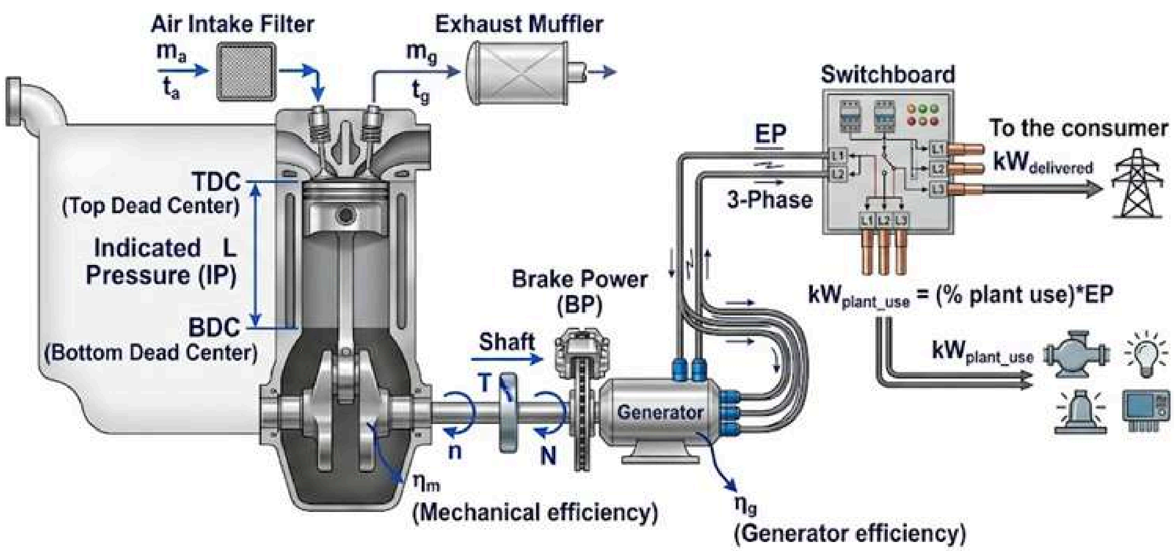
Diesel Electric Power Plant

A Diesel Electric Power Plant represents a complete thermodynamic-mechanical-electrical energy conversion train. The process begins inside the cylinder volume during the combustion phase, where the chemical energy bound within the fuel oil is rapidly liberated as high-temperature thermal energy. This thermal energy causes a dramatic expansion of the gas column, forcing the piston downward from Top Dead Center (TDC) to Bottom Dead Center (BDC) through its stroke length (L). This volumetric displacement converts expansion work into linear reciprocating mechanical power.



The energy train then experiences its first major thermodynamic transition at the crankshaft. The reciprocating motion of the piston-connecting rod assembly is translated into rotational shaft work. This rotational kinetic energy is subsequently directed to the rotor shaft of a coupled synchronous electrical generator (alternator). The rotating magnetic fields within the stator windings induce an electromotive force, transforming rotational mechanical power into usable alternating current (AC) electricity. This electricity is routed through a centralized switchboard system to internal plant auxiliaries and out to the consumer grid.

Power Definitions

To evaluate the overall performance of a diesel station, an engineer must track power across three distinct stages of the energy cascade:

- **Indicated Power (IP):** This is the gross mechanical power developed purely within the combustion chamber. It is calculated from the mean effective pressure acting directly on the piston face and represents the total power generated before accounting for any internal physical losses.

- **Brake Power (BP):** This is the net mechanical power available at the engine crankshaft. It represents the actual mechanical output delivered to the generator shaft after deducting the power required to overcome the engine's internal mechanical constraints.
- **Electrical Power (EP or P_e):** This is the raw electrical power generated by the alternator. It represents the output after the mechanical shaft power experiences electrical, magnetic, and windage losses within the generator core.

Fuel Energy Input Rate

The total thermal power input liberated by the combustion of fuel per unit time is a function of the mass flow rate of the fuel and its specific heating value:

$$Q_{in} = m_f q_f$$

Where m_f is the fuel mass flow rate in kilograms per second (kg/s) and q_f is the higher or lower heating value of the fuel in kiloJoules per kilogram (kJ/kg), yielding an input rate in kiloWatts (kW).

Mechanical Efficiency (η_M) and Friction Losses

As the engine operates, a portion of the indicated power is consumed to overcome parasitic mechanical losses. These losses include boundary friction between the piston rings and cylinder walls, bearing friction along the crankshaft, and the work required to drive vital integrated auxiliaries such as the oil pump, water jacket pump, and fuel injection system. This collective loss is termed **Friction Power (FP)**:

$$BP = IP - FP$$

The mechanical efficiency (η_M) quantifies the percentage of combustion power that successfully reaches the engine crankshaft:

$$\eta_M = \frac{BP}{IP} = 1 - \frac{FP}{IP}$$

Generator Efficiency (η_g) and Electrical Conversion

When the brake power is transferred to the generator shaft, further energy degradation occurs due to I^2R copper losses in the stator and rotor windings, iron core hysteresis, eddy currents, and mechanical windage losses within the housing. The power output available at the generator terminals (EP) is written as:

$$EP = BP - \text{Generator Losses}$$

The generator efficiency (η_g) expresses the effectiveness of this mechanical-to-electrical transformation:

$$\eta_g = \frac{EP}{BP} = 1 - \frac{Generator\ Loss}{BP}$$

Combined Mechanical-Electrical Efficiency (η_{ME})

The total efficiency of the combined engine-generator set, excluding external plant accessories, is the product of the mechanical and electrical efficiencies. This parameter relates the final electrical output directly back to the initial indicated power developed inside the cylinders:

$$\eta_{ME} = \eta_M \eta_g = \left(\frac{BP}{IP}\right)\left(\frac{EP}{BP}\right) = \frac{EP}{IP}$$

Net Station Output and Plant Auxiliary Consumption

A functioning power station requires a portion of its own generated electricity to run peripheral support systems, including station lighting, fuel transfer pumps, cooling tower fans, and control room automation. This parasitic draw is termed **Plant Use ($kW_{plant\ use}$)** and is typically modeled as a fixed percentage of the gross generator output (EP):

$$kW_{plant\ us} = (\%\ plant\ use)(EP)$$

The actual net electrical power delivered to the consumer grid ($kW_{delivered}$) is the remainder of the generated power after deducting this internal load:

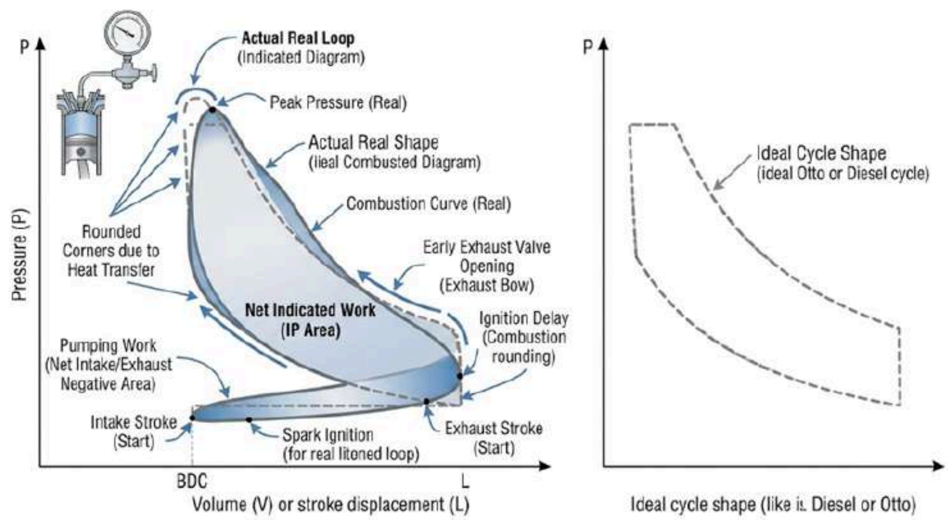
$$kW_{delivered} = EP - kW_{plant\ use}$$

$$kW_{delivered} = EP\ (1 - \%\ plant\ use)$$

Indicated Power

The analysis of an internal combustion engine, such as a diesel electric prime mover, is divided into its theoretical cycle limits and its actual physical performance. The **Ideal Power** (often called the theoretical network network rate, W_{net}) represents the thermodynamic upper boundary dictated by the air-standard Diesel cycle. It depends entirely on the initial intake suction pressure (P_1), the compression ratio (r_k), the cut-off ratio (r_c), and the specific heat ratio (k). It establishes the absolute maximum power an engine could produce if there were zero thermal, fluid, or mechanical losses.

In contrast, **Indicated Power (IP)** is the actual mechanical power developed within the engine cylinders, captured directly from the real gas expansion profile. Because real-world combustion experiences chemical dissociation, variable specific heats, heat transfer through the cylinder walls, and valve pumping losses, the actual pressure profile deviates significantly from the ideal cycle shape.



To capture this real performance, an engine indicator mechanism measures the changing pressure inside the firing cylinder and plots it against the stroke displacement, producing a physical geometric loop known as an **indicator card**.

The Kinematics of Piston Displacement

To translate pressure inside the cylinder into an absolute rate of power work output, engineers must quantify the engine's volumetric displacement capacity per unit time. The engine kinematics depend on physical geometric properties: the bore diameter (D), the stroke length (L), the number of cylinders (c), and whether the engine acts on one or both sides of the piston face (a, the number of actings).

The velocity of the piston is not uniform due to the angularity of the connecting rod assembly. Therefore, engineers utilize the **mean piston speed (v_P)**, which represents the average total linear distance covered by the piston face per unit time. Because the piston travels down and up (two lengths of the stroke, 2L) for every single revolution of the crankshaft, the mean piston speed is a direct function of rotational speed (n). This geometric relationship allows the volumetric flow rate of the engine to be elegantly calculated by combining the physical cross-sectional piston area with its linear structural speed parameters.

Ideal Mean Effective Pressure (P_m) & Ideal Power

The mean effective pressure is a fictitious, constant pressure that, if exerted on the piston face throughout the entire power stroke, would produce the exact same net work output as the variable pressure profile of the actual cycle. For an ideal air-standard Diesel cycle, this parameter is calculated as:

$$P_m = P_1 r_k \left[\frac{k\, r_k^{\,k-1}\left(r_c - 1\right) - \left(r_c^{\,k} - 1\right)}{\left(r_k - 1\right)(k-1)} \right]$$

The compression ratio (r_k) and cut-off ratio (r_C) are defined based on the clearance percentage fraction (c_l):

$$r_k = \frac{c_l + 1}{c_l}$$

$$r_c = \frac{c_l + c_o}{c_l}$$

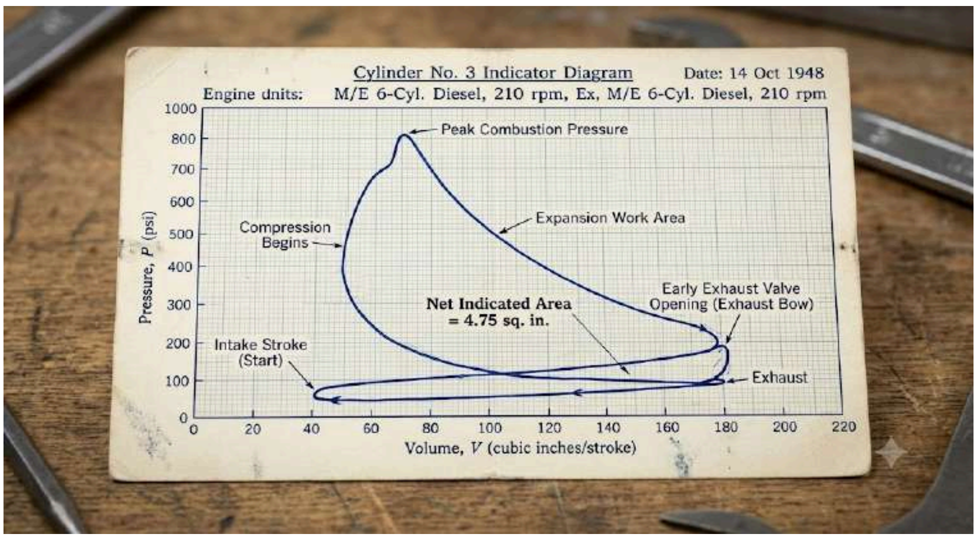
Once $\mathbf{P_m}$ is established, the ideal net power output (W_{net}) is the product of this mean pressure and the engine's volumetric displacement rate (V_D):

$$W_{net} = P_m V_D$$

Indicated Mean Effective Pressure (P_{mi}) from Indicator Cards

For a running engine, the actual indicated mean effective pressure ($\mathbf{P_{mi}}$) is extracted directly from the indicator card diagram. The area enclosed by the card curve is measured (typically via a planimeter) and divided by the physical length of the diagram card to find its average height. This height is then scaled by the stiffness constant of the indicator mechanism spring:

$$P_{mi} = \frac{Area\ of\ Card\ (A_C)\ x\ Scale\ of\ Indicator\ Spring\ \left(S_c\right)}{Length\ of\ Indicator\ Card\ \left(L_c\right)}$$



Once P_{mi} is determined, the actual **Indicated Power (IP)** generated within the combustion chambers is calculated as:

$$IP = P_{mi} V_D$$

Engine Volumetric Displacement Rate (V_D)

The total volume swept by the engine pistons per second is derived by calculating the cross-sectional area of a single cylinder, multiplying it by the stroke length to find the single swept volume, and scaling it across all active firing cycles per second:

$$V_D = \left[\frac{\pi D^2}{4}\right] L N_S$$

Where N_S represents the total active cycles per second for the entire multi-cylinder engine. Accounting for the number of cylinders (c), whether the engine is single or double acting (a), the rotational speed in rpm (n), and the stroke cycle index (s = 4 for a 4-stroke engine; s = 2 for a 2-stroke engine), the equation expands to:

$$V_D = \left[\frac{\pi D^2}{4}\right] L \left[\frac{c \cdot a\left(\frac{n}{60}\right) \cdot 2}{s}\right]$$

Linear Piston Speed (v_P) Integration

The mean linear speed of the piston moving within the cylinder walls is written as:

$$v_P = \frac{2Ln}{60}$$

By substituting the mean piston speed back into the displacement formulation, the volumetric displacement rate can also be computed as a direct function of the piston's linear velocity and the cylinder configuration:

$$V_D = \left[\frac{\pi D^2}{4}\right] v_P \left[\frac{c \cdot a}{s}\right]$$

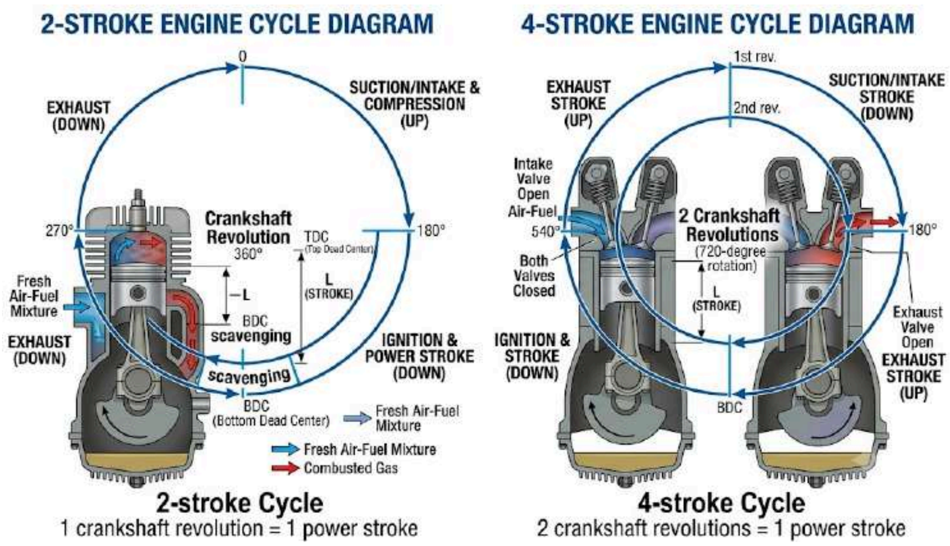
Synchronous Speed of Electrical Generation (n)

In a diesel electric power station, the engine crankshaft is directly coupled to an alternator. To deliver electrical current at a stable target frequency (f), the engine must be governed to run precisely at its required synchronous rotational speed (n), which depends on the number of magnetic poles (P) built into the generator stator:

$$n = \frac{120\,f}{P}$$

Mechanics of 2-Stroke vs. 4-Stroke Engine Cycles

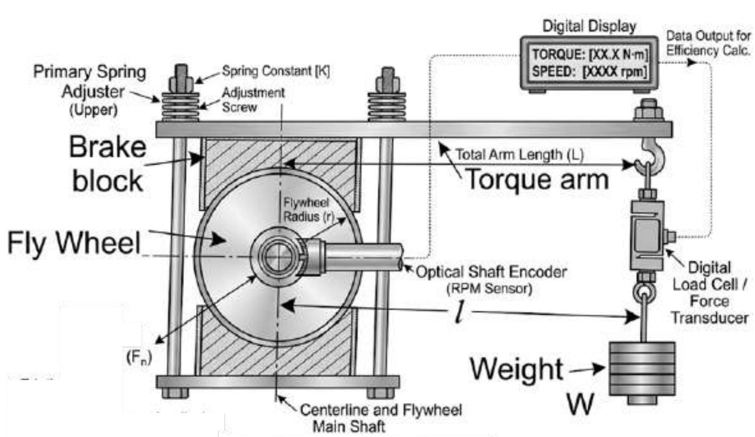
The operation of internal combustion engines is defined by how mechanical strokes relate to thermodynamic power delivery. A **stroke** is the single linear transit of the piston from one dead-center limit to another. In a **2-stroke cycle**, the entire thermodynamic sequence (suction, compression, expansion, and exhaust) is completed within a single revolution of the crankshaft (360°). This is accomplished by using cylinder ports rather than mechanical valves, allowing fresh air to enter while exhaust gases leave at the bottom of the power stroke. Consequently, every single downward stroke functions as a power stroke.



Conversely, a **4-stroke cycle** distributes these phases across two full revolutions of the crankshaft (720°). The engine dedicates individual strokes exclusively to mechanical tasks: the first downward stroke draws in air, the first upward stroke compresses the charge, the second downward stroke executes the thermal power expansion, and the final upward stroke expels the combustion products. Understanding this kinematics is crucial when sizing multi-cylinder power plant engines, as a 4-stroke configuration cuts the structural frequency of power pulses in half relative to a 2-stroke engine turning at identical rotational speeds.

Brake Dynamometry

While indicated power represents the raw energy developed inside the cylinder, engineers require a physical measurement of the useful work available at the engine crankshaft. This is known as **Brake Power (BP)**. To determine this value experimentally without destroying the engine, a loading device called a **dynamometer** is coupled to the rotating output shaft to apply a controlled braking torque. For low-speed industrial engines, a mechanical **Prony Brake** assembly is historically utilized, whereas high-speed modern engines employ fluid or electrical eddy-current dynamometers.



The mechanical Prony brake operates by clamping adjustable friction brake shoes directly around a water-cooled engine flywheel. As the flywheel rotates at speed n, the friction force (F_W) at the flywheel radius (R_W) attempts to rotate the entire brake arm structure. This rotation is constrained by a load cell or weighing scale positioned at a fixed arm length (L_{arm}). The scale registers a raw force that must be corrected for the dead structural weight of the overhanging arm assembly (known as the tare weight, T_W) to isolate the true net force (P_n) generated by engine torque.

Brake Power (BP)

Brake power can be evaluated either through fluid volumetric pressure mapping or directly via rotational torque dynamics:

$$BP = P_{mB} V_D$$

$$BP = \frac{2\pi T_B n}{60}$$

Where P_{mB} is the brake mean effective pressure (kPa), V_D is the volumetric displacement rate (m^3/s), n is the rotational speed (rpm), and T_B is the brake torque expressed in standard Newton-meters (N-m), yielding net Brake Power directly in kiloWatts (kW).

Mechanical Prony Brake Torque Balance

The torque exerted by the engine crankshaft is equal to the product of the internal friction forces acting at the flywheel radius, which perfectly balances the external torque measured by the scale assembly:

$$T_B = F_W R_W = P_n L_{arm}$$

The net driving force (P_n) is determined by subtracting the structural tare weight (TW) of the brake arm from the gross weight (GW) recorded on the weighing scale indicator:

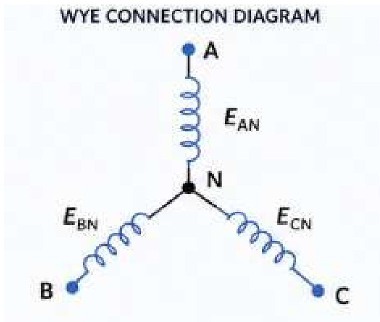
$$P_n = GW - TW$$

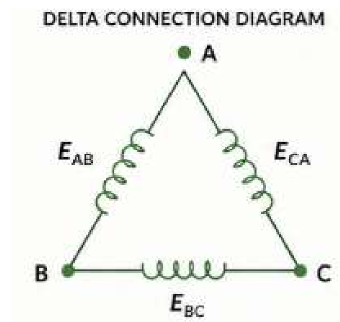
$$T_B = (GW - TW)\, L_{arm}$$

Three-Phase Electrical System Layout

Once the mechanical power is transferred into the alternator core, it is converted into a three-phase electrical output. Industrial generators arrange their internal stator windings into one of two configurations: **Wye (Y-connected)** or **Delta (Δ-connected)**.

- In a **Wye connection**, the three wind ends meet at a common neutral point. This configuration splits the potential fields, making the line-to-line voltage (E_L) higher than the individual phase winding voltage (E_p) by a factor of $\sqrt{3}$, while the current passing out through the lines (I_L) matches the phase current (I_p) identically.





- In a **Delta connection**, the phases are wired end-to-end in a closed loop. Here, the line voltage matches the phase voltage, but the current splits at the nodes, causing the line current to exceed the individual phase winding current by a factor of $\sqrt{3}$,.

To properly track real energy consumption and evaluate power station efficiency, electrical engineers monitor the **Power Triangle**. Alternating current systems carry an **Apparent Power (S, measured in kVA)** which represents the absolute total vector capacity flowing through the lines. This apparent power splits into two components: **True Power (EP or P, measured in kW)**, which does the actual work, and **Reactive Power (Q, measured in kVAR)**, which sustains the alternating magnetic fields inside transformers and motor windings. The ratio of true power to apparent power is defined as the **Power Factor (cos θ)**. A lower power factor means a higher current must be drawn to deliver the same amount of useful work, which increases resistive line losses throughout the power plant.

Three-Phase Electrical Power Metrology (Volt-Meter Method)

Regardless of whether the alternator stator is configured in a Wye or Delta network, the total three-phase real electrical power (EP) delivered at the generator terminals depends on the line-to-line metrics:

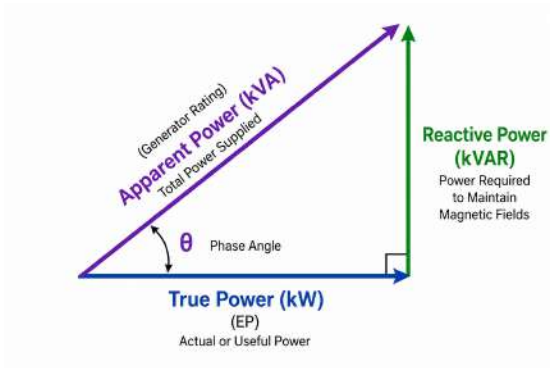
EP = $\sqrt{3}\, E_L I_L \cos\theta$

Where E_L is the line-to-line voltage (kV), I_L is the line current (Amperes), and $\cos\theta$ is the operating power factor.

- For a **Wye (Y) connection**: $I_L = I_p$ and $E_L = \sqrt{3}\, E_p$
- For a **Delta (Δ) connection**: $E_L = E_p$ and $I_L = \sqrt{3}\, I_p$

Note: If a three-phase question does not explicitly state the winding configuration of the alternator, you must default to a **Wye (Y) connection** as the standard design basis.

Electrical Power Triangle Formulations



The geometric relationship between true power, reactive power, and total vector apparent power is governed by trigonometry:

$$Apparent\ Power(S) = \sqrt{3}\, E_L I_L = 3E_p I_p \quad ; \quad \text{kVA}$$

$$True\ Power(EP) = S \cdot cos\theta \quad ; \quad \text{kW}$$

$$Reactive\ Power(Q) = S \cdot sin\theta \quad ; \quad \text{kVAR}$$

$$cos\theta = \frac{True\ Power(EP),\ kW}{Apparent\ Power(S),\ kVA}$$

Note: If the operating power factor ($cos\theta$) of the generator or connected load is completely unstated in the problem text, use the standard default engineering value of **85% (0.85 lagging)**.

Integrated Time-Average Metrology (kW-hr Method)

When evaluating power plant performance over an extended operational window during an engine test, the instantaneous real electrical power (EP) can be calculated by monitoring the plant's integrating watt-hour meters over a measured test duration:

$$EP = \frac{Reading_{END} - Reading_{START}}{Duration\ of\ Test\ (\Delta t)}$$

Where the numerator readings are taken in kiloWatt-hours (kW-hr) and the test duration Δt is input in hours (hr), yielding the average electrical power output in kiloWatts (kW).

Thermal Efficiency (e)

This represents the ratio of useful power output to the total chemical heat input rate liberated by fuel combustion. As energy passes from the cylinder gas column (IP) to the crankshaft (BP), then to the generator terminals (EP), and finally past the plant's auxiliary loads ($kW_{delivered}$), structural losses accumulate. Consequently, thermal efficiency decreases at each subsequent stage of the energy train.

- **Indicated Thermal Efficiency:** $e_i = \frac{IP}{Q_{in}}$ x 100%
- **Brake Thermal Efficiency:** $e_B = \frac{BP}{Q_{in}}$ x 100%
- **Combined Thermal Efficiency:** $e_C = \frac{EP}{Q_{in}}$ x 100%
- **Plant Net Thermal Efficiency:** $e_p = \frac{kW_{delivered}}{Q_{in}}$ x 100%

Engine Efficiency (η)

Often called the *relative efficiency*, this parameter compares the actual mechanical or electrical power developed by the engine to the theoretical net power (W_{net}) calculated from an ideal, air-standard thermodynamic Diesel cycle. It quantifies how successfully the actual physical engine minimizes losses like real-world fluid friction, heat transfer through the block, and incomplete combustion relative to its ideal thermodynamic limit.

- **Indicated Thermal Efficiency:** $\eta_i = \frac{IP}{W_{net}}$ x 100%
- **Brake Thermal Efficiency:** $\eta_B = \frac{BP}{W_{net}}$ x 100%
- **Combined Thermal Efficiency:** $\eta_C = \frac{EP}{W_{net}}$ x 100%

Volumetric Efficiency (η_v)

This parameter measures the breathing capacity of a naturally aspirated or supercharged engine. It is the ratio of the actual volume flow rate of ambient air drawn into the cylinder intake system to the physical volume displacement rate of the moving pistons. A high volumetric efficiency ensures that sufficient oxygen is trapped within the cylinder to completely combust the injected fuel.

$$\eta_v = \frac{V_a}{V_D} \text{ x } 100\%$$

Specific Fuel Consumption

Evaluating the commercial viability and thermodynamic performance of a diesel electric power station requires measuring how efficiently fuel is converted into power. **Specific Fuel Consumption (SFC)** defines the mass flow rate of fuel consumed per unit of power produced. Because power is monitored at different stages of the mechanical-electrical energy train, SFC is subdivided into **Indicated (ISFC)**, **Brake (BSFC)**, **Combined/Generator (CSFC)**, and **Plant Net (PSFC)** consumption rates. A lower SFC indicates a highly efficient plant that requires less fuel to produce the same work output.

To compute the specific fuel consumption on a standard hourly operating basis, use the fuel mass flow rate (m_f, input in kg/s) and match it to the corresponding power metric:

- **Indicated Specific Fuel Consumption:**

$$ISFC = \frac{m_f \ x \ 3{,}600}{IP} \quad ; \quad \left[\frac{kg}{kW-hr}\right]$$

- **Brake Specific Fuel Consumption:**

$$BSFC = \frac{m_f \ x \ 3{,}600}{BP} \quad ; \quad \left[\frac{kg}{kW-hr}\right]$$

- **Combined Generator Fuel Consumption:**

$$CSFC = \frac{m_f \ x \ 3{,}600}{EP} \quad ; \quad \left[\frac{kg}{kW-hr}\right]$$

- **Plant Net Specific Fuel Consumption:**

$$PSFC = \frac{m_f \ x \ 3{,}600}{kW_{delivered}} \quad ; \quad \left[\frac{kg}{kW-hr}\right]$$

Heat Rates

A closely related performance metric is the **Heat Rate (HR)**, which represents the total thermal energy input required from fuel combustion to generate one unit of electrical or mechanical work output (kJ/kW-hr). While specific fuel consumption depends on the mass of the fuel, the heat rate directly incorporates the fuel's heating value. This allows engineers to compare plants burning different grades of fuel oil.

The heat rate scales the total thermal power input rate ($Q_{in} = m_f q_f$ in kW) over an hourly baseline relative to system power outputs:

$$HR = \frac{Q_{in} \ x \ 3{,}600}{Power} \quad ; \quad \left[\frac{kJ}{kW-hr}\right]$$

- **Indicated Heat Rate:** $IHR = \frac{Q_{in} \, x \, 3,600}{IP}$

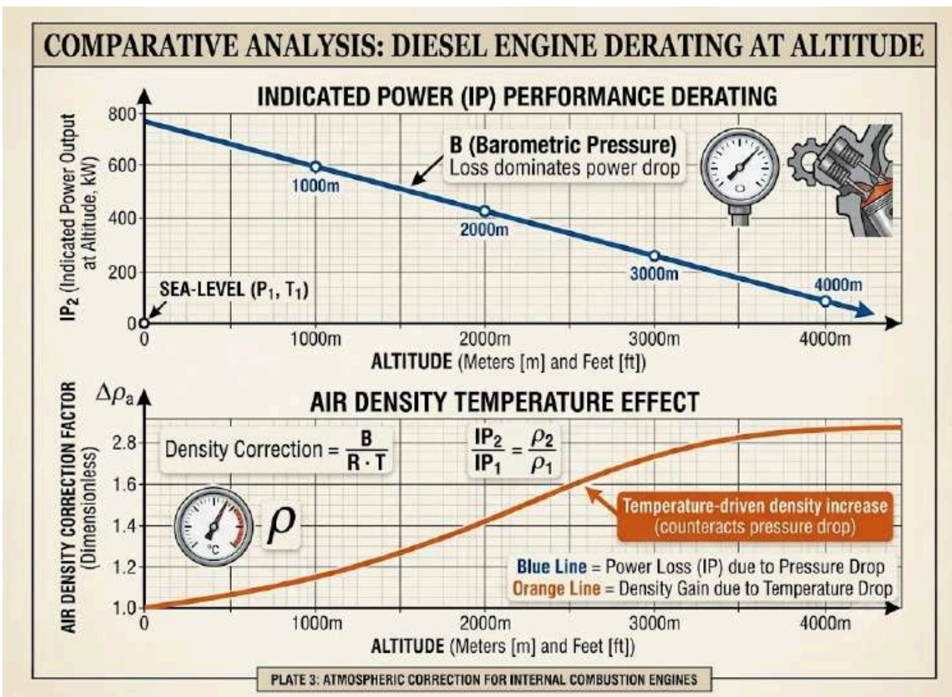
- **Brake Heat Rate:** $BHR = \frac{Q_{in} \, x \, 3,600}{BP}$

- **Combined Heat Rate:** $CHR = \frac{Q_{in} \, x \, 3,600}{EP}$

- **Plant Net Heat Rate:** $PHR = \frac{Q_{in} \, x \, 3,600}{kW_{delivered}}$

Engines at High Elevation

When a stationary diesel engine is installed at high altitudes, the surrounding ambient atmospheric pressure and temperature drop significantly. The decrease in ambient barometric pressure reduces the density of the air (ρ_a). Because the engine pistons displace a fixed physical volume per stroke (V_D), a drop in air density means a smaller total mass of oxygen is drawn into the combustion chamber during each intake cycle.



To maintain a proper, stoichiometric air-fuel ratio and prevent dangerous, smoky incomplete combustion, the mass flow rate of the fuel must be held constant or derated to match this reduced oxygen supply. Since the indicated power (IP) developed inside the combustion chamber is directly proportional to the mass of oxygen available to burn fuel, **IP scales linearly with ambient air density**.

Furthermore, as elevation increases, ambient temperature drops. This cooler air slightly increases air density, which helps counteract a portion of the pressure loss. To accurately predict the available power output of a power plant built in mountainous regions, engineers must apply mathematical derating factors that combine these environmental pressure and temperature changes.

When an engine operates at an elevated altitude (subscript 2) compared to a reference sea-level station (subscript 1), several parameters are assumed to remain constant to protect the mechanical components:

- **Friction Power Invariance:** $FP_1 = FP_2$
- **Fuel Mass Flow Constraint:** $m_{f1} = m_{f2}$
- **Volumetric Air Flow Rate Invariance:** $V_{a1} = V_{a2}$

The engine's indicated power varies directly with changes in internal air density caused by environmental conditions:

$$\frac{IP_1}{IP_2} = \frac{\rho_{a1}}{\rho_{a2}}$$

To isolate changes in power based on specific environmental factors, use the following formulas:

A. Considering Barometric Pressure Effects Only

$$P_e = P_s\left(\frac{B}{29.92}\right) \qquad \text{where: } B = 29.92 - \frac{h}{1000}$$

Where P_e is the available power at altitude (kW), P_s is the sea-level rated power (kW), B is the barometric pressure at altitude (in. Hg), and h is the elevation height (feet).

B. Considering Atmospheric Temperature Effects Only

$$P_e = P_s\sqrt{\frac{T}{520}} \qquad \text{where: } T = 520 - \frac{36h}{1000}$$

Where T is the absolute ambient temperature at altitude measured in **Degrees Rankine (°R)**, and the reference baseline sea-level temperature is assumed to be 520 **°R** (60 °F).

C. Combined Pressure and Temperature Derating Equation

When evaluating a power plant installation where both ambient barometric pressure and temperature drop simultaneously with altitude, apply the combined derating formula:

$$P_e = P_s\left(\frac{B}{29.92}\right)\sqrt{\frac{T}{520}}$$

Fuel Density and Temperature Dependency

In power plant design, accurate bookkeeping of fuel mass is crucial for efficiency calculations and combustion control. Because liquid petroleum products expand and contract significantly with temperature, measuring fuel by volume alone is unreliable. Instead, engineers use a standard reference point for fuel properties, typically set at **60°F** (or **15.6°C**). The mass density of the fuel (ρ_f) at this baseline is calculated by multiplying its specific gravity (SG_f) by the density of pure water (ρ_w).

The mass density of liquid fuel at the standard reference temperature is calculated as:

$$\rho_f = \rho_w \cdot SG_{f(@60^\circ F\ or\ 15.6^\circ C)}$$

Where the reference density of water (ρ_w) is:

- ρ_w = 1000 kg/m^3 (SI units)
- ρ_w = 62.4 lbm/ft^3 (English units)

When fuel is pumped through an operating plant at temperatures (t_o) that deviate from this standard baseline, its physical density drops due to thermal expansion. To account for this change, engineers apply empirical temperature correction factors. For every degree Celsius above the standard temperature, the specific gravity of a typical petroleum fuel decreases by approximately **0.0007** (or **0.0004** per degree Fahrenheit). Adjusting for these changes ensures that the fuel injection system delivers the exact mass flow rate required for complete combustion.

To calculate the corrected specific gravity at an actual operating temperature (t_o), apply the appropriate thermal correction factor based on your unit system:

- **SI Units (Celsius):**

$$SG_{corrected} = SG_{f(@15.6^\circ C)}\left[1 - 0.0007(t_o - 15.6^oC)\right]$$

- **English Units (Fahrenheit):**

$$SG_{corrected} = SG_{f(@60^\circ F)}\left[1 - 0.0004(t_o - 60^oF)\right]$$

Petroleum Specific Gravity Scales: API and Baumé

To simplify hydrometer readings in the field, the petroleum and chemical industries developed specialized, non-linear gravity scales:

- **The American Petroleum Institute (API) Scale:** This is the standard scale used globally for pricing and classifying crude oil and liquid fuels. It has an inverse relationship with specific gravity; lighter, higher-quality oils have higher AP values, while heavier oils have lower values. Pure water has an API gravity of exactly **10°**.

$$\circ API = \frac{141.5}{SG_{f(@60\circ F)}} - 131.5 \qquad \Rightarrow \qquad SG_{f(@60\circ F)} = \frac{141.5}{\circ API + 131.5}$$

- **The Baumé Scale:** Historically preferred by industrial chemists, this scale operates on a similar mathematical framework to track the density of various liquids.

$$\circ BAUME = \frac{140}{SG_{f(@60\circ F)}} - 130 \qquad \Rightarrow \qquad SG_{f(@60\circ F)} = \frac{140}{\circ BAUME + 130}$$

Both scales anchor their formulas to the specific gravity of the fluid measured at the standard reference temperature of **60°F**.

The total chemical energy released as heat when a unit mass of fuel burns completely is called its **heating value** or **calorific value**. This value changes depending on whether the water produced during combustion is condensed or remains as vapor:

- **Higher Heating Value (HHV):** Also known as the *Gross Calorific Value*, this metric assumes that all water vapor produced during combustion is fully condensed back into liquid form. This condensation releases the latent heat of vaporization, maximizing the total measured heat output. This value is used to evaluate external combustion systems, like steam boiler plants, which can sometimes recover this energy.

  When laboratory bomb calorimetry data is unavailable, the gross heating value of petroleum fuels can be estimated using these standardized empirical formulas:

  ASME Power Test Code Formula:

  HHV = 41130 + (139.6)(°API) ; kJ/kg

  Bureau of Standards Formula:

  HHV = 51716 - 8793.8 $\left[SG_{f(@60\circ F)}\right]^2$; kJ/g

- **Lower Heating Value (LHV):** Also known as the *Net Calorific Value*, this metric assumes the water remains entirely in a gaseous state as it exits the system. Because the latent heat required to vaporize this water is lost to the environment, the LHV is always lower than the HHV. Internal combustion engines, such as diesel power units, cannot condense exhaust moisture during their power stroke, making the LHV the appropriate baseline for calculating their thermal efficiency.

  $LHV = HHV - q_v$

  The vaporization loss is determined by the total mass of water (m_W) produced from the hydrogen content in the fuel, multiplied by the latent heat of vaporization (h_{fg} = 2442 kJ/kg):

$$q_v = m_W \cdot h_{fg}$$

$$q_v = \left[9\left(\frac{\%G_{H_2}}{100}\right)\right] \cdot 2442$$

$$q_v = 219.78\left(\%G_{H_2}\right)$$

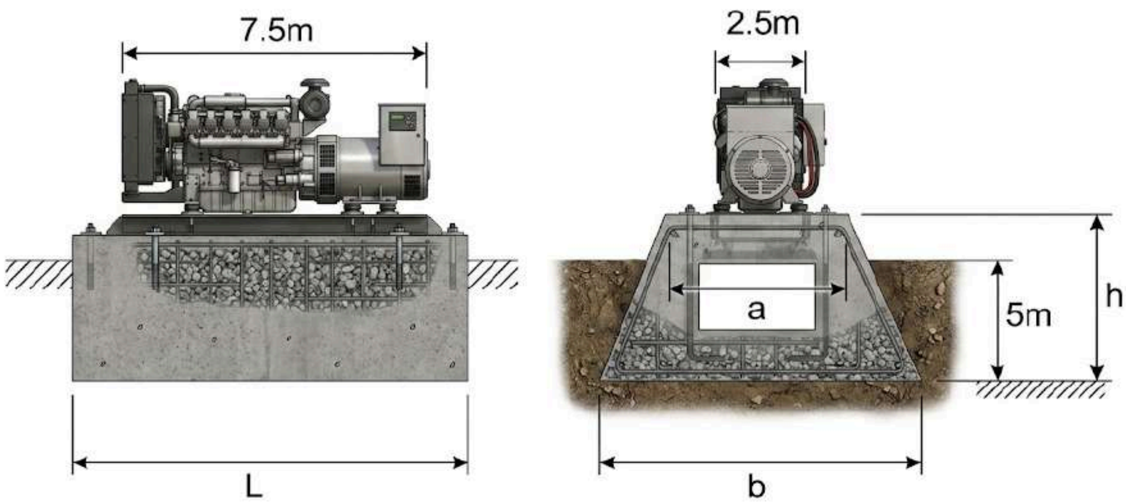
Where the internal percentage of hydrogen $\left(\%G_{H_2}\right)$ is derived from the ultimate analysis of the fuel's specific gravity:

$$\%G_{H_2} = 26 - 15\left[SG_{f(@15.6°C)}\right]$$

Machine Foundations

A machine foundation is a specialized structural element designed to support heavy engineering equipment, such as diesel electric generating sets, while safely transferring all static and dynamic loads into the underlying soil column. According to the **Philippine Mechanical Code** and standard engineering practices outlined by Frederick T. Morse, a machine foundation must fulfill three core criteria:

1. It must support the static weight of both the machine and its accessories without exceeding the safe bearing capacity of the soil.
2. It must maintain perfect alignment of the engine crankshaft and generator shaft to prevent structural warping and mechanical fatigue.
3. It must provide sufficient mass to absorb and damp the cyclic vibrations generated by the reciprocating components of the engine.



Engineers use two distinct methods to determine the required mass of a foundation, depending on the engine's design speed:

- **The Empirical Mass Coefficient Method:** For slow-speed, heavy-duty industrial diesel engines, the foundation mass is calculated using a fixed empirical mass coefficient based on the brake horsepower (bhp) of the unit. This specific weight multiplier typically ranges from 400 kg/hp to 500 kg/hp. This approach relies on brute weight to stabilize the system and counteract low-frequency vibration pulses.
- **The PSME Code Dynamic Square-Root Method:** For modern medium-to-high-speed standby engines, the **Philippine Society of Mechanical Engineers (PSME) Code** applies a dynamic empirical equation:

$$W_f = e \cdot W_m \cdot \sqrt{N}$$

This formula scales the foundation weight (W_f) relative to the weight of the machine $\left(W_m\right)$ and the square root of its operational speed (N, in rpm). The empirical coefficient (e) accounts for the differences in vibration profiles between multi-cylinder configurations. This ensures that lighter, higher-speed engines are supported by a foundation that prevents dangerous harmonic resonance with the surrounding soil.

Soil Bearing Capacity and the Structural Safety Factor

The structural stability of a power plant depends entirely on the interaction between the lower face of the concrete foundation and the soil subgrade. The maximum allowable pressure the ground can support without settling or shifting is known as the **Safe Load Bearing Capacity (S_b)**.

Because reciprocating engines exert continuous dynamic loads and cyclical vibrations, civil and mechanical codes require a strict **Safety Factor of 2.0** when calculating the foundation's footprint. This means the actual combined static pressure exerted by the machine and the concrete block $\left(\frac{W_m + W_f}{Area}\right)$ must never exceed **half** of the site's allowable soil bearing capacity $\left(\frac{S_b}{2}\right)$

Concrete Mix Metrology and Structural Reinforcement

To build a machine foundation capable of resisting dynamic shear stress, engineers typically specify a standard **1:2:4 structural concrete mixture** (Class A concrete, containing 1 part cement, 2 parts sand, and 4 parts gravel or stone by volume). In power plant engineering, the standard mass density of cured reinforced concrete is assumed to be **2406 kg/m^3**.

While concrete has excellent compressive strength, it has low tensile strength. The continuous vibration of an internal combustion engine generates alternating tensile stresses across the bottom and sides of the block. To prevent cracking, steel reinforcing bars (rebars) must be embedded within the pour. This internal reinforcement typically makes up **0.5% to 1.0%** of the total foundation weight.

Structural Design Formulations & Governing Equations

1. Geometric Footprint and Clearance Bounds

The length (L) and width (W) of a foundation block are determined by adding a standard maintenance clearance distance (C, typically 1 foot or 0.3048 m) around the perimeter of the machine's metal bedplate:

$$L = L_{bedplate} + 2C$$

$$W = W_{bedplate} + 2C$$

2. Foundation Mass Determination Methods

- **Slow-Speed Specific Weight Method:**

$$W_f = [Specific\ Weight\ Factor,\ kg/hp] \cdot [Brake\ Horsepower,\ bhp]$$

$$bhp = \frac{Electrical\ Output\ Capacity,\ kW}{0.746 \cdot \eta_{genset}}$$

- **PSME Dynamic Operational Method (High-Speed Units):**

$$W_f = e \cdot W_m \cdot \sqrt{N}$$

Where e = 0.11 (standard coefficient for inline multi-cylinder units per PSME Table 2.2), W_m is the total engine weight (kg), and N is the engine speed (rpm).

3. Soil Stability Verification Boundary

$$\frac{S_b}{2} \geq \frac{W_m + W_f}{W \cdot L}$$

Note: Ensure all terms in the numerator match the weight units of the soil bearing capacity S_b (e.g., convert kg to metric tons if S_b is in tons/m^2).

4. Depth and Volume Relationships

Once the required foundation mass (W_f) is determined, the total volumetric space (V_f) is calculated using the density of concrete ($\rho_{concrete}$ = 2406 kg/m^3):

$$V_f = \frac{W_f}{\rho_{concrete}} = \frac{W_f}{2406}$$

The physical depth (D) of the required excavation is then derived from the geometric volume of a rectangular prism:

$$D = \frac{V_f}{W \cdot L}$$

5. Concrete Material Yield Estimations (1:2:4 Volumetric Proportions)

For every 1m^3 of finished concrete volume, the following raw material quantities are required:

- **Cement:** 7.8 sacks (or 7.8 ft^3)
- **Sand (Fine Aggregate):** 0.44 m^3
- **Stone (Coarse Aggregate):** 0.88 m^3

6. Steel Reinforcement Estimation

Total Steel Rebar Mass (W_{steel}) = Reinforcement Ratio (μ) $\cdot W_f$

Number of Rebars = $\frac{W_{steel}}{Unit\ Mass\ per\ Rebar\ Length}$

Where μ typically ranges between 0.005 and 0.010 (chosen as a 0.75% median baseline).