers I



Heat Transfer Temperature Profiles in Steam Condensers II

Key Points

- The diagram illustrates the temperature profile inside a steam condenser where steam temperature (T_{sat}) remains constant.
- Cooling water temperature increases from inlet T_{w1} to outlet T_{w2} as it absorbs latent heat from the condensing steam.
- Logarithmic Mean Temperature Difference (LMTD) is used for sizing the heat exchanger area: $Q = U \cdot A \cdot \text{LMTD}$.
- The constant steam temperature simplifies heat transfer analysis and makes counter-flow and parallel-flow thermal performance theoretically identical.

Comparative Performance and Industrial Applications I

Selecting between jet and surface condensers involves trade-offs in water quality, space, capital cost, and thermodynamic efficiency.

Comparative Performance and Industrial Applications II

Key Points

- Cooling water requirement: Jet condensers require less cooling water because of direct contact, while surface condensers require 20-30% more.
- Condensate quality: Surface condensers produce pure condensate suitable for high-pressure boilers; jet condensers mix condensate with cooling water, requiring treatment.
- Vacuum efficiency: Surface condensers achieve higher vacuum (up to 73.5 cm of Hg) compared to jet condensers, improving turbine thermal efficiency.
- Capital & Maintenance: Jet condensers are cheaper, simpler, and require less space, whereas surface condensers are expensive and prone to tube scaling/corrosion.

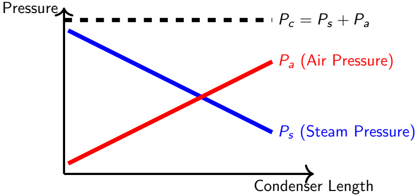
Sources of Air Inleakage in Steam Condensers I

Air enters the condenser vacuum system through multiple sources under sub-atmospheric pressure operations.

Key Points

- Inleakage at flange joints, shaft glands of low-pressure turbines, and valve spindles due to pressure differential.
- Dissolved air in the boiler feed-water (typically 10-30 ppm) which is carried over with the steam and released upon condensation.
- Air entering through auxiliary connections, vacuum breaker valves, and instrument fittings under high vacuum (0.05 to 0.1 bar a).
- The rate of air leakage is typically specified in kg/hr; for a medium-sized power plant, it is kept below 5 to 10 kg/hr via continuous extraction.

Dalton's Law of Partial Pressures & Pressure Distribution I



Dalton's Law of Partial Pressures & Pressure Distribution II

Key Points

- According to Dalton's Law of Partial Pressures, the total pressure inside the condenser is the sum of partial pressures of steam (P_s) and air (P_a): $P_c = P_s + P_a$.
- As steam condenses along the flow path, the steam partial pressure decreases as steam density decreases, causing the partial pressure of air to rise significantly near the air cooler section.
- Air accumulation raises the total condenser pressure, which increases the turbine backpressure, thereby reducing the enthalpy drop across the turbine and decreasing overall plant thermal efficiency.
- For every 0.01 bar increase in condenser backpressure, the turbine output can decrease by approximately 0.5% to 1.5%.

Thermodynamic Consequences of Air Accumulation I

Air is a non-condensable gas that severely impairs the heat transfer characteristics and thermodynamics of the steam condenser.

Key Points

- Reduction in Heat Transfer Coefficient: Air forms a stagnant film around the cold condenser tubes. Since air has a very low thermal conductivity (≈ 0.026 W/m·K), this boundary layer acts as an insulator, drastically reducing the overall heat transfer coefficient (U).
- Lowering of Condensation Temperature: Since $P_s = P_c - P_a$, the partial pressure of steam is lower than the condenser pressure. Consequently, the saturation temperature of steam (T_s) drops below the saturation temperature corresponding to P_c, reducing the temperature difference (ΔT) driving heat transfer.

Thermodynamic and Heat Transfer Penalties (Cont.) I

Key Points

- Increased Under-cooling of Condensate: The condensate is cooled below the saturation temperature corresponding to the total pressure. This thermal under-cooling represents a direct loss, as more heat must be supplied in the boiler to raise the feed-water to saturation temperature.
- Increased Power Consumption: The air extraction pump (steam jet ejector or liquid ring vacuum pump) must work harder to remove the increased mass of air, leading to higher auxiliary power consumption.

Beyond thermal performance degradation, air leakage introduces chemical and mechanical risks to the entire steam-water cycle.

Corrosion, Degradation, and Mechanical Integrity II

Key Points

- Oxygen Corrosion (Oxidation): The leaked air introduces oxygen into the feed-water. Dissolved oxygen causes pitting corrosion in boiler tubes, feed-water heaters, and economizers at high temperatures.
- Carbon Dioxide Corrosion: Air also contains CO_2, which dissolves in the condensate to form carbonic acid (H_2CO_3), leading to acid attack on steel piping and condenser shells.
- Cavitation in Condensate Extraction Pumps: Non-condensables can form bubbles in the suction line of the condensate extraction pump (CEP), leading to cavitation, impeller erosion, and flow instability.
- Vibration and Mechanical Stresses: Uneven distribution of non-condensable gases causes localized cooling variations, leading to thermal stresses and vibration in the condenser tube bundle.

Mass and Volumetric Calculations of Leakage I

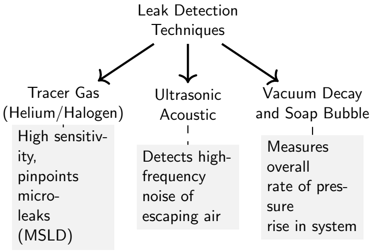
Thermodynamic equations govern the behavior of the air-steam mixture and determine the load on the air extraction pump.

Mass and Volumetric Calculations of Leakage II

Key Points

- Mass ratio of air to steam:
  $m_a/m_s = (P_a \cdot V_a/(R_a \cdot T))/(P_s \cdot V_s/(R_s \cdot T)) = (P_a/P_s) \cdot (R_s/R_a) = 0.622 \cdot (P_a/P_s)$, where $R_s = 461.5$ J/kg·K and $R_a = 287$ J/kg·K.
- Total mass of mixture extracted per hour:
  $m_{mix} = m_a + m_s = m_a \cdot (1 + P_s/(0.622 \cdot P_a))$. This demonstrates that removing a unit mass of air inevitably extracts a significant amount of associated water vapor.
- The volume of mixture to be handled by the air pump is:
  $V = (m_a \cdot R_a \cdot T)/P_a$. Lowering the extraction temperature T reduces both V and the amount of steam carried away with the air.
- This is why condensers feature an 'air cooler section' near the air pump suction, where the mixture is cooled below the main condensation temperature to minimize steam carryover.

Modern and Classic Condenser Leak Detection Methods I



Modern and Classic Condenser Leak Detection Methods II

Key Points

- Vacuum Decay Test (Drop Test): Isolating the condenser under vacuum and measuring the rate of pressure rise over time. A rise of more than 1-2 mbar/min indicates significant leakage.
- Helium Mass Spectrometer Leak Detection (MSLD): The gold standard. Helium tracer gas is sprayed around potential leak sites, and a mass spectrometer connected to the air ejector outlet detects helium infiltration down to 10^{-9} mbar $\cdot$ L/s.
- Ultrasonic Leak Detection: Handheld sensors detect the high-frequency acoustic emissions (typically 38-42 kHz) generated by turbulent air flow through micro-orifices under vacuum.
- Foam/Soap Bubble Test: Applying a foaming agent to suspected leak paths (while the system is pressurized during maintenance) and watching for bubble formation.

Operational Best Practices and Mitigation I

Maintaining condenser vacuum through prevention and scheduled maintenance is vital for thermodynamic efficiency.

Operational Best Practices and Mitigation II

Key Points

- Implement routine vacuum drop tests (weekly/monthly) to establish baseline leakage rates and detect progressive deterioration.
- Ensure periodic inspection and replacement of turbine gland steam sealing systems (keeping steam pressure slightly above atmospheric at 0.1-0.2 bar g).
- Use high-efficiency steam jet air ejectors (SJAE) or mechanical liquid ring vacuum pumps (LRVP) to maintain low non-condensable partial pressure.
- Correlate cooling water inlet/outlet temperatures with condenser pressure to distinguish between cooling water fouling and air leakage issues.

Thermodynamics of Cooling Towers I

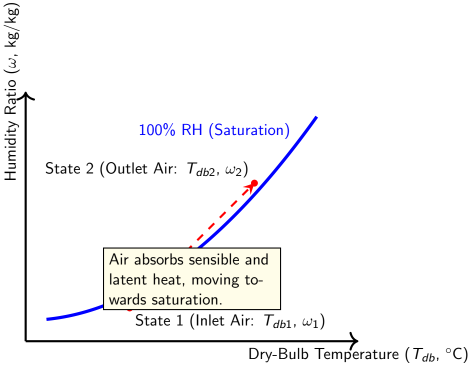
Cooling towers reject waste heat to the atmosphere through the evaporative cooling of a fraction of the circulating water. This process relies on both sensible heat transfer due to temperature differences and latent heat transfer due to mass transfer (evaporation).

Thermodynamics of Cooling Towers II

Key Points

- Energy balance is governed by: m_w * c_pw * (T_in - T_out) = m_a * (h_out - h_in), where m_w is water flow rate, m_a is dry air flow rate, and h is air enthalpy.
- Latent heat transfer accounts for approximately 75% to 90% of the total heat rejection under typical operating conditions.
- The driving force for heat transfer is the enthalpy difference between the saturated air at the water temperature and the bulk air enthalpy: dh = h_s - h_a.
- The limit of cooling is theoretically dictated by the wet-bulb temperature (WBT) of the entering air, which represents the lowest temperature water can reach by evaporation.

Psychrometric State Transitions I



Psychrometric State Transitions II

Key Points

- The process line on the psychrometric chart shows entering air at State 1 absorbing both moisture and heat, shifting up and to the right toward the saturation curve (State 2).
- Air enthalpy increases significantly: h_2 ¿ h_1, due to evaporation of water and sensible heating of air by warm inlet water.
- The humidity ratio increases from ω_1 to ω_2, representing the mass transfer of evaporated water into the air stream.
- Makeup water is required to replenish this evaporative loss, typically computed as: m_makeup $\approx$0.0008 * m_water * delta_T_water.

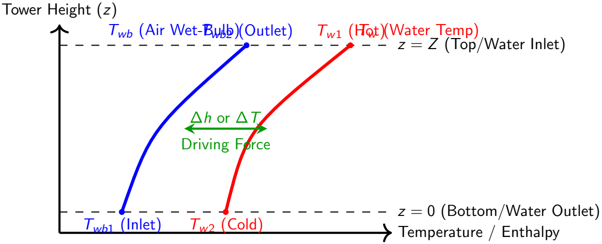
Cooling towers are primarily classified by the mechanism of air movement (Natural Draft vs. Mechanical Draft) and direction of air-water flow (Crossflow vs. Counterflow).

Classification of Cooling Towers II

Key Points

- Natural Draft: Utilizes large hyperbolic concrete chimneys to create a draft via density differences between warm, humid air inside and cooler, dry air outside.
- Mechanical Draft (Forced/Induced): Employs fans to move air. Forced draft fans push air in from the bottom; induced draft fans pull air out from the top.
- Counterflow: Air flows vertically upward while water droplets fall vertically downward. Offers high thermal efficiency due to maximum temperature differential.
- Crossflow: Air flows horizontally across vertically falling water. Results in lower pressure drops, lower fan energy, but requires more heat transfer surface area.

Thermal Profiles along Tower Height I



Thermal Profiles along Tower Height II

Key Points

- Along the cooling tower height (z), water temperature decreases continuously from the inlet at the top ($z=Z$) to the outlet at the bottom ($z=0$).
- Air wet-bulb temperature or enthalpy increases as it rises through the tower, picking up sensible heat and moisture from the falling water.
- The difference between water temperature and air wet-bulb temperature (or local air enthalpy) acts as the thermodynamic driving force.
- At any horizontal section, the water temperature must remain above the air wet-bulb temperature to ensure a positive heat transfer rate.

Evaluating cooling tower performance requires monitoring key indices: Cooling Range, Approach, Effectiveness, and the heat rejection capacity.

Cooling Tower Performance Metrics II

Key Points

- Cooling Range: The difference between inlet and outlet water temperatures: Range = T_w,in - T_w,out.
- Approach: The difference between the outlet cold water temperature and entering air wet-bulb temperature: Approach = T_w,out - T_wb,in. A lower approach indicates better performance.
- Effectiveness: Defined as the ratio of actual range to maximum theoretical range: η= Range / (Range + Approach) = (T_w,in - T_w,out) / (T_w,in - T_wb,in).
- Heat Rejection Load: Q = m_w * c_pw * (T_w,in - T_w,out). A typical cooling tower requires about 1.5 to 2.0 kW of electrical fan power per ton of refrigeration (TR).

Merkel's Theory of Cooling Towers I

Merkel's equation is the foundation for cooling tower design and rating. It relates the mass transfer coefficient, surface area, water flow rate, and enthalpy change.

Merkel's Theory of Cooling Towers II

Key Points

- Governing Integral: KaV / L = $\int$_T_w,out$\hat{T}$_w,in dT_w / (h_w - h_a), where K = mass transfer coefficient (kg/s.m$\hat{2}$), a = contact area per unit volume (m$\hat{2}$/m$\hat{3}$), V = active volume (m$\hat{3}$), L = water mass flow rate (kg/s).
- The left side (KaV/L) is often called the Tower Characteristic or NTU (Number of Transfer Units) of the cooling tower.
- h_w is the enthalpy of saturated air-water vapor mixture at the local bulk water temperature T_w.
- h_a is the enthalpy of the local moist air stream, calculated step-wise along the tower height.
- Integration is typically solved numerically using Chebyshev's 4-point method or Simpson's rule.

Industrial Challenges & Water Treatment I

Operating industrial cooling towers involves managing water quality and system efficiency. Evaporation continuously concentrates dissolved solids, leading to scaling, corrosion, and biological fouling.

Industrial Challenges & Water Treatment II

Key Points

- Cycles of Concentration (CoC): The ratio of dissolved solids in circulating water to that in makeup water. Optimizing CoC (typically 3 to 7) balances water conservation against scaling risk.
- Blowdown Rate: Water intentionally discharged to maintain CoC limits: $B = E / (CoC - 1) - D$, where E is evaporative loss and D is drift/windage loss.
- Biological Fouling & Legionella: Warm water in cooling towers is a breeding ground for bacteria like Legionella pneumophila. Continuous biocide dosing (chlorine, bromine, ozone) is mandatory.
- Drift Eliminators: Specifically designed baffles placed at the air discharge to capture water droplets entrained in the air stream, reducing drift loss to ¡0.005% of circulating water.

Concept and Function of Cooling Towers I

A cooling tower is a specialized direct-contact heat exchanger designed to reject waste heat from steam condensers and industrial processes to the atmosphere.

Concept and Function of Cooling Towers II

Key Points

- Primary function: Cools the warm water exiting the steam condenser so it can be recirculated and reused, minimizing freshwater consumption.
- Thermodynamic role: Serves as the ultimate heat sink in a steam power cycle, rejecting the latent heat of condensation to the ambient air.
- Maintains a low turbine exhaust backpressure (vacuum) by supplying cold cooling water to the surface condenser, maximizing overall cycle thermal efficiency.
- Economic and environmental benefit: Prevents thermal pollution of natural water bodies and reduces make-up water requirements to only 1.5% to 3.0% of total circulation.

The heat rejection mechanism in a cooling tower involves simultaneous heat and mass transfer between falling warm water droplets and rising ambient air.

Governing Physical Principles & Evaporative Cooling II

Key Points

- Sensible heat transfer: Driven by the temperature difference between the warm water and the dry-bulb temperature of the air stream ($Q_{sensible} = h_c A(T_w - T_{db})$).
- Latent heat transfer: Driven by the difference in vapor pressure between the saturated air film at the droplet surface and the bulk air stream.
- Mass transfer of water vapor: Evaporation of a small fraction (approx. 1% for every 6°C temperature drop) cools the remaining bulk water.
- Ultimate thermodynamic limit: The minimum theoretical temperature to which water can be cooled is the wet-bulb temperature (T_{wb}) of the entering air.

Classification of Cooling Towers I

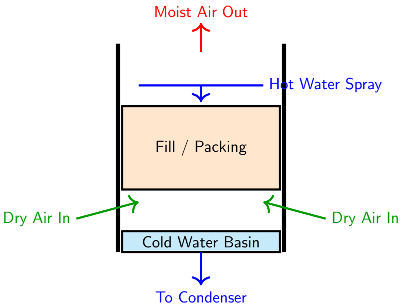
Cooling towers are classified by their method of air circulation, relative flow directions of water and air, and heat transfer mechanism.

Classification of Cooling Towers II

Key Points

- Natural Draft Towers: Utilize a tall hyperbolic concrete chimney to induce air flow through density differences between warm, moist internal air and cooler ambient air.
- Mechanical Draft Towers: Rely on fans to move air. Divided into Forced Draft (fans at air inlets push air in) and Induced Draft (fans at air outlet pull air through).
- Counter-Flow Arrangement: Water flows vertically downward while air flows vertically upward. Highly efficient thermodynamic temperature distribution.
- Cross-Flow Arrangement: Water flows downward while air flows horizontally across the falling water. Offers lower air-side pressure drops and easier fan maintenance.

Schematic of a Counter-Flow Wet Cooling Tower I



Schematic of a Counter-Flow Wet Cooling Tower II

Key Points

- Warm water from the condenser is pumped to the top distribution deck and distributed via spray nozzles.
- The packing/fill structure breaks water into thin films or small droplets, maximizing the interfacial surface area for heat and mass transfer.
- Air enters the lower section through louvers and travels upward, absorbing both sensible heat and evaporated water vapor.
- Cooled water accumulates in the basin at the base of the tower and is pumped back to the condenser inlet.

Key Constructional Features and Components I

A high-performance industrial cooling tower requires precise mechanical and structural components working in unison.

Key Constructional Features and Components II

Key Points

- Fills/Packing: Splash fills break water using successive grids; film fills spread water into thin, continuous sheets to maximize surface-to-volume ratio.
- Drift Eliminators: Specially shaped baffles that collect entrained water droplets from the exiting air stream, reducing drift losses to $< 0.005\%$ of circulation rate.
- Water Distribution Header: Pressurized pipe system and spray nozzles designed to provide uniform water distribution over the fill area without clogging.
- Air Intake Louvers: Guide incoming air uniformly into the tower fill zone while preventing water from splashing out and blocking external debris.

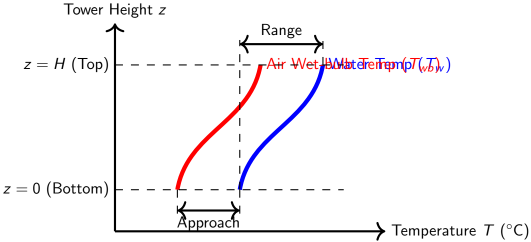
Thermal Performance Parameters & Equations I

Evaluating cooling tower performance relies on defining the temperature ranges, thermodynamic limits, and energy balance equations.

Key Points

- Cooling Range (R): The temperature difference between entering hot water and leaving cold water: $R = T_{w1} - T_{w2}$. Typically 6°C to 15°C.
- Approach (A): The temperature difference between leaving cold water and entering air wet-bulb temperature: $A = T_{w2} - T_{wb1}$. Usually 3°C to 8°C.
- Tower Effectiveness (ϵ): The ratio of actual cooling to the maximum theoretical cooling possible: $\epsilon = \frac{T_{w1} - T_{w2}}{T_{w1} - T_{wb1}} \times 100\%$.
- Energy Balance: The heat lost by the water equals the enthalpy gained by the moist air: $m_w c_{pw}(T_{w1} - T_{w2}) = m_a(h_{a2} - h_{a1})$.

Temperature Profiles along Tower Height I



Temperature Profiles along Tower Height II

Key Points

- Water temperature (T_w) decreases monotonically as it falls from the top ($z = H$) to the bottom basin ($z = 0$).
- Air wet-bulb temperature (T_{wb}) increases from T_{wb1} at the air inlet ($z = 0$) to T_{wb2} at the air exit ($z = H$) due to heat and moisture absorption.
- The driving potential for heat transfer at any cross-section is represented by the horizontal distance between the water temperature and air wet-bulb curves.
- Design trade-off: A smaller approach value increases tower effectiveness but requires a exponentially larger tower volume.

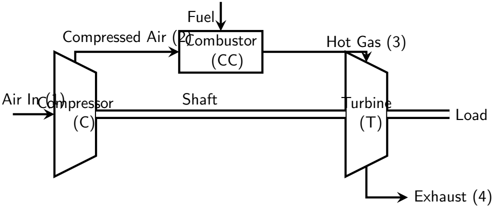
Introduction & Fundamental Concepts I

A gas turbine is a rotary internal combustion engine that converts the chemical energy of fuel into mechanical energy or thrust using a gaseous working fluid.

Key Points

- The working fluid remains in a single gaseous phase throughout the entire thermodynamic cycle, unlike steam turbines where phase change occurs.
- Major components include a rotary compressor, a combustion chamber (combustor), and a turbine expander.
- Characterized by a very high power-to-weight ratio, making them ideal for aviation propulsion and stationary power generation.
- Continuous combustion process allows for extremely high mass flow rates compared to reciprocating internal combustion engines.

Open Cycle Gas Turbine Schematic I



Open Cycle Gas Turbine Schematic II

Key Points

- State 1 to 2: Ambient air is drawn into the rotary compressor where it undergoes isentropic compression.
- State 2 to 3: High-pressure air enters the combustion chamber where fuel is injected and burned at constant pressure.
- State 3 to 4: High-temperature, high-pressure combustion products expand through the turbine to produce mechanical work.
- Exhaust gases at State 4 are discharged to the atmosphere, requiring a constant supply of fresh air.

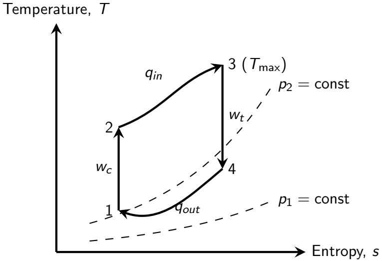
Classification of Gas Turbines I

Gas turbines are classified based on thermodynamic cycles, flow path geometry, combustion processes, and industrial applications.

Key Points

- Based on Cycle Type: Open cycle (working fluid is discharged) vs. Closed cycle (working fluid is cooled and recirculated).
- Based on Fluid Flow Path: Axial flow (fluid flows parallel to shaft, for high mass flows) vs. Radial flow (fluid flows radially, for small capacity).
- Based on Combustion Method: Constant pressure combustion (Brayton cycle) vs. Constant volume combustion (Holzwarth cycle, obsolete).
- Based on Application: Aircraft propulsion (Turbojet, Turbofan, Turboprop) vs. Industrial power generation and marine propulsion.

Brayton Cycle T-s Diagram I



Brayton Cycle T-s Diagram II

Key Points

- Process 1-2: Reversible adiabatic (isentropic) compression of the gas in the compressor.
- Process 2-3: Constant-pressure heat addition (q_{in}) from the external source or combustion.
- Process 3-4: Reversible adiabatic (isentropic) expansion of the gas in the turbine.
- Process 4-1: Constant-pressure heat rejection (q_{out}) to the surroundings or a cooler.

Air-Standard Assumptions & Ideal Cycle I

To simplify the analysis of gas turbine cycles, we use the cold air-standard assumptions.

Key Points

- The working fluid is air, which behaves as an ideal gas with constant specific heats evaluated at 25°C ($C_p = 1.005$ kJ/kg·K, $C_v = 0.718$ kJ/kg·K, $\gamma = 1.4$).
- The combustion process is replaced by a heat-addition process from an external source.
- The exhaust process is replaced by a heat-rejection process to the ambient air to restore the cycle to its initial state.
- All processes are internally reversible, meaning friction and turbulence losses are neglected.

Thermodynamic Analysis & Efficiency I

The thermal efficiency of the ideal Brayton cycle is derived from the net work output and heat input.

Key Points

- Compressor Work: $w_c = C_p(T_2 - T_1)$ — Turbine Work: $w_t = C_p(T_3 - T_4)$.
- Heat Input: $q_{in} = C_p(T_3 - T_2)$ — Heat Rejected: $q_{out} = C_p(T_4 - T_1)$.
- Thermal Efficiency: $\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$.
- Since $T_2/T_1 = T_3/T_4 = r_p^{(\gamma-1)/\gamma}$, the efficiency simplifies to: $\eta_{th} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}$, where $r_p = p_2/p_1$ is the pressure ratio.

Performance Parameters & Comparisons I

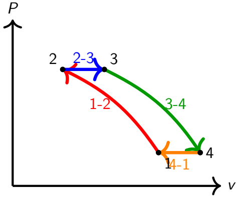
Evaluating the performance factors that govern the design and operation of Brayton cycle gas turbines.

Performance Parameters & Comparisons II

Key Points

- Effect of Pressure Ratio (r_p): Thermal efficiency increases monotonically with r_p, but net work output reaches a peak and then declines.
- Optimum Pressure Ratio for Maximum Net Work: $r_{p,opt} = (\frac{T_3}{T_1})^{\frac{\gamma}{2(\gamma-1)}}$, where T_3 is the metallurgical limit of the turbine blades.
- Open vs. Closed Cycles: Open cycles are lighter and use atmospheric air directly, whereas closed cycles can use helium or argon to enhance heat transfer.
- Back Work Ratio (BWR): Gas turbines have a high BWR (typically 40% to 60%) because the compressor consumes a large fraction of turbine work.

Ideal Brayton Cycle on P-v Coordinates I

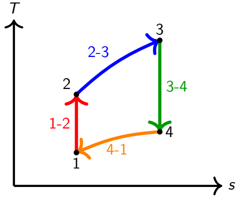


Ideal Brayton Cycle on P-v Coordinates II

Key Points

- Process 1-2 (Isentropic Compression): Reversible adiabatic compression in the compressor, modeled by P*v^γ= C. Work input W_c = h2 - h1.
- Process 2-3 (Isobaric Heat Addition): Heat addition in the combustor at constant pressure P2 = P3. Heat input Q_in = h3 - h2.
- Process 3-4 (Isentropic Expansion): Reversible adiabatic expansion in the turbine, modeled by P*v^γ= C. Work output W_t = h3 - h4.
- Process 4-1 (Isobaric Heat Rejection): Constant pressure heat rejection (or exhaust/intake in open cycle) at P4 = P1. Heat rejected Q_out = h4 - h1.

Ideal Brayton Cycle on T-s Coordinates I



Ideal Brayton Cycle on T-s Coordinates II

Key Points

- Process 1-2 (Isentropic): Vertical line on T-s coordinates. Entropy is constant (s1 = s2), representing zero internal friction or heat transfer.
- Process 2-3 (Isobaric Heating): Sloped curve upwards to the right. The slope of a constant-pressure line on T-s is positive and proportional to T (dT/ds = T/Cp).
- Process 3-4 (Isentropic Expansion): Vertical line representing expansion in the turbine (s3 = s4) down to the ambient/exhaust pressure.
- Process 4-1 (Isobaric Cooling): Sloped curve downwards to the left, representing the return of the working fluid (or air) to the initial state.

Irreversibilities in the Actual Gas Turbine Cycle I

Unlike the ideal Brayton cycle, real-world systems suffer from fluid friction, pressure drops, and heat losses, which drastically affect the overall cycle performance.

Irreversibilities in the Actual Gas Turbine Cycle II

Key Points

- Frictional Pressure Drops: Viscous effects in the combustion chamber and air ducts cause a pressure drop (P3 ¡ P2), requiring higher compressor work.
- Compressor Irreversibility: Aerodynamic drag and turbulence in compressor blades increase entropy (s2 ¿ s1), leading to higher exit temperature (T2a ¿ T2s).
- Turbine Irreversibility: Entropy increases during expansion (s4 ¿ s3) due to fluid friction, reducing the turbine work output and increasing exhaust temperature (T4a ¿ T4s).
- Heat Dissipation: Non-adiabatic operation causes heat losses from the combustion chamber, turbine shell, and interconnecting ducting to the atmosphere.

Isentropic Compressor Efficiency (\{}eta_c) I

Isentropic compressor efficiency quantifies the deviation of the actual work input from the ideal reversible work input for the same pressure ratio.

Key Points

- Definition: Ratio of isentropic work input to actual work input: η_c = W_isentropic / W_actual.
- Formula: η_c = (h2s - h1) / (h2a - h1) = (T2s - T1) / (T2a - T1) under constant specific heat (Cp) assumption.
- Temperature Rise: Actual discharge temperature T2a = T1 + (T2s - T1) / η_c. Higher entropy leads to T2a ¿ T2s.
- Engineering Range: Modern industrial axial-flow compressors operate with isentropic efficiencies between 85% and 92%.

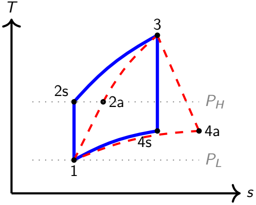
Isentropic Turbine Efficiency (\{}eta_t) I

Isentropic turbine efficiency measures the fraction of the ideal reversible work that is successfully converted into actual shaft work output.

Key Points

- Definition: Ratio of actual work output to isentropic work output: η_t = W_actual / W_isentropic.
- Formula: η_t = (h3 - h4a) / (h3 - h4s) = (T3 - T4a) / (T3 - T4s) assuming constant specific heat (Cp).
- Exhaust Temperature: Actual turbine exit temperature T4a = T3 - η_t * (T3 - T4s). Frictional heating results in T4a ¿ T4s.
- Engineering Range: High-pressure turbine stages achieve 88% to 94% efficiency, using advanced aerodynamic blades and film cooling.

Actual vs. Ideal Cycle on T-s Coordinates I



Actual vs. Ideal Cycle on T-s Coordinates II

Key Points

- Path Deviation: The actual compression path (1-2a) curves to the right, indicating entropy generation (s2a ¿ s1) inside the compressor.
- Actual Expansion Path: The actual expansion path (3-4a) curves to the right (s4a ¿ s3), meaning less work is extracted.
- Increased Work Input: The vertical distance between T1 and T2a is larger than T1 to T2s, representing higher compressor work input.
- Reduced Work Output: The vertical distance between T3 and T4a is smaller than T3 to T4s, representing lower turbine work output.

Performance Metrics & Overall Efficiency I

Irreversibilities drastically lower net work output and thermal efficiency while raising the back work ratio.

Key Points

- Net Work Output (W_net): W_net,actual = W_turbine,actual - W_compressor,actual = (h3 - h4a) - (h2a - h1).
- Back Work Ratio (BWR): BWR = W_c,actual / W_t,actual. Can exceed 50-60% in gas turbines compared to 1-2% in steam Rankine cycles.
- Combustor Efficiency (η_b): Accounts for heat release losses and incomplete fuel combustion, typically ranging from 97% to 99%.
- Thermal Efficiency (η_th): η_th,actual = W_net,actual / Q_in,actual. Typically ranges from 30% to 45% in simple cycles, compared to ¿50% in ideal calculations.

Open Cycle Gas Turbines (Brayton Cycle) I

An open cycle gas turbine draws ambient air, compresses it, heats it via internal combustion of fuel, expands it in a turbine to generate mechanical shaft power, and exhausts the combustion gases back to the atmosphere. The working fluid is continuously replaced.

Open Cycle Gas Turbines (Brayton Cycle) II

Key Points

- Atmospheric intake: Fresh air is drawn continuously at ambient conditions (P_1, T_1).
- Compression stage: Air is compressed to P_2, T_2 in a rotary axial or centrifugal compressor (isentropic efficiency $\eta_c \approx 80 - 85\%$).
- Internal combustion: Fuel is injected and combusted at constant pressure ($P_3 \approx P_2$), reaching metallurgical limit temperature T_3.
- Expansion and exhaust: High temperature gases expand to $P_4 \approx P_1$ in the turbine to produce net shaft work, followed by atmospheric disposal.
- System characteristics: Compact footprint, low capital cost, quick starting (seconds to minutes), but dependent on high-grade distillate fuels.

Closed Cycle Gas Turbines I

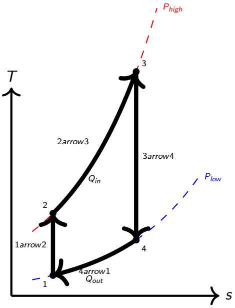
In a closed cycle gas turbine, the working fluid (air, helium, or nitrogen) is recirculated continuously in a closed loop. Heat is added externally through a heat exchanger, and the exhaust fluid is cooled before re-compression.

Closed Cycle Gas Turbines II

Key Points

- External heat source: Fuel combustion occurs externally; heat is transferred via a tube-bundle heat exchanger or a nuclear reactor core.
- Alternative working fluids: Helium ($C_p = 5.19$ kJ/kg·K) or Helium-Xenon mixtures improve thermal conductivity and reduce compressor size.
- Cycle pressurization: System operates at high density by maintaining elevated pressure levels (e.g., $P_{min} = 10 - 20$ bar), improving heat transfer and power density.
- Load control: Power output is regulated by varying the system mass flow rate (inventory control) while keeping temperatures and efficiency constant.
- System complexity: Requires massive cooling water flow for the precooler, larger heat exchangers, and has high initial capital costs.

T-s Diagram of Ideal Brayton Cycle I



T-s Diagram of Ideal Brayton Cycle II

Key Points

- Process 1-2: Isentropic compression. Work input is given by: $w_c = C_p(T_2 - T_1) = C_p T_1(r_p^{(\gamma-1)/\gamma} - 1)$.
- Process 2-3: Constant pressure heat addition. Heat supplied is: $q_{in} = C_p(T_3 - T_2)$.
- Process 3-4: Isentropic expansion. Turbine work output is: $w_t = C_p(T_3 - T_4) = C_p T_3(1 - 1/r_p^{(\gamma-1)/\gamma})$.
- Process 4-1: Constant pressure heat rejection. Heat rejected is: $q_{out} = C_p(T_4 - T_1)$.
- Ideal Brayton Efficiency: $\eta_{th} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}$, where $r_p = P_2/P_1$ is the pressure ratio.

Performance Improvement: Regeneration I

Regeneration preheats the compressed air before it enters the combustor by using the hot exhaust gases from the turbine. This recovers waste heat, reducing the required fuel input.

Performance Improvement: Regeneration II

Key Points

- Prerequisite: Turbine exit temperature (T_4) must be higher than compressor exit temperature (T_2), i.e., $T_4 > T_2$.
- Regenerator effectiveness: $\epsilon = \frac{h_x - h_2}{h_4 - h_2} \approx \frac{T_x - T_2}{T_4 - T_2}$, representing the ratio of actual to maximum possible heat transfer.
- Fuel savings: Combustor heat input drops from $C_p(T_3 - T_2)$ to $C_p(T_3 - T_x)$, increasing thermal efficiency without changing net work.
- Efficiency relation: $\eta_{reg} = 1 - \frac{T_1}{T_3}(r_p)^{(\gamma-1)/\gamma}$. Regeneration is highly effective at lower pressure ratios.
- Physical limits: Pressure drops in heat exchanger channels reduce net work output slightly due to friction losses.

Performance Improvement: Intercooling I

Intercooling splits the compression process into multiple stages with an intermediate heat exchanger to cool the working fluid. This reduces the specific volume of the gas, decreasing the required compressor work.

Performance Improvement: Intercooling II

Key Points

- Work reduction: Compression work $w_c = \int VdP$ is lowered because cooling the gas decreases its specific volume V at the intermediate state.
- Optimal stage pressure: For a two-stage compressor with perfect intercooling ($T_3 = T_1$), the intermediate pressure is $P_i = \sqrt{P_1 P_2}$.
- Thermal efficiency impact: Since the final delivery temperature T_4 is lower than without intercooling, more fuel is needed to reach T_5, which lowers thermal efficiency if used alone.
- Combined regeneration: Because the compressor outlet temperature is lower, intercooling is almost always combined with regeneration to maximize cycle efficiency.
- Mechanical implementation: Requires a continuous supply of cool water or large air-to-air cooling surfaces.

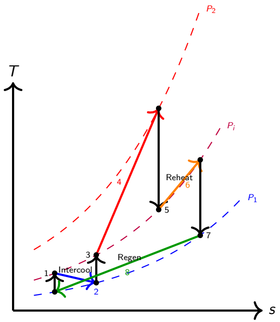
Performance Improvement: Reheating I

Reheating expands the gas through a high-pressure turbine stage, reheats it back to near-maximum temperature in a secondary combustor, and expands it through a low-pressure turbine stage, maximizing work output.

Key Points

- Work output increase: Turbine work $w_t = \int V dP$ is increased because reheating increases the specific volume of the expanding gas.
- Optimal intermediate pressure: For two-stage expansion with reheat back to T_3 ($T_5 = T_3$), the optimal pressure is $P_i = \sqrt{P_3 P_4}$.
- Exhaust heat penalty: Reheating increases the LP turbine exhaust temperature T_6, which increases heat rejection losses and lowers efficiency if used alone.
- Synergy with regeneration: The elevated exhaust temperature ($T_6 > T_4$) makes regeneration highly efficient when combined with reheating.
- Metallurgical limit: The maximum reheat temperature is constrained by the low-pressure turbine blade materials.

Intercooled-Reheated-Regenerative Cycle I



Intercooled-Reheated-Regenerative Cycle II

Key Points

- Ericsson cycle approach: By increasing the number of intercooling and reheating stages, the cycle behavior approaches isothermal heat addition and rejection.
- Thermodynamic advantages: Combining intercooling, reheating, and regeneration increases both net work output (W_{net}) and thermal efficiency.
- Regeneration role: Preheats the cold air leaving the high-pressure compressor stage using LP turbine exhaust, reducing the fuel demand.
- Practical applications: High-capacity power plants, marine gas turbine engines, and advanced industrial cogeneration installations.
- System limits: Increased mechanical complexity, complex piping friction losses, and higher pressure drops across heat exchangers.

Thermodynamic Fundamentals of Intercooling I

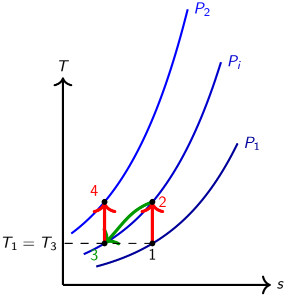
Intercooling is the process of cooling the working fluid between successive stages of compression to reduce the total work required for compression.

Thermodynamic Fundamentals of Intercooling II

Key Points

- Compressor work input is given by $W_c = \int v dp$. By cooling the gas between stages, its specific volume (v) is reduced, thereby decreasing the work needed for subsequent compression stages.
- For two-stage compression, the air is compressed from pressure P_1 to an intermediate pressure P_i, cooled back to the initial temperature T_1 (perfect intercooling), and then compressed to the final pressure P_2.
- Optimum intermediate pressure for minimum total compressor work is $P_i = \sqrt{P_1 P_2}$ when intercooling is perfect ($T_3 = T_1$) and stage efficiencies are equal.
- While intercooling decreases compressor work and increases net work output ($W_{net} = W_t - W_c$), it lowers the temperature at the compressor outlet, requiring more heat input (Q_{in}) in the combustor, which can reduce thermal efficiency unless regeneration is employed.

T-s Diagram: Brayton Cycle with Intercooling I



T-s Diagram: Brayton Cycle with Intercooling II

Key Points

- Process 1-2: First-stage isentropic compression from pressure P_1 to intermediate pressure P_i.
- Process 2-3: Intercooling at constant pressure P_i. In perfect intercooling, temperature drops back to $T_3 = T_1$.
- Process 3-4: Second-stage isentropic compression from pressure P_i to final pressure P_2.
- The shaded area reduction or shift showing lower overall work input compared to single-stage compression (1 to a higher state at P_2 directly).

Optimum Intermediate Pressure Derivation I

To minimize total compressor work in a two-stage compressor with perfect intercooling ($T_3 = T_1$):

Key Points

- Total work: $W_c = W_{c1} + W_{c2} = \frac{\gamma R}{\gamma - 1}[(T_2 - T_1) + (T_4 - T_3)]$.
- Expressing in terms of pressure ratios: $W_c = \frac{\gamma R T_1}{\gamma - 1}[(\frac{P_i}{P_1})^{\frac{\gamma-1}{\gamma}} - 1 + (\frac{P_2}{P_i})^{\frac{\gamma-1}{\gamma}} - 1]$ (assuming $T_3 = T_1$ and equal stage efficiencies).
- Differentiating W_c with respect to P_i and setting to zero: $\frac{dW_c}{dP_i} = 0$.
- This yields: $P_i = \sqrt{P_1 P_2}$, indicating that the pressure ratio across each stage must be equal: $\frac{P_i}{P_1} = \frac{P_2}{P_i}$.

Thermodynamic Fundamentals of Reheating I

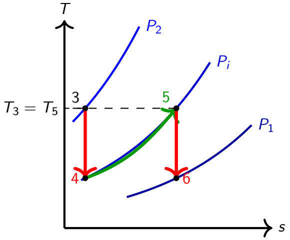
Reheating involves expanding the gas through multiple turbine stages and heating it between stages to increase the net work output of the turbine.

Thermodynamic Fundamentals of Reheating II

Key Points

- Reheating increases the average temperature at which heat is added to the cycle, thereby increasing the turbine work output (W_t).
- For two-stage expansion, gas expands from P_2 to P_i in the high-pressure turbine, is reheated at constant pressure P_i back to the maximum cycle temperature $T_5 = T_3$, and expands from P_i to P_1 in the low-pressure turbine.
- The optimum intermediate pressure for maximum turbine work is $P_i = \sqrt{P_1 P_2}$ (assuming perfect reheat and equal stage efficiencies).
- Like intercooling, reheating increases the net work output (W_{net}), but it also increases the exhaust gas temperature, which increases the heat rejected (Q_{out}) and can lower thermal efficiency unless paired with regeneration.

T-s Diagram: Reheating in Gas Turbines I



T-s Diagram: Reheating in Gas Turbines II

Key Points

- Process 3-4: Expansion in high-pressure turbine (HPT) from P_2 to intermediate pressure P_i.
- Process 4-5: Constant pressure reheating at P_i back to maximum temperature ($T_5 = T_3$).
- Process 5-6: Expansion in low-pressure turbine (LPT) from P_i to exhaust pressure P_1.
- Reheating increases the total area under the expansion curve on the T-s diagram, corresponding to larger turbine work output.

Combined Cycle: Intercooling, Reheating, and Regeneration I

To maximize both work output and thermal efficiency, modern gas turbine power plants combine intercooling, reheating, and regeneration.

Combined Cycle: Intercooling, Reheating, and Regeneration II

Key Points

- Individually, intercooling and reheating increase the net work output (W_{net}), but lower the thermal efficiency due to higher heat input requirements.
- Regeneration utilizes the high-temperature exhaust from the low-pressure turbine to preheat the compressed air before it enters the combustor, recovering waste heat.
- When intercooling and reheating are combined with regeneration, the thermal efficiency is significantly improved, achieving values up to 45% - 50% in advanced industrial gas turbine systems.
- Multi-stage systems approach the Ericsson cycle or Stirling cycle limit as the number of stages of intercooling and reheating approaches infinity.

Industrial Applications and Performance Metrics I

Practical implementation of intercooling and reheating in modern utility power plants, marine propulsion, and heavy industries.

Industrial Applications and Performance Metrics II

Key Points

- Used in large-scale stationary power plants (e.g., General Electric LMS100 aeroderivative gas turbine, which uses intercooling to achieve over 44% simple-cycle efficiency).
- Key performance metrics: Specific Fuel Consumption (SFC) is reduced, and Work Ratio (W_{net}/W_t) is increased, making the system less sensitive to component inefficiencies.
- Challenges: Increased complexity, higher capital cost, pressure drops in the intercooler/reheater ducts, and requirement of cooling water or heat exchangers.
- Design trade-offs: Number of stages is typically limited to two or three due to diminishing returns, piping losses, and mechanical design limitations.

The Concept of Regeneration I

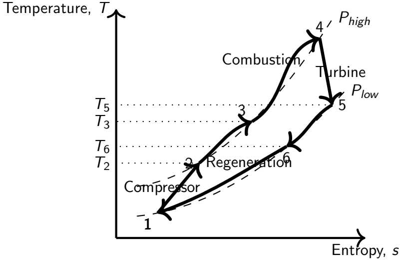
Regeneration is a thermodynamic modification to the Brayton cycle where exhaust gas energy is utilized to preheat compressed air before it enters the combustor. This process directly reduces the external heat input required, thereby increasing the thermal efficiency of the cycle.

The Concept of Regeneration II

Key Points

- Energy Recovery: The turbine exhaust temperature (T5) is often significantly higher than the compressor exit temperature (T2).
- Heat Exchanger (Regenerator): A counter-flow heat exchanger transfers heat from the hot exhaust gas stream to the relatively colder compressed air stream.
- Efficiency Boost: Since less fuel is burned in the combustion chamber to reach the turbine inlet temperature (T4), thermal efficiency increases.
- Operational Constraints: Regeneration is only thermodynamically feasible when the turbine exhaust temperature is higher than the compressor exit temperature (T5 ¿ T2).

T-s Diagram of Regeneration Cycle I



T-s Diagram of Regeneration Cycle II

Key Points

- 1 to 2: Isentropic compression of air in the compressor, raising temperature from T1 to T2.
- 2 to 3: Constant pressure preheating of compressed air inside the regenerator, raising temperature to T3.
- 3 to 4: Constant pressure heat addition (combustion) inside the combustion chamber, reaching peak cycle temperature T4.
- 4 to 5: Isentropic expansion of gas in the turbine, generating work and exiting at temperature T5.
- 5 to 6: Constant pressure heat rejection from turbine exhaust gas inside the regenerator, cooling it to T6.
- 6 to 1: Heat rejection to the atmosphere to close the thermodynamic cycle.

Regenerator Effectiveness & Efficiency Analysis I

The performance of a regenerator is quantified by its effectiveness (ϵ), defined as the ratio of actual heat transfer to the maximum possible heat transfer. Mathematically: $\epsilon = \frac{h\_3 - h\_2}{h\_5 - h\_2} \approx \frac{T\_3 - T\_2}{T\_5 - T\_2}$ (assuming constant specific heat Cp).

Regenerator Effectiveness & Efficiency Analysis II

Key Points

- Effectiveness Value: Typical industrial regenerators operate with an effectiveness (ϵ) of 60% to 80%. High effectiveness requires larger heat transfer area.
- Thermal Efficiency Equation: For an ideal Brayton cycle with regeneration, the thermal efficiency is given by: η_th $=$ 1 - ($\frac{T\_1}{T\_3}$) (r_p)^$\frac{\gamma-1}{\gamma}$ $=$ 1 - $\frac{T\_1}{T\_4}$(r_p)^$\frac{\gamma-1}{\gamma}$ where rp is the pressure ratio.
- Pressure Ratio Impact: Unlike the standard Brayton cycle, the efficiency of a regenerative cycle decreases as the pressure ratio increases.
- Optimal Pressure Ratio: Regeneration is highly beneficial at lower pressure ratios. At high pressure ratios, T2 exceeds T5, making regeneration counterproductive.

Introduction to Combustion Chambers (Section 7.5) I

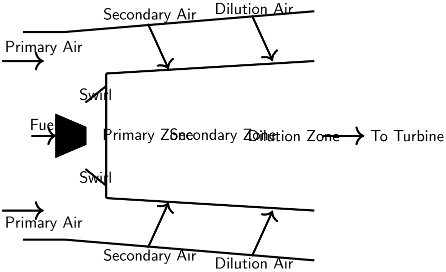
The combustion chamber (combustor) in a gas turbine is responsible for burning fuel with high-pressure air to generate high-temperature gas. It must operate continuously, stably, and efficiently under extreme aerodynamic and thermal conditions.

Introduction to Combustion Chambers (Section 7.5) II

Key Points

- High Heat Release Rate: Gas turbine combustors release heat at rates of 100 to 200 MW/m$\hat{3}$-atm, far exceeding conventional boiler furnaces.
- Pressure Loss Minimization: Pressure drop across the combustor (typically 3% to 8% of compressor delivery pressure) must be minimized to maintain cycle efficiency.
- Wide Operating Range: Stable combustion must be sustained over a wide range of fuel-to-air ratios and mass flow rates.
- Combustion Efficiency: Modern combustors achieve an efficiency of 98% to 99% under design conditions.

Combustion Chamber Zones and Air Distribution I



Combustion Chamber Zones and Air Distribution II

Key Points

- Primary Zone: Receives 15-20% of total air. Fuel is injected here and mixed with swirling air to create a stable flame. Equivalence ratio is close to stoichiometric (1.0).
- Secondary Zone: Receives 30% of total air. Helps complete the combustion of partially burned fuel and cools the dissociation products.
- Dilution Zone: Receives the remaining 50% of air. Cools the exhaust gases to the precise Turbine Inlet Temperature (TIT) profile required to prevent blade damage.
- Film Cooling: A portion of the air is routed along the inner liner wall to form a cooling boundary layer, protecting the metal from flame temperatures exceeding 2000 K.

Performance Parameters and Fuel Injection I

The design of a combustion chamber depends heavily on fuel atomization, mixing rates, and aerodynamic stability. Essential parameters include the reference velocity ($U_{ref} = \frac{\dot{m}}{\rho A_{ref}}$), combustion intensity, and pressure loss factor.

Performance Parameters and Fuel Injection II

Key Points

- Reference Velocity: Typical velocities inside the combustor casing range between 15 and 30 m/s to prevent flame blowout (blow-off).
- Combustion Intensity: Quantified as I = (m_f * CV) / (V_c * P), representing heat release per unit volume per unit pressure.
- Pressure Loss: Divided into fundamental loss (due to heat addition/expansion) and cold loss (due to friction and turbulence in swirlers/dilution holes).
- Fuel Atomization: Done via pressure-swirl or air-blast atomizers to reduce fuel droplet size (Sauter Mean Diameter ¡ 50 microns) for rapid evaporation.

Practical Types of Combustion Chambers & Regeneration Limits I

Gas turbine installations employ different combustor configurations based on application. Regeneration implementation also faces practical limitations that dictate mechanical design choices.

Practical Types of Combustion Chambers & Regeneration Limits II

Key Points

- Can-Type: Individual cylindrical chambers arranged around the shaft. Easy to maintain and test, but has high pressure loss and ignition propagation issues.
- Annular Combustor: A single continuous ring-shaped chamber. Highly compact, provides optimal outlet temperature profile, but is mechanically complex to service.
- Can-Annular: Individual liners nested inside a single outer annular casing, combining the structural strength of can-type with the compactness of annular.
- Regeneration Limits: High temperature gradients in regenerators induce thermal stress, leading to cracking. Additionally, soot deposition increases thermal resistance.

Industrial Power Generation Applications I

Gas turbines are widely used in utility and industrial power generation due to their rapid start-up capability, compact size, and flexibility.

Key Points

- Base Load CCGT Plants: Combined Cycle Gas Turbine plants achieve high thermal efficiency (above 60%) by using exhaust heat to run a steam turbine cycle.
- Peaking Power Stations: Simple-cycle gas turbines can start up from cold to full load in 5 to 10 minutes, stabilizing grid frequency during sudden load surges.
- Cogeneration / CHP Systems: Generate electricity while capturing exhaust heat to produce process steam or hot water for industrial processes.

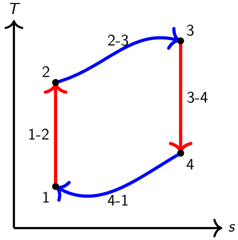
Aviation and Marine Propulsion I

Gas turbines are the primary propulsion systems for commercial and military aircraft, as well as high-speed marine vessels.

Key Points

- Aviation Turbofans: High bypass ratios provide high propulsive efficiency, lower fuel consumption, and reduced noise levels in civil aviation.
- Aviation Turbojets & Turboprops: Turbojets power high-speed supersonic aircraft; turboprops drive cargo and regional transport aircraft efficiently.
- Marine Gas Turbines: Power naval ships (e.g., destroyers, frigates) and fast ferries because of high power density, rapid starting, and ease of maintenance.

Brayton Cycle T-s Diagram I

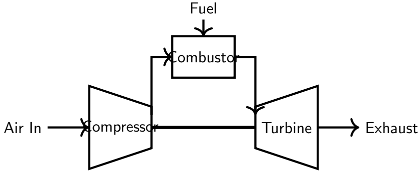


Brayton Cycle T-s Diagram II

Key Points

- Process 1-2: Isentropic compression of air in the compressor (reversibly increases pressure and temperature).
- Process 2-3: Constant pressure heat addition in the combustor (combustion of fuel at constant pressure).
- Process 3-4: Isentropic expansion of gas through the turbine (delivers mechanical energy).
- Process 4-1: Constant pressure heat rejection to complete the cycle (cooling the working fluid or exhaust).

Open-Cycle Gas Turbine Schematic I



Open-Cycle Gas Turbine Schematic II

Key Points

- Air In: Ambient air enters the compressor inlet where mechanical compression increases its energy state.
- Combustor: Heat addition occurs by spraying fuel into the high-pressure air stream and igniting it.
- Turbine: Exploding gases expand through the turbine rotor blades, driving the compressor and load.
- Exhaust: Hot combustion products are discharged to the atmosphere, completing the open cycle.

Brayton Cycle Performance Formulas I

Analysis of a gas turbine cycle requires calculating temperatures at state points to determine efficiency and work output.

Key Points

- Pressure Ratio: $r_p = P_2/P_1 = P_3/P_4$, which governs the temperature rise across the compressor.
- Isentropic Temperature Relations: $T_2/T_1 = T_3/T_4 = (r_p)^{(\gamma-1)/\gamma}$, where $\gamma = 1.4$ for air.
- Thermal Efficiency: $\eta_{th} = 1 - 1/(r_p)^{(\gamma-1)/\gamma}$, illustrating efficiency depends solely on pressure ratio.
- Net Work Output: $w_{net} = w_t - w_c = C_p(T_3 - T_4) - C_p(T_2 - T_1)$, where $C_p = 1.005$ kJ/kg-K.
- Work Ratio (WR): $WR = w_{net}/w_t$, indicating the fraction of turbine work available for external load.

Numerical Problem: Simple Gas Turbine I

Consider a simple Brayton cycle gas turbine operating with a pressure ratio of 6. The inlet air temperature is 300 K (27 C) and the maximum cycle temperature is 1100 K. Assuming isentropic efficiencies of 100%, find the cycle efficiency and net work output. Take $C_p = 1.005$ kJ/kg-K and $\gamma = 1.4$.

Key Points

- Inlet Parameters: Inlet air temperature $T_1 = 300$ K, and cycle pressure ratio $r_p = 6$.
- Peak Temperature: Combustor exit temperature $T_3 = 1100$ K, representing the maximum permissible metallurgical temperature.
- Fluid Properties: Constant specific heat $C_p = 1.005$ kJ/kg-K, and specific heat ratio $\gamma = 1.4$ for standard air.
- Calculation Goals: Determine the temperatures at all cycle states, thermal efficiency, and specific net work output.

Numerical Solution & Calculations I

Step-by-step calculations for state point temperatures, work terms, and efficiency.

Numerical Solution & Calculations II

Key Points

- Compressor Discharge Temperature:
  $T_2 = T_1 \times (r_p)^{(\gamma-1)/\gamma} = 300 \times (6)^{0.286} = 500.5$ K.
- Turbine Exhaust Temperature:
  $T_4 = T_3/(r_p)^{(\gamma-1)/\gamma} = 1100/(6)^{0.286} = 659.3$ K.
- Specific Work Computations: Compressor work
  $w_c = C_p(T_2 - T_1) = 1.005 \times (500.5 - 300) = 201.5$ kJ/kg.
  Turbine work
  $w_t = C_p(T_3 - T_4) = 1.005 \times (1100 - 659.3) = 442.9$ kJ/kg.
- Net Specific Work Output:
  $w_{net} = w_t - w_c = 442.9 - 201.5 = 241.4$ kJ/kg.
- Cycle Thermal Efficiency:
  $\eta_{th} = 1 - 1/(r_p)^{(\gamma-1)/\gamma} = 1 - 1/1.6685 = 40.07\%$. (Verified as $w_{net}/q_{in} = 241.4/[1.005 \times (1100 - 500.5)] = 40.07\%$.)