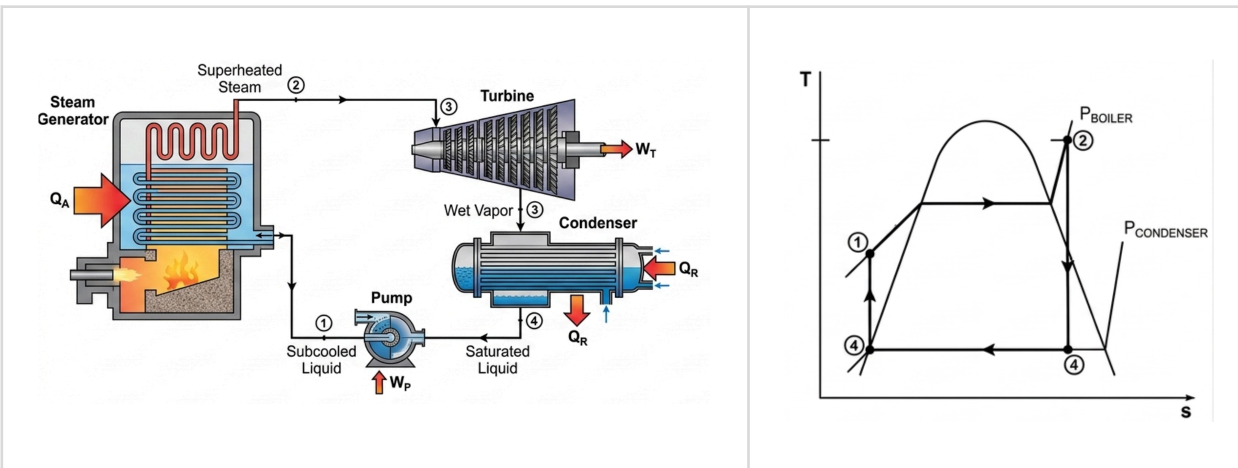
RANKINE CYCLE

The Rankine cycle serves as the fundamental thermodynamic model for mechanical engineering applications involving vapor power plants. It is a highly structured closed system designed to continuously convert thermal energy into useful mechanical work. In practice, water typically serves as the working fluid due to its high latent heat capacity, chemical stability, and widespread availability.

Unlike gas power cycles where the working fluid remains strictly in the gaseous phase, the Rankine cycle relies on a continuous phase change between the liquid and vapor states. Heat is supplied externally from a combustion chamber, nuclear reactor, or geothermal pocket to an isolated heat exchanger—the steam generator. This setup decouples the energy source from the working fluid, protecting the mechanical machinery downstream from corrosive combustion byproducts.



A standard, simple Rankine cycle is evaluated through four distinct thermodynamic processes across four main components:

Process 1 to 2: Isobaric Heat Addition

Subcooled liquid water at a high operating pressure enters the steam generator (boiler). Energy (Q_A) is transferred to the fluid at a constant pressure (P_{Boiler}). The fluid transitions from a subcooled liquid to a saturated liquid, undergoes vaporization into a saturated vapor, and is typically heated further into the superheated vapor region. Superheating is a critical engineering safeguard: it ensures that the fluid entering the turbine has a high energy content and prevents premature condensation, which would cause liquid droplets to erode the turbine blades.

Process 2 to 3: Isentropic Expansion

Superheated steam at state 2 enters the power-producing turbine. The fluid expands through the turbine stages to a lower condenser pressure ($P_{Condensr}$), doing shaft work (W_T) that drives an electrical generator. In an ideal cycle, this expansion is assumed to be perfectly reversible and adiabatic, making it an isentropic process ($s_2 = s_3$). As the steam expands and relinquishes its energy, it crosses into the wet vapor region, exiting at state 3 as a high-quality mixture of liquid and vapor.

Process 3 to 4: Isobaric Heat Rejection

The wet vapor mixture enters the condenser, a low-pressure heat exchanger. Heat (Q_R) is rejected from the steam to a cooling medium, such as river water or atmospheric air, maintaining a constant pressure ($P_{Condenser}$). The fluid undergoes a complete phase change, shedding its latent heat of vaporization until it reaches the saturated liquid state at state 4. It is crucial that the fluid is fully condensed to a saturated liquid before entering the pump to prevent mechanical damage.

Process 4 to 1: Isentropic Compression

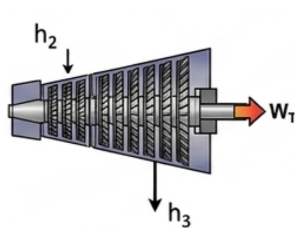
The saturated liquid water at state 4 enters the liquid pump. The pump raises the fluid pressure from the low condenser pressure back up to the high steam generator operating pressure at state 1. This process is ideally reversible and adiabatic, meaning it is an isentropic compression ($s_4 = s_1$). Because liquid water is nearly incompressible, the volume remains virtually constant, requiring a relatively small amount of mechanical work input (W_P) compared to the work output generated by the turbine.

COMPONENT-BY-COMPONENT THERMODYNAMIC ANALYSIS

To perform a rigorous energy balance across the individual components of an ideal Rankine cycle, engineers apply the **First Law of Thermodynamics for an Open System under Steady-Flow Conditions**. Neglecting changes in kinetic and potential energy, the governing equation simplifies to a straightforward energy balance:

$$E_{in} = {}_{Eout}$$

The Steam Turbine

	The high-pressure, high-temperature steam enters the turbine casing at state 2 and exits at state 3. Evaluating the control volume surrounding the turbine, the energy entering the system is carried by the inlet fluid enthalpy, while the exiting energy consists of the lower outlet fluid enthalpy along with the mechanical shaft work delivered to the external load.

Energy Balance: $m_s h_2 = m_s h_3 + W_T$

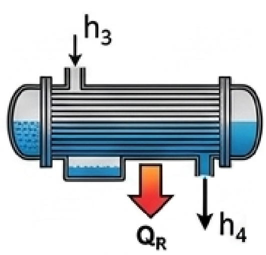
Rearranging the terms to isolate the total power output (W_T) produced by the expanding steam yields:

$$W_T = m_s(h_2 - h_3)$$

Where:

- W_T = Turbine power output (kW or kJ/s)
- m_s = Total mass flow rate of the steam (kg/s)
- h_2 = Enthalpy of the superheated steam entering the turbine (kJ/kg)
- h_3 = Enthalpy of the wet vapor mixture exiting the turbine (kJ/kg)

The Condenser

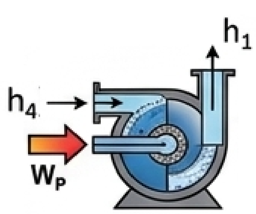
	The low-pressure wet vapor mixture at state 3 enters the condenser, drops its thermal payload, and exits as a saturated liquid at state 4. The energy entering the system is carried by the inlet enthalpy (h_3), while the energy exiting the system leaves via the fluid enthalpy (h_4) and the rejected heat (Q_R).

Energy Balance: $m_s h_3 = m_s h_4 + Q_R$

Isolating the rate of thermal energy rejection (Q_R) gives:

$$Q_R = m_s(h_3 - h_4)$$

The Liquid Feed Pump

	The pump receives a saturated liquid stream at state 4 and compresses it to the high operating pressure at state 1. The energy entering the pump consists of the fluid enthalpy (h_4) combined with the external mechanical work input (W_P) supplied by an electric motor. The exiting energy is entirely contained within the high-pressure fluid enthalpy (h_1).

Energy Balance: $W_P + m_s h_4 = m_s h_1$

Isolating the mechanical work input required by the pump (W_p):

$$W_P = m_s(h_1 - h_4)$$

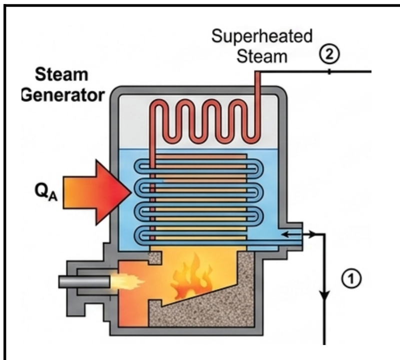
Because liquid water is practically incompressible, its specific volume (v) remains virtually constant throughout the compression process ($v_4 = v_1 = v$). Integrating the steady-flow work relation $W = \int vdP$ provides an alternative method to compute pump work without needing subcooled liquid enthalpy values from state 1:

$$W_P = m_s v_4(P_1 - P_4)$$

Where:

- W_P = Pump power input (kW)
- v_4 = Specific volume of the saturated liquid at the condenser pressure (P_4), in m^3/kg
- P_1 = High operating pressure of the steam generator (Boiler Pressure), in kPa
- P_4 = Low operating pressure of the condenser, in kPa

The Steam Generator (Boiler)



The pressurized subcooled liquid at state 1 enters the steam generator tubes, absorbs heat from external combustion, and exits as superheated steam at state 2. The entering energy consists of the fluid enthalpy (h_1) plus the rate of external heat addition (Q_A). The exiting energy is carried away by the fluid enthalpy (h_2).

Energy Balance: $Q_A + m_s h_1 = m_s h_2$

Isolating the total heat addition rate (Q_A) yields:

$$Q_A = m_s(h_2 - h_1)$$

SYSTEM PERFORMANCE & THERMAL EFFICIENCY

The ultimate metric used to quantify the performance of a power cycle is its **thermal efficiency (η_{th}).** Thermal efficiency is defined by the fundamental engineering principle of maximizing output relative to the input cost:

$$\eta_{th} = \frac{Net\,Useful\,Work\,Output}{Total\,heat\,Input} = \frac{W_{net}}{Q_A}$$

The net work output (W_{net}) produced by the cycle is the difference between the gross power generated by the turbine (W_T) and the power consumed by the liquid feed pump (W_P):

$$W_{net} = W_T - W_P$$

Substituting this into the primary efficiency equation provides:

$$\eta_{th} = \frac{W_T - W_P}{Q_A} x\ 100\%$$

Alternatively, we can evaluate the net work of the entire cycle by applying a global energy balance. Over a complete thermodynamic cycle, the net work output must equal the net heat input:

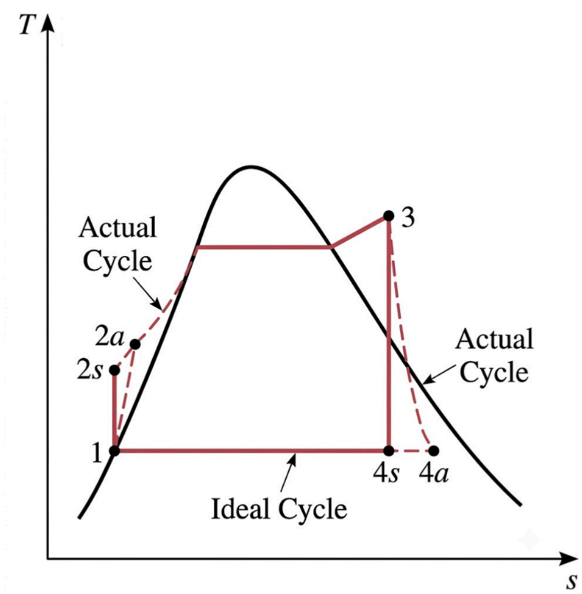
$$W_{net} = Q_A - Q_R$$

Substituting this relationship into the efficiency equation provides an alternative formulation:

$$\eta_{th} = \frac{Q_A - Q_R}{Q_A} x\ 100\% = \left(1 - \frac{Q_R}{Q_A}\right) x\ 100\%$$

Using this equation, engineers can evaluate the efficiency of a power plant by tracking only the heat input at the boiler and the heat rejected at the condenser.

CYCLE IRREVERSIBILITIES



In real-world power plant operations, a steam power cycle deviates from the idealized, frictionless baseline due to various operational factors. Cycle irreversibilities are primarily caused by fluid friction and stray heat losses to the surrounding environment from the cycle components and the pipes that connect those components. As steam flows through connecting pipelines, friction induces a pressure drop between major components, while temperature gradients between the piping walls and the ambient environment lead to unavoidable heat losses. These deviations reduce the overall thermal performance of the facility, meaning actual power plants require a more detailed thermodynamic analysis than an ideal Rankine cycle.

COMPONENT-SPECIFIC REAL LOSSES

When evaluating real power cycles for engineering board exams, the focus shifts from ideal isentropic processes to non-isentropic behaviors within the work-producing and work-consuming components.

Turbine Irreversibilities and Isentropic Efficiency

In an ideal turbine, the expansion process is modeled as perfectly frictionless and adiabatic—meaning it is an isentropic expansion where the entropy remains constant ($s_2 = s_3$). However, inside a real turbine casing, fluid friction acts against the steam as it flows through the nozzle cascades and past the moving turbine blades. This internal fluid friction dissipates a portion of the mechanical energy back into the working fluid as heat, causing the entropy of the fluid to increase as it moves through the turbine stages ($s_{3'} > s_2$).

Because of these frictional losses, a real turbine drops less temperature and enthalpy than an ideal one, resulting in a lower mechanical shaft work output. These internal losses are quantified by the **turbine isentropic efficiency (η_T)**, which evaluates the ratio of the actual work output produced by the turbine to the work that would be generated under ideal, isentropic conditions:

$$\eta_T = \frac{Actual\,Work}{Ideal\,Work} = \frac{h_2 - h_{3'}}{h_2 - h_3}$$

Where:

- h_2 = Enthalpy of the steam entering the turbine (kJ/kg)
- h_3 = Enthalpy at the turbine exit state under ideal, isentropic conditions (kJ/kg)
- $h_{3'}$ = Enthalpy at the turbine exit state under actual, non-isentropic conditions (kJ/kg)

Pump Irreversibilities and Isentropic Efficiency

An ideal liquid feed pump is assumed to compress the fluid from the condenser pressure to the boiler pressure via a frictionless, isentropic path ($s_4 = s_1$). In an actual pump, mechanical and fluid friction losses mean the pump needs more work input to raise the water pressure to the target operating level. This additional friction causes an increase in entropy as the fluid passes through the pump impeller ($s_{1'} > s_4$), meaning the fluid leaves the pump at a slightly higher temperature and enthalpy than it would in an ideal process.

Because a real pump consumes more external mechanical energy than a frictionless one, its efficiency equation is inverted relative to the turbine. The **pump isentropic efficiency (η_P)** is defined as the ratio of the ideal mechanical work required during a frictionless process to the actual mechanical work input consumed by the pump:

$$\eta_P = \frac{Ideal\ Work}{Actual\ Work} = \frac{h_1 - h_4}{h_{1'} - h_4}$$

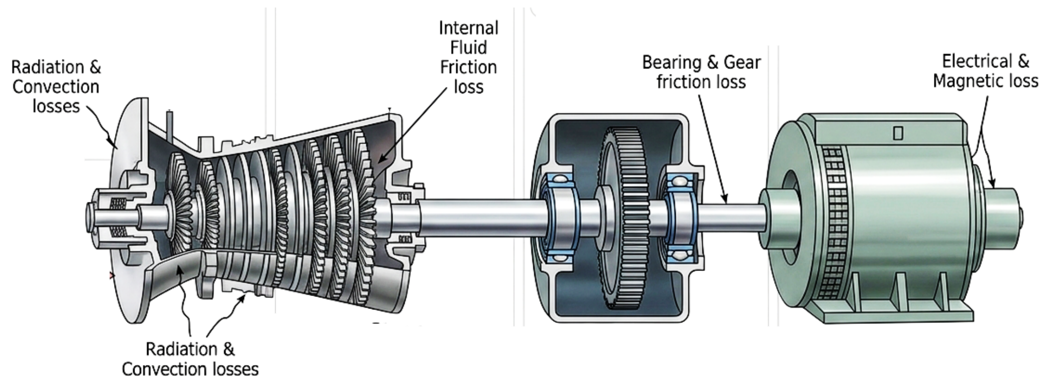
Where:

- h_4 = Enthalpy of the liquid entering the pump (kJ/kg)
- h_1 = Enthalpy at the pump discharge state under ideal, isentropic conditions (kJ/kg)
- $h_{1'}$ = Enthalpy at the pump discharge state under actual, non-isentropic conditions (kJ/kg)

STEAM TURBINE

Mechanical Component Efficiencies (Brake & Generator)

Once energy is successfully extracted by the turbine blades, it must be transferred via the mechanical rotor shaft to an electrical generator.



- **Brake Engine/Mechanical Efficiency (η_M):** Accounts for frictional drag within the rotor bearings and packing glands. It represents the ratio of the net usable mechanical power measured at the shaft coupling—known as **Brake Power (BP)**—to the actual internal fluid work ($W_{T'}$):

$$\eta_M = \frac{BP}{W_{T'}} x\, 100\%$$

- **Generator Efficiency (η_g):** Measures the conversion performance of the generator capsule as it transforms mechanical shaft power (BP) into net electrical output—termed **Electrical Power (EP)**. Frictional windage, copper losses, and core magnetic losses reduce this value:

$$\eta_g = \frac{EP}{BP} x\, 100\%$$

SYSTEMIC THERMAL EFFICIENCY CONCEPTS

Thermal efficiency (η_{th}) measures a system's net work output relative to the total heat input (Q_A) supplied to the steam generator. Depending on which stage of the power delivery chain is selected as the output metric, three distinct thermal efficiencies are utilized:

Turbine Thermal Efficiency

This represents the thermal performance based strictly on the energy converted by the expanding fluid inside the turbine casing, prior to any external shaft or electrical losses:

$$\eta_{th,\,turbine} = \frac{W_T}{Q_A}\ x\ 100\% = \frac{W_T}{m_s\left(h_2 - h_3\right)}\ x\ 100\%$$

3Brake Thermal Efficiency

This represents the net thermal efficiency referenced to the usable mechanical work output available at the rotating shaft coupling (BP), accounting for both internal fluid losses and mechanical bearing friction:

$$\eta_{th,\,brake} = \frac{BP}{Q_A}\ x\ 100\% = \frac{BP}{m_s\left(h_2 - h_3\right)}\ x\ 100\%$$

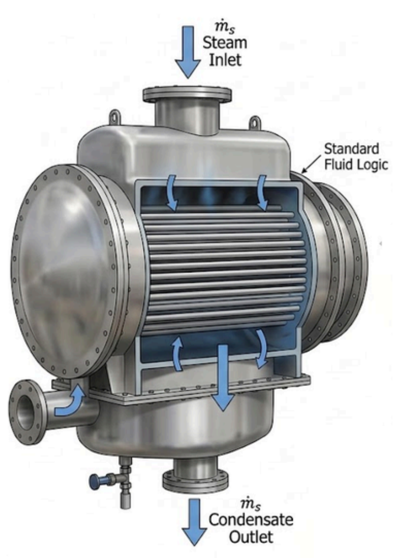
Combined Thermal Efficiency

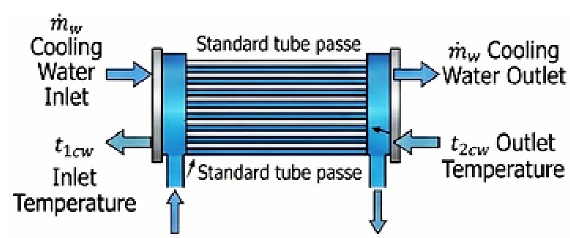
This evaluates the overall thermal efficiency of the power plant, measuring the final utility electrical output (EP) against the original fuel energy added to the steam generator (Q_A):

$$\eta_{th,\,combined} = \frac{EP}{Q_A}\ x\ 100\% = \frac{EP}{m_s\left(h_2 - h_3\right)}\ x\ 100\%$$

CONDENSER ANALYSIS & COOLING WATER REQUIREMENTS

The condenser acts as a constant-pressure heat exchanger that rejects latent heat from the turbine exhaust steam to an isolated external cooling water circuit.





Applying a standard energy balance across the condenser boundaries shows that the thermal energy rejected by the condensing steam (Q_R) must equal the thermal energy absorbed by the external cooling water circuit ($Q_{Cooling\ Water}$):

$$Q_R = Q_{Cooling\ Water}$$

$$m_s\left(h_3 - h_{f4}\right) = m_w C_{Pw}\left(t_2 - t_1\right)$$

Isolating the required cooling water mass flow rate (m_w) yields:

$$m_w = \frac{m_s\left(h_3 - h_{f4}\right)}{C_{Pw}\left(t_2 - t_1\right)}$$

Where:

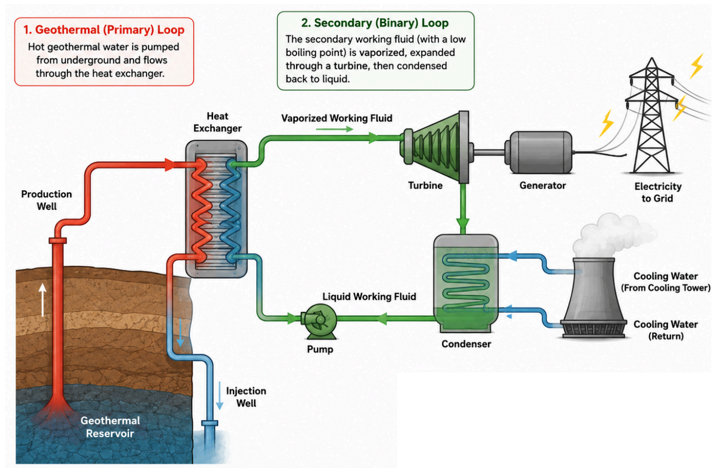
- m_w = Mass flow rate of the required cooling water (kg/s)
- m_s = Mass flow rate of the system steam loop (kg/s)
- h_3 = Enthalpy of the steam entering the condenser inlet shell (kJ/kg)
- h_{f4} = Enthalpy of the saturated liquid condensate exiting the hotwell (kJ/kg)
- t_2 , t_1 = Inlet and outlet temperatures of the cooling water stream respectively (°C or K)
- C_{Pw} = Specific heat capacity of liquid water, taken as a standard constant value of $4.187 \frac{kJ}{kg-K}$

GEOTHERMAL POWER PLANT

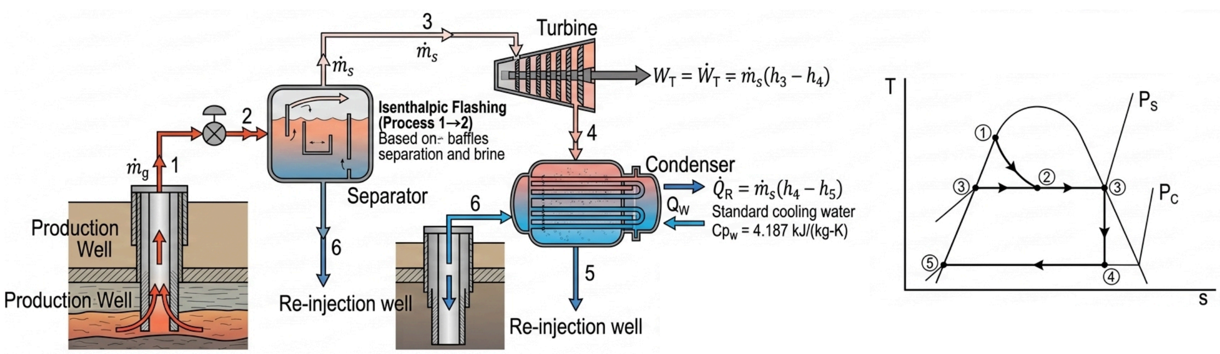
Geothermal energy represents the thermal energy produced and sustained by the vast temperature gradients stored within the Earth's interior. In engineering applications, this subterranean heat is harnessed to generate massive amounts of utility-scale power by utilizing water and steam as naturally occurring thermodynamic working fluids. Depending on the geological state of the underground resource (whether it yields pure vapor, a high-pressure liquid mixture, or lower-temperature water), geothermal power plants are broadly categorized into three fundamental designs:

Dry Steam (Vapor-Dominated Systems)

Dry steam power plants are deployed when the underground reservoir natively produces superheated or high-quality dry saturated steam. These systems draw steam directly from underground production wells and pipe it directly into a steam turbine without requiring complex phase-separation equipment. Because the fluid is already completely in the vapor phase, it represents the simplest and historically earliest form of geothermal power generation.



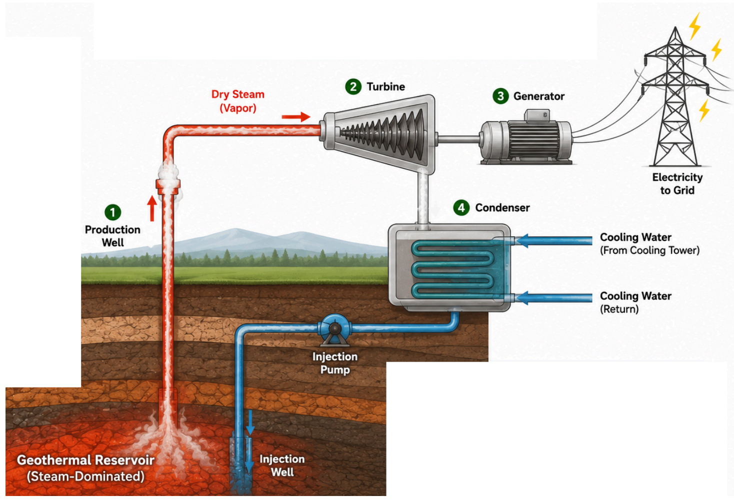
Flash Steam (Liquid-Dominated Systems)



Flash steam systems are the most prevalent type of geothermal power plants encountered in industrial operations. These installations draw high-pressure, high-temperature hydrothermal fluids from underground wells, which are predominantly in the liquid domain. As the pressurized fluid ascends the production well, it undergoes a sudden pressure drop, causing a portion of the liquid to rapidly "flash" into steam. The resulting two-phase mixture must pass through a specialized component known as a **separator** before it can be utilized. The separator isolates the pure vapor from the residual liquid brine, directing the dry steam to the turbine blades while routing the remaining liquid back into the earth via re-injection wells.

Binary Cycle (Liquid-Dominated Systems)

Binary cycle power plants are engineered to exploit lower-temperature geothermal reservoirs that cannot efficiently flash into high-pressure steam. Instead of routing the geothermal fluid directly into a turbine, a binary system utilizes a closed secondary loop. The hot geothermal water passes through a heat exchanger where it transfers its thermal energy to a secondary working fluid that possesses a much lower boiling point than water (such as an organic hydrocarbon like isobutane or pentane). This organic working fluid is completely vaporized within the heat exchanger and subsequently expanded through a specialized organic Rankine cycle turbine to generate power.



THERMODYNAMIC ANALYSIS OF A FLASH STEAM PLANT

Focusing strictly on the standard single-flash configuration illustrated in the schematic, the thermodynamic cycle transitions through a sequence of localized processes from the wellhead to the re-injection matrix:

Process 1 to 2: The Throttling (Flashing) Process

The hot geothermal fluid leaves the production wellhead at state 1 under intense pressure. To induce vaporization, the fluid undergoes a pressure-reducing restriction, acting as a classic **throttling process**. In thermodynamics, an ideal throttling process is modeled as strictly **isenthalpic**, meaning the specific enthalpy remains constant across the restriction ($h_1 = h_2$). Because the lower pressure drops the fluid into the wet vapor region under the dome, the mixture separates into a two-phase state characterized by a specific steam quality (x). The governing enthalpy balance is expressed as:

$$h_1 = h_2 = h_{f2} + x_2 h_{fg2}$$

Where:

- h_1 = Initial enthalpy of the geothermal fluid at the wellhead (kJ/kg)
- h_2 = Enthalpy of the two-phase mixture entering the separator (kJ/kg)
- h_{f2} = Enthalpy of saturated liquid at the separator flashing pressure (kJ/kg)
- h_{fg2} = Latent heat of vaporization at the separator flashing pressure (kJ/kg)
- x_2 = Steam quality after the throttling process (expressed as a decimal fraction)

Separator Mass Flow Interactions

Once the two-phase fluid enters the separator casing at state 2, it is mechanically split based on phase density. The saturated liquid component sinks to the bottom and emerges as liquid brine at state 6, which is promptly diverted to a re-injection well. Meanwhile, the saturated vapor exits the upper manifold as dry steam at state 3 ($h_3 = h_{g2}$). The mass flow rate of this pure dry steam (m_s) is directly proportional to the total mass flow rate of the ground water mixture (m_g) entering from the well, scaled by the dryness fraction (x) achieved during the flash:

$$m_s = xm_g$$

Process 3 to 4: Turbine Power Production

The separated dry steam at state 3 is channeled into the turbine assembly, expanding down to the lower condenser pressure at state 4 to produce mechanical shaft work (W_T). Neglecting kinetic energy shifts and stray heat losses, the steady-flow energy equation yields the power output equation based on the steam loop mass flow:

$$W_T = m_s(h_3 - h_4)$$

Where:

- W_T = Power generated by the turbine (kW or MW)
- h_3 = Enthalpy of dry saturated steam entering the turbine inlet (kJ/kg)
- h_4 = Enthalpy of the wet steam mixture exhausting into the condenser chamber (kJ/kg)

Process 4 to 5: Condenser Thermal Rejection

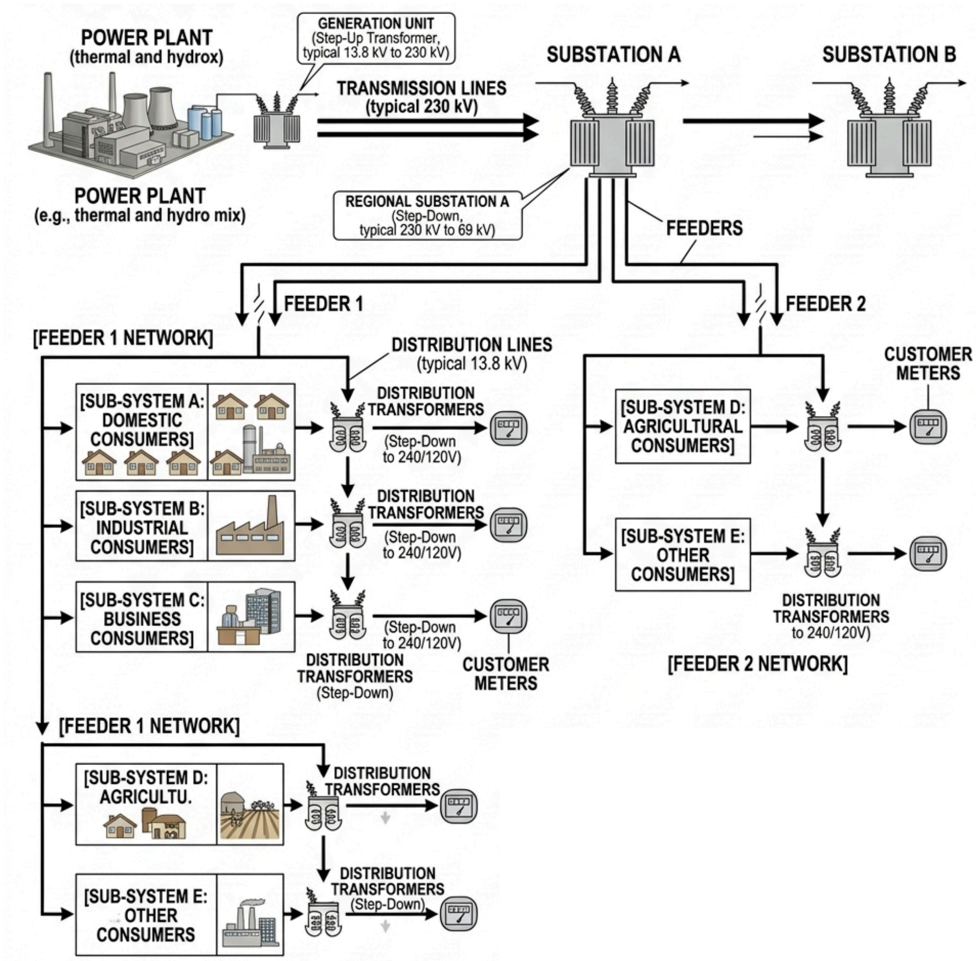
The low-pressure wet steam exiting the turbine at state 4 enters the condenser shell, where it is completely condensed into liquid form by transferring its remaining thermal energy to an external cooling medium. The saturated liquid condensate exits the hotwell at state 5 and is piped down a re-injection well back into the subterranean geological formation to maintain reservoir pressure equilibrium. The heat rejection rate (Q_R) within the condenser shell is defined as:

$$Q_R = m_s(h_4 - h_5)$$

Where h_5 represents the specific enthalpy of the liquid condensate leaving the condenser unit (kJ/kg)

VARIABLE LOAD ANALYSIS

An electrical power system is structured to transport energy efficiently from the point of generation to localized consumer loads. Under this framework, electrical energy is produced at the **Power Plant** and stepped up to high voltages for transmission. It is then received by regional **Substations**, which reduce the voltage to intermediate levels.



From these substations, localized distribution lines—known as **Feeders**—carry the power to various areas. The feeders connect to **Distribution Transformers**, which step the voltage down further to safe operating levels for final consumers, including domestic, industrial, and business customers.

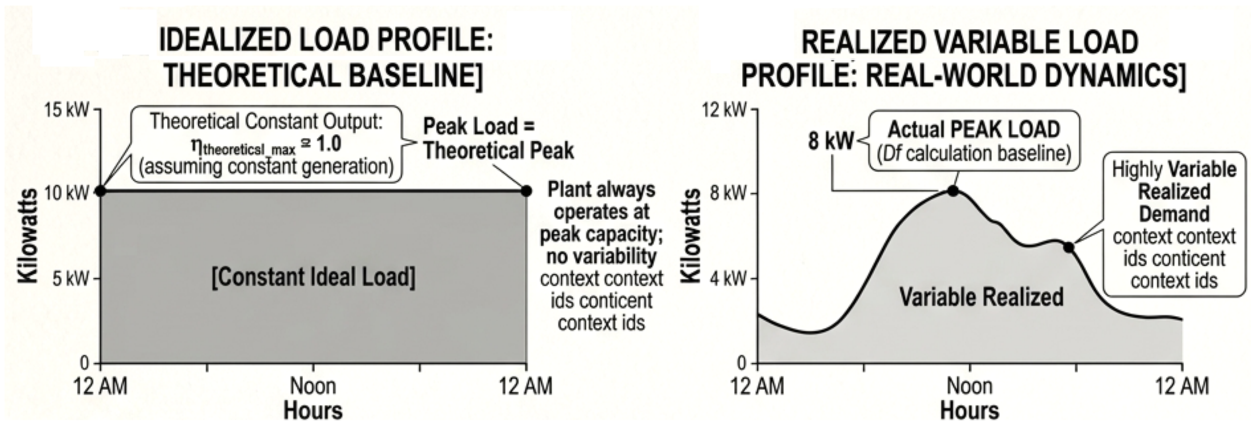
The **Connected Load** of any specific feeder or subsystem represents the continuous cumulative sum of the power ratings of all load-consuming apparatus (appliances, motors, lighting, etc.) connected to the system. For example, the total connected load of Feeder No. 1 is simply the sum of the individual connected loads of its downstream branches or consumer categories (e.g., a + b + c).

LOAD CURVES AND CONSUMPTION PATTERNS

A **Load Curve** is a graphical representation that illustrates consumer load demand variations with respect to time. In engineering practice, the total area under a load curve is mathematically equal to the total electrical energy generated during that period. Load curves are typically assessed over three distinct intervals:

- **Daily Load Curve:** Constructed using an hourly time interval over a standard 24-hour cycle.
- **Monthly Load Curve:** Constructed using daily averages or daily peaks plotted over the days of the month.
- **Annual Load Curve:** Constructed using monthly data plotted over a 12-month period.

Ideal Load vs. Realized Load



- **Ideal Load:** Represents a theoretically constant, flat load curve where demand remains completely uniform over its operating period. In this scenario, the system operates at maximum efficiency because there are no sudden power surges or idle periods.
- **Realized Load:** Reflects actual consumer behavior. It is characterized by notable peaks (usually in the evening or mid-day) and deep troughs (during late-night or early-morning hours), showing highly variable load demand. Managing a realized load curve requires dynamic generation control to match fluctuating demand in real time.

To design, operate, and analyze variable load systems, engineers utilize several dimensionless factors and metrics. These values evaluate system efficiency, capacity margins, and equipment utilization.

Average Load

The average demand of a power system over a specified timeframe:

$$\text{Average Load} = \frac{Total\ Energy\ Generated\ in\ Kilowatt{-}Hours\ (kWh)}{Number\ of\ Hours\ Considered}$$

Load Factor

The ratio of the average load to the peak load over a specific period:

$$\text{Load Factor} = \frac{Average\ Load}{Peak\ Load}$$

Note: Peak load is defined as the maximum load consumed or produced by a system in a stated period of time. A high load factor is highly desirable because it indicates a relatively stable demand, which minimizes the need for expensive peaking plants.

Demand Factor

The ratio of the actual maximum power demand of a system to its total connected load:

$$\text{Demand Factor} = \frac{Actual\ Maximum\ Demand}{Total\ Connected\ Load}$$

Because all connected appliances are rarely turned on simultaneously, the demand factor is typically less than 1.0 (or 100%).

Diversity Factor

The ratio of the sum of individual maximum demands of the various subdivisions of a system to the maximum demand of the entire system:

$$\text{Diversity Factor} = \frac{\Sigma(Individual\ Maximum\ Demand)}{Maximum\ Demand\ of\ the\ System}$$

Note: The "system" analyzed can represent a group of customers served by a single transformer, or a group of transformers served by a single feeder line. Because individual peaks occur at different times of the day, the sum of individual peaks is always greater than the coincident system peak. Therefore, ***the Diversity Factor is always greater than 1.0****. A higher diversity factor is highly desirable.*

Capacity Factor

The ratio of the average load on a machine, piece of equipment, or entire plant to its rated nameplate capacity:

$$\text{Capacity Factor} = \frac{Average\ Load\ on\ Equipment\ for\ a\ Period\ of\ Time}{Rating\ of\ the\ Machine\ or\ Equipment}$$

Plant Use Factor (or Output Factor)

This metric measures the actual energy produced during a given period relative to the maximum energy the plant could have produced during its actual running hours:

$$\text{Plant Use Factor} = \frac{Annual\ Kilowatt-Hours\ Production}{(Kilowatt\ of\ Capacity)(Number\ of\ Hours\ the\ Plant\ was\ in\ Operation)}$$

Utilization Factor

Evaluates how intensively the maximum system demand utilizes the rated capacity of the installation:

Utilization Factor = $\frac{Maximum\ Demand}{Rated\ Capacity}$

Operation Factor

Measures the duration of actual service of a machine or system relative to the total elapsed period of time considered:

Operation Factor = $\frac{Duration\ of\ Actual\ Service\ of\ a\ Machine\ or\ Equipment}{Total\ Duration\ of\ the\ Period\ of\ Time\ Considered}$

POWER CLASSIFICATIONS AND STATION TERMINOLOGY

For both hydro-electric and thermal power plant analysis, engineers use specific terms to categorize capacity types, generation reserves, and auxiliary demands.

System Reserve Definitions

- **Reserved Load:** The safety margin between a plant's installed capacity and its peak load demand:

  Reserved Load = Plant Capacity - Peak Load

- **Reserved Equipment:** The installed plant equipment available in excess of that required to carry the current peak load.
- **Cold Reserve:** Generating capacity that is fully available for service but is not currently operating (i.e., shut down or cold).
- **Hot Reserve:** Generating capacity that is fired up and in operation but is not currently delivering power to the grid (running idle, ready to be loaded).
- **Spinning Reserve:** Generating capacity that is connected directly to the bus and is synchronized, rotating, and ready to immediately pick up load.
- **System Reserve:** The extra capacity in generating equipment and electrical conductors installed throughout the power system in excess of the peak load requirement.
- **Spare Equipment:** Completely assembled machines or individual parts kept on hand to facilitate rapid repairs or replacements.

Power Type Classifications

- **Firm Power:** The high-reliability power intended to be always available to consumers, even during emergency conditions.
- **Prime Power:** The maximum potential power (chemical, mechanical, or hydraulic) constantly available for transformation into electricity.
- **Dump Power:** Excess power (commonly generated by run-of-the-river hydroelectric plants) produced by surplus water flow in excess of normal consumer load requirements.

Plant Auxiliary Definitions

- **Generating Station Auxiliary Power:** The electrical power consumed within the power plant itself to run auxiliary machines (such as feedwater pumps, cooling fans, and coal pulverizers).
- **House Turbine:** A dedicated, auxiliary turbine installed inside the power plant solely to provide a reliable source of auxiliary power to internal systems.
- **Run-of-the-River Station:** A type of hydro-electric generating station that utilizes natural river streamflow directly for power generation without using water storage reservoirs.