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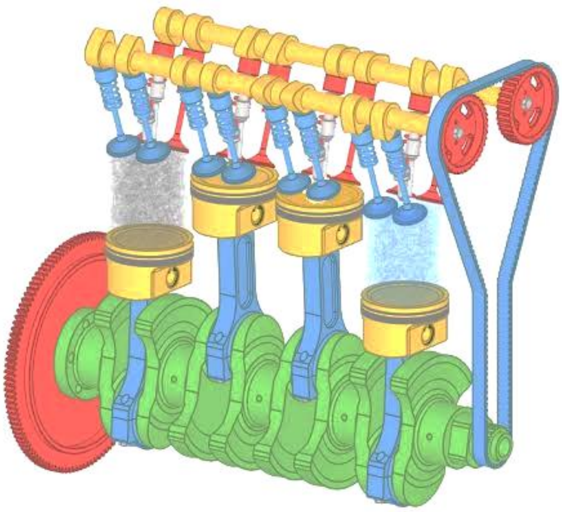
Field of study: Mechanical Engineering

Specialty: Mechanical Construction

Handout of courses

Internal Combustion Engine



Sixth semester LMD, Mechanical Construction

Presented by

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Nomenclature

L	The length of stroke	[mm]
A	The cross sectional area of piston	[mm²]
$CR = r_v = \varepsilon$	Compression ratio	[ ]
D	Bore	[mm]
S	Stroke	[mm]
V_d	Displaced volume	[mm^3]
$V_0 = V_c$	Clearance volume	[mm^3]
V_T	Total volume	[mm^3]
C_m	Average piston speed	[m/s]
ΔU	The system's internal energy	[J]
δW	The work done by the system	[J]
δQ	The heat added/ lost to the system	[J]
P	Gas pressure	[*Pa*]
T	Gas temperature	[*K*]
V	Gas volume	[m^3]
n	Number of mole	[*mole*]
R	Gas constant	[*J/mole.K*]
r	Massic gas constant	[*J/kg.K*]
m	Mass of gas in cylinder	[*kg*]
C_p	Specific heat at constant pressure	[*J/kg.K*]
C_v	Specific heat at constant volume	[*J/kg.K*]
η_{th}	The the thermal efficiency	[ ]
MEP	The mean effective pressure	[Pa]
CA	Crank angle	[deg.]
C	The engine torque	[N.m]
N	The engine speed	[rpm]
ω	Angular speed of the crankshaft	[rad/s]
P_e	Effective power	[W]
$x(\beta)$	Instantaneous position of the piston	[m]
β	Crank angle from top dead center	[rad]
λ	The connecting rod-crank ratio	[ ]
t	Time	[s]

$\dot{x}(\beta)$	The instantaneous piston speed	[m/s]
$\overline{v}$	Mean piston speed	[m/s]
$\ddot{x}(\beta)$	The instantaneous piston acceleration	[m/s²]
$V(\beta)$	Instantaneous combustion chamber volume	[m^3]
F	The resulting force or pressure force	[N]
F_i	The inertial forces	[N]
MIP	Mean Indicated Pressure	[Pa]
W_i	Indicated work	[J]
W_e	Effective work	[J]
Pi	Mean indicated power	[W]
Pe	Effective Power	[W]
η_{mec}	The mechanical efficiency	[ ]
η_i	Indicated efficiency	[ ]
η_e	Effective efficiency	[ ]
m_f	The quantity of fuel	[kg]
b_i	Specific Fuel Consumption	[g/kWh]
be	Effective Specific Fuel Consumption [g/kWh]	[g/kWh]

Abbreviations

ICE	Internal combustion engine	
ECE	External combustion engine	
SI	Spark-ignition	
CI	Compression-ignition	
LGP	Liquefied petroleum gas	
LNG	Liquefied Natural Gas	
TDC	Top dead center	
BDC	Bottom dead center	
BTDC	Before top dead center	
IA	Ignition advance (auto-ignition)	
IOA	Intake valve opening advance	
ICD	Intake valve closing delay	
EOA	Exhaust valve opening advance	
ECD	Exhaust valve closing delay	
LHV	Low heating value of the fuel	[kJ/kg]

Introduction

This course is intended for third-year undergraduate students (LMD system) specializing in Mechanical Construction. It is designed to provide students with the knowledge and methodological tools required to carry out thermodynamic analysis and synthesis studies of internal combustion engines. The course content has been developed in accordance with the official curriculum of the sixth semester of the third-year undergraduate program in Mechanical Construction.

Internal combustion engines are among the most important technological achievements in modern mechanical engineering. They are widely used in transportation, industrial applications, power generation, agriculture, and marine systems due to their ability to efficiently and reliably convert the chemical energy of fuels into usable mechanical energy. The study of internal combustion engines integrates several engineering disciplines, including thermodynamics, fluid mechanics, heat transfer, combustion science, machine dynamics, and materials engineering.

This handout is designed for 3rd Year undergraduate mechanical engineering students, particularly those studying engine technology and power conversion systems. It provides a comprehensive introduction to the fundamental principles, operating characteristics, thermodynamic cycles, and mechanical behavior of internal combustion engines. The course aims to develop the theoretical understanding and practical knowledge necessary for the analysis, design, operation, and maintenance of modern engines.

The first chapter begins with an overview of internal combustion engines and their classification according to ignition method, fuel type, number of strokes, cooling system, and application. Particular attention is paid to the operating principles of spark-ignition (SI) and compression-ignition (CI) engines, which represent the two main categories used in the automotive and industrial sectors.

Chapter 2 includes a detailed analysis of thermodynamic cycles, including the Beau-de-Rochas (Otto) cycle, the Diesel cycle, and the Sabatier cycle. These ideal cycles are analyzed to understand the energy transformation within the cylinder and to evaluate engine performance using parameters such as thermal efficiency, compression ratio, pressure, temperature, and heat transfer. The study also presents real engine cycles and compares them with theoretical models to highlight the practical limitations resulting from energy losses during combustion, friction, heat dissipation, and incomplete gas exchange processes.

Chapter 3 details the actual operating cycle of diesel engines through the four basic stages: intake, compression, combustion and expansion, and exhaust. The chapter presents diagrams and key engine characteristics to help students understand engine behavior under various operating conditions. It emphasizes the relationship between theoretical analysis and practical engine performance.

Chapter 4: Dynamics of Reciprocating Engines. This chapter examines the kinematic and dynamic analysis of the sliding crankshaft mechanism, connecting rod motion, balancing of rotating and reciprocating masses, and valve timing systems to understand vibration phenomena, mechanical stresses, and power transmission within the engine. These concepts are fundamental to improving engine durability, smooth operation, and efficiency.

Chapter 5 addresses engine performance parameters, such as indicated power, braking capacity, torque, specific fuel consumption, volumetric efficiency, and thermal efficiency. Engine characteristics are analyzed under full and partial load operating conditions according to international standards. The impact of fuel properties, combustion quality, and ignition systems on engine operation is also presented.

The objective of this handout is therefore to provide students with a solid scientific and technical foundation in internal combustion engines, and to prepare them for advanced studies and professional applications in automotive engineering, energy systems, manufacturing industries, and research fields related to thermal machinery and sustainable energy technologies.

Chapter I: General Information

1.1. Introduction

A heat engine is an energy conversion device that transforms thermal energy into mechanical energy. Since its invention, it has undergone continuous development and refinement in order to maximize efficiency and extract the greatest possible amount of usable energy.

More specifically, the internal combustion engine (ICE) converts the thermal energy released during fuel combustion into mechanical work. In an ICE, the chemical energy stored in the fuel is released through combustion within the engine's combustion chamber. Fuels such as gasoline (petrol), diesel, liquefied petroleum gas (LPG), and natural gas undergo exothermic reactions that generate high-temperature and high-pressure gases. These gases transfer energy to the working medium, producing mechanical work through the motion of pistons or turbine components.

The energy conversion process is governed by a thermodynamic cycle, which consists of a sequence of complex physicochemical transformations occurring periodically within the engine. The thermodynamic efficiency of this cycle is a key performance indicator, as it directly influences power output, specific fuel consumption, operational reliability, emission characteristics, and engine durability. Consequently, improving cycle efficiency remains a central objective in engine research and development.

1.2. Classification of Heat Engines

Heat engines are generally classified into two principal categories based on the location of the combustion process: external combustion and internal combustion.

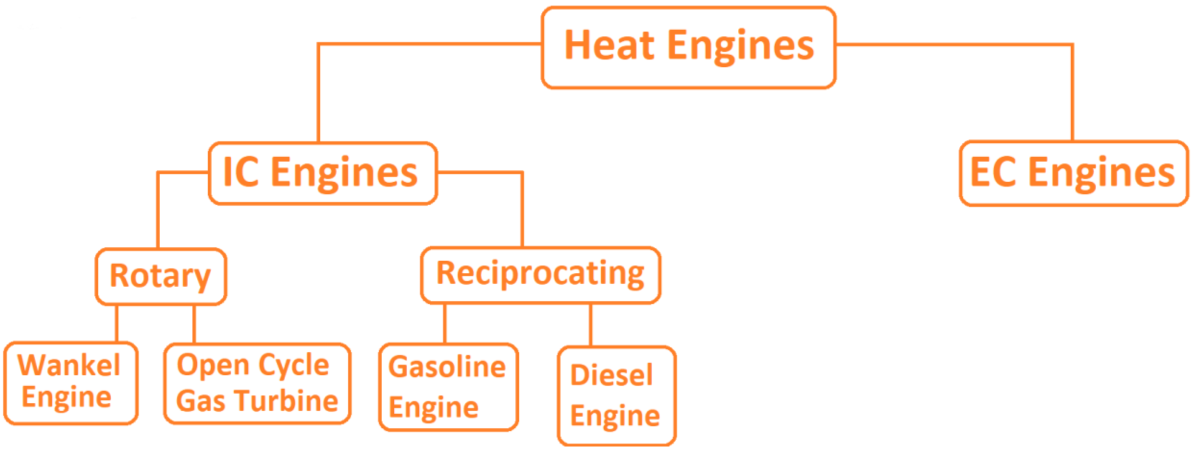


Figure 1.1. *Classification of heat engines*

1.2.1. External Combustion Engine (ECE)

An ECE is a heat engine in which fuel is burned outside the engine is working cylinders. The heat generated from combustion is transferred to a separate working fluid (such as steam, air, or another gas) which then expands to perform mechanical work. It is characterized by the following features:

- Combustion occurs externally, typically in a furnace, boiler, or heater.
- Heat is transferred to the working fluid through a heat exchanger.
- The heated working fluid expands and drives a piston, turbine, or similar mechanism to produce power.
- The combustion gases do not come into direct contact with the engine's moving parts

1.2.2. Internal Combustion Engine (ICE)

An ICE is a heat engine in which the combustion of fuel takes place inside the engine cylinder or combustion chamber. The heat generated during combustion increases the temperature and pressure of the air (or air-fuel mixture) within the cylinder. The resulting high-pressure gases expand and push the piston, causing it to move back and forth. This reciprocating motion is converted into rotary motion by the crankshaft, thereby producing mechanical work. Most internal combustion engines are ***reciprocating engines,*** where a piston moves back and forth inside a cylinder. The following features characterize the ICE are:

- Combustion occurs internally within the engine cylinder or combustion chamber.
- The combustion gases directly act on the piston or turbine blades.
- Power is produced from the expansion of high-temperature, high-pressure gases.
- Common fuels include petrol (gasoline), diesel, natural gas, and biofuels

1.3. Classification of Internal Combustion Engines

Internal combustion engines convert the chemical energy found in fuels such as gasoline, diesel, LPG, or natural gas into mechanical energy. Inside the engine's combustion chamber, fuel reacts chemically with air to produce heat energy. This heat increases the pressure of the gases in the chamber, forcing the piston to move.

The lubricants used in vehicle engines are defined according to the fuel type, and oil standards and specifications are set by authorized global institutions.

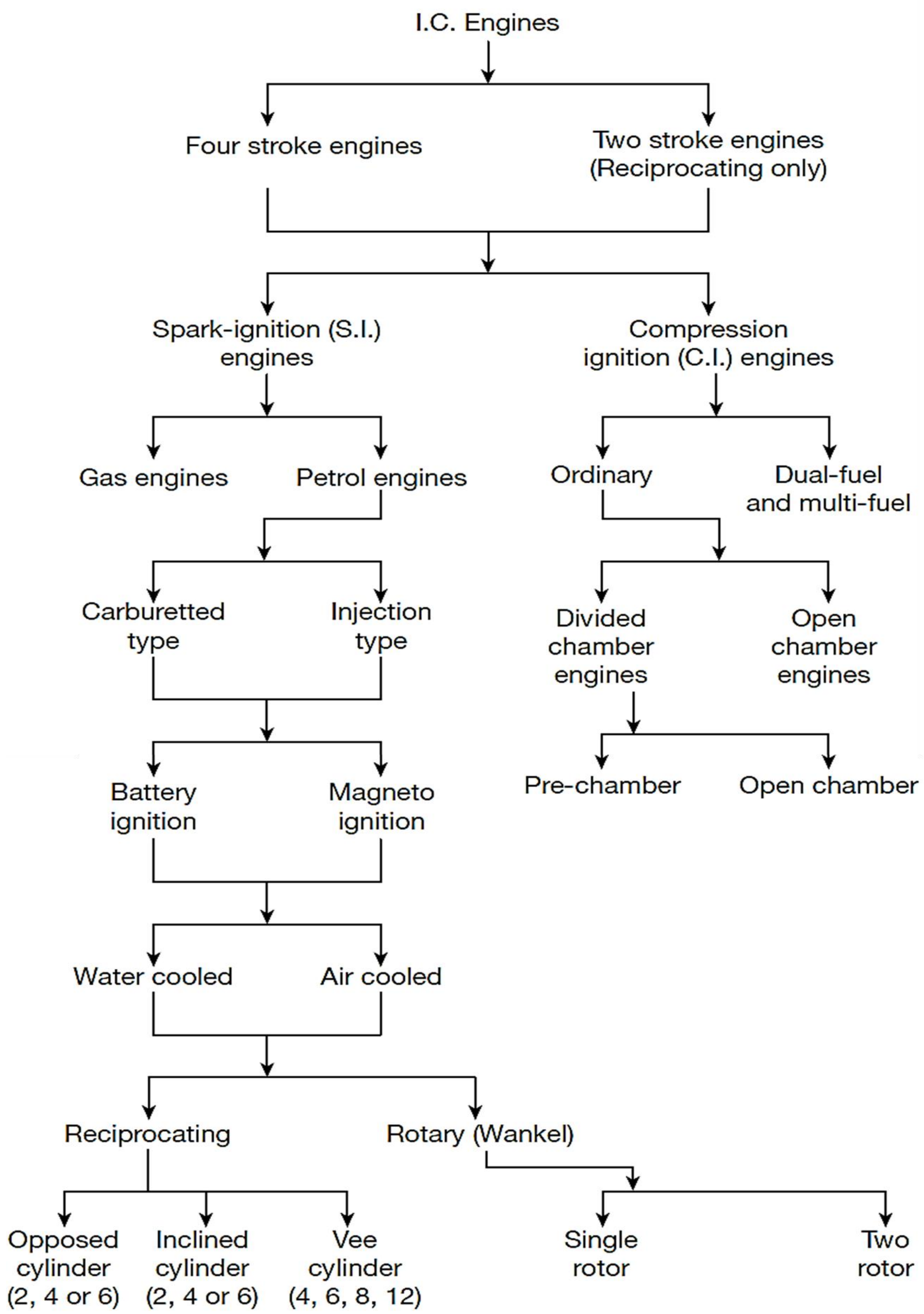


Figure 1.2. *Internal combustion engine classification*

Internal combustion engines (ICEs) can be classified according to several criteria:

1.3.1. According to Their Theoretical Cycle

- **Two stroke engine (2 stroke)**

  When the cycle is completed in one revolution of the crankshaft, it is called two-stroke cycle engine.
- **Four stroke engine (4 stroke)**

  When the cycle is completed in two revolutions of the crankshaft, it is called four-stroke cycle engine.

1.3.2. According to Rotational Speed

- **High speed engines**

  Used in passenger cars and trucks; operate at high rpm.
- **Medium speed engines**
- **Low-speed engines**

  Typically large engines such as marine propulsion engines.

1.3.3. According to the Type of Fuel

- **Liquid fuels**
  - Gasoline
  - Diesel
- **Gaseous fuels**
  - LPG (Liquefied Petroleum Gas)
  - LNG (Liquefied Natural Gas)

1.3.4. According to the Mixture Preparation Method

- **Direct injection (internal mixture formation)**

  Fuel is injected directly into the combustion chamber (common in diesel engines).
- **External mixture formation**

  Air-fuel mixture is prepared outside the combustion chamber (e.g., carbureted engines).

1.3.5. According to the Ignition Method

1. Spark ignition engine (petrol or gasoline engines)
2. Compression ignition engine (diesel engines)

- **Spark ignition engine** : A mixture of air and fuel is drawn in to the engine cylinder. Ignition of fuel is done by using a spark plug. The spark plug produces a spark and ignites the air- fuel mixture. Such combustion is called constant volume combustion (C.V.C.).

- **Compression ignition engine** : In compression ignition engines air is compressed in to the engine cylinder,. Due to this the temperature of the compressed air rises to 700-900°C. At this stage diesel is sprayed in to the cylinder in fine particles. Due to a very high temperature, the fuel gets ignited. This type of combustion is called constant pressure combustion (C.P.C.) because the pressure inside the cylinder is almost constant when combustion is taking place.

1.3.6. According to the Air Intake Method

- **Naturally aspirated engines**
  Air enters the cylinder under atmospheric pressure.
- **Supercharged engines**
  Air is compressed before entering the cylinder (e.g., turbocharged engines).

1.3.7. According to the Type of Cooling

- **Air-cooled engines**
  Cooling is achieved directly by air flowing over the engine's external surface (cooling fins are usually provided).
- **Liquid-cooled engines**
  A liquid coolant (water or antifreeze solution) circulates through the engine to remove heat.

1.3.8. According to the Compression Ratio

- **High compression ratio engines (14:1 to 25:1)**
  Typical of diesel engines.
  Higher compression increases thermal efficiency and allows self-ignition of fuel.
- **Low compression ratio engines (4:1 to 12:1)**
  Typical of gasoline engines.
  Limited by knocking in spark-ignition engines.

1.3.9. According to the Number of Cylinders

- **Single-cylinder engine**
  Only one cylinder; simple design, commonly used in small motorcycles and small machinery.
- **Multi-cylinder engine**
  Two or more cylinders (2, 3, 4, 5, 6,etc.); smoother operation and higher power output.

1.3.10. According to the Cylinder Arrangement

- **Inline (Straight) engine**

  Cylinders arranged in a single line.

  Can be vertical, horizontal, or inclined.

- **V-engine**

  Cylinders arranged in two banks forming a “V” shape.

- **Flat engine (Horizontally opposed / Boxer engine)**

  Cylinders arranged in two opposite horizontal banks.

- **Radial engine**

  Cylinders arranged radially around the crankshaft (commonly used in aircraft).

1.4. Internal combustion engine components

An internal combustion engine (ICE) has many parts that work together to convert fuel energy into mechanical power. Each component has a specific function necessary for proper engine operation. The main components are usually grouped into structural parts, moving parts, valve mechanism parts, and auxiliary systems. The main components of an IC Engine are given below:

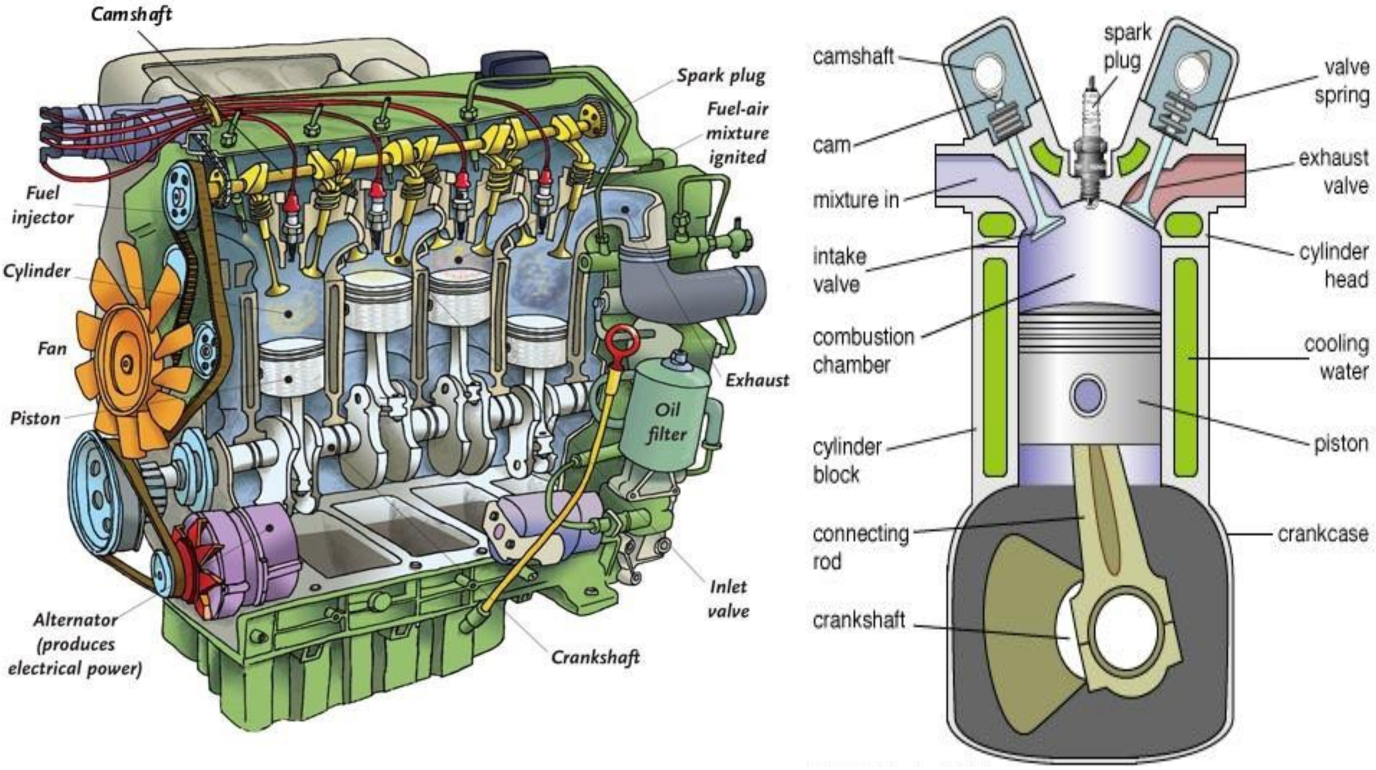


Figure 1.3. *Internal combustion engine components*

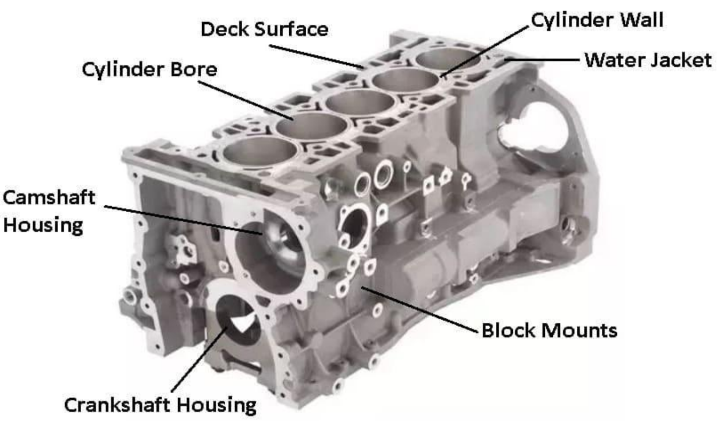
1.4.1. Structural parts

1.4.1.1. Cylinder or Block of engine

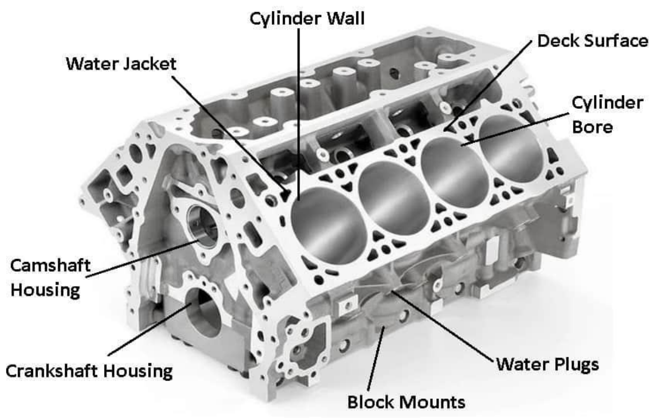
The cylinder is a part of the engine that confines the expanding gases and forms the combustion chamber. It is the basic part of the engine. It provides the space in which the piston moves to draw in air or the air–fuel mixture. The piston then compresses this charge, and after combustion the gases expand inside the cylinder, producing power for useful work. Cylinder block is the solid casting body which includes the cylinder and water jackets (cooling fins in the air cooled engines).

Cylinders are usually made of high-grade cast iron. The cylinder is often considered the heart of the engine, as the combustion of fuel takes place inside it. The inside diameter of the cylinder is called the bore.

To protect the cylinder from wear, liners or sleeves are sometimes inserted inside the cylinder. The material used for the cylinder must be able to withstand the high pressure and high temperature produced during fuel combustion.



a. Four Cylinders in Line



b. Eight cylinders in V (V8)

Figure 1.4. *Cylinder or Block of engine*

1.4.1.2. Cylinder Head

The cylinder head is a detachable part of an engine that covers the cylinders and forms part of the combustion chamber. It is fixed on top of the engine block and contains components such as valves, spark plugs, or fuel injectors. The cylinder head closes the upper part of the cylinders, forming the combustion chamber. It allows the intake of fresh gases and the exhaust of burnt gases, supports the components of the valve timing and ignition systems, and helps dissipate heat produced during combustion.

Cylinder heads are generally manufactured by casting and machining cast iron or aluminum alloys. Aluminum alloys are often preferred because they are lighter and have better thermal conductivity, which improves heat dissipation. Inside the cylinder head there are passages for water and oil that allow cooling and lubrication of engine components. The head gasket ensures the seal between the engine block and the cylinder head.

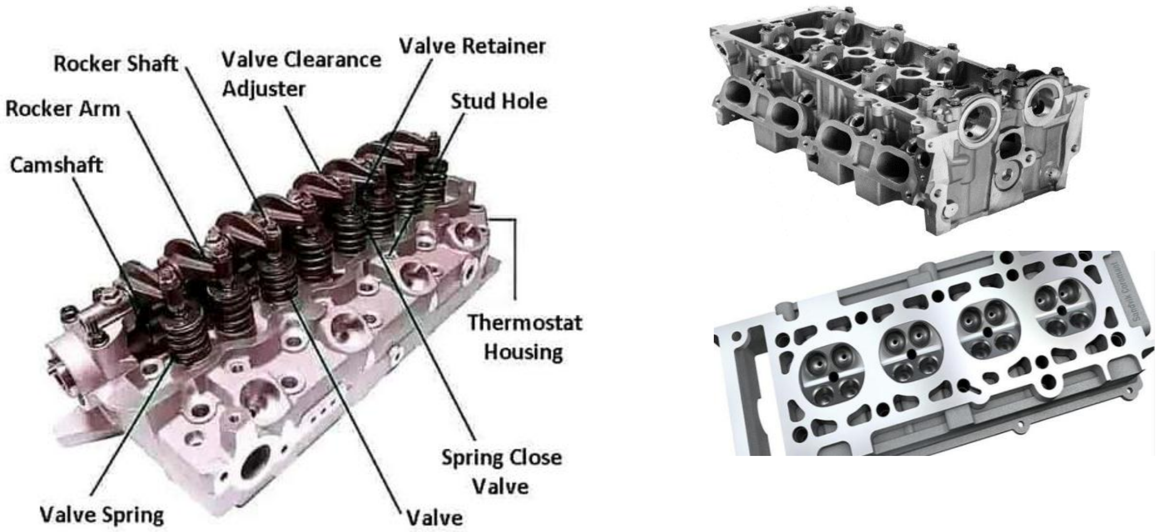


Figure 1.5. *Cylinder head*

1.4.1.3. Crankcase (crankcase oil pan)

The crankcase is that part of the engine which supports and encloses the crankshaft and camshaft. It provides a reservoir for the lubricating oil. It also serves as a mounting unit for such accessories as the oil pump, oil filter, starting motor and ignition components. The upper portion of the crankcase is usually integral with cylinder block. The lower part of the crankcase is commonly called oil pan and is usually made of cast iron or cast aluminum.

A crankcase is a protective housing or shell that encloses and supports internal mechanical parts, keeps them sealed from external contaminants, and contains the lubricant necessary for their smooth functioning

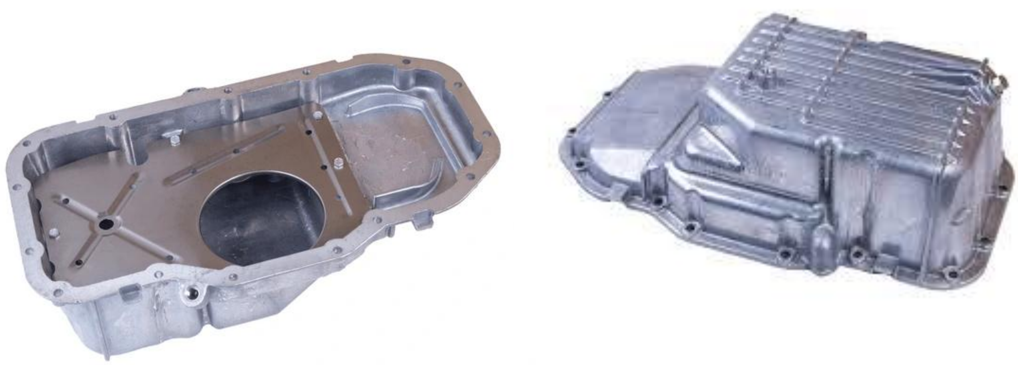
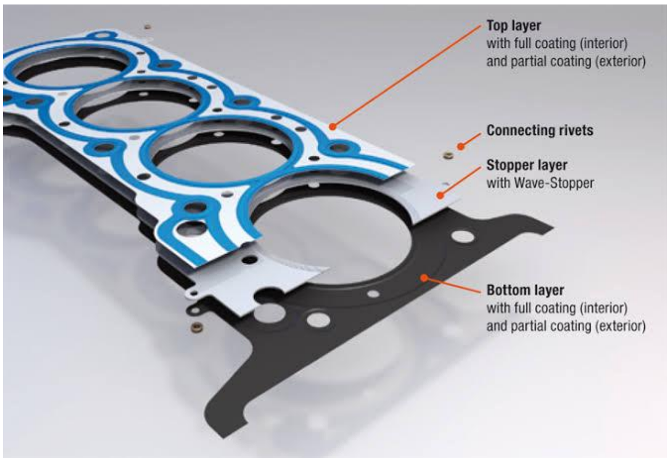


Figure 1.6. *Crankcase*

1.4.1.4. Head Gasket (Cylinder Head Gasket)

The head gasket is a critical sealing component located between the engine block and the cylinder head in an internal combustion engine. Its main purpose is to ensure a tight seal at this junction so that the engine operates properly.



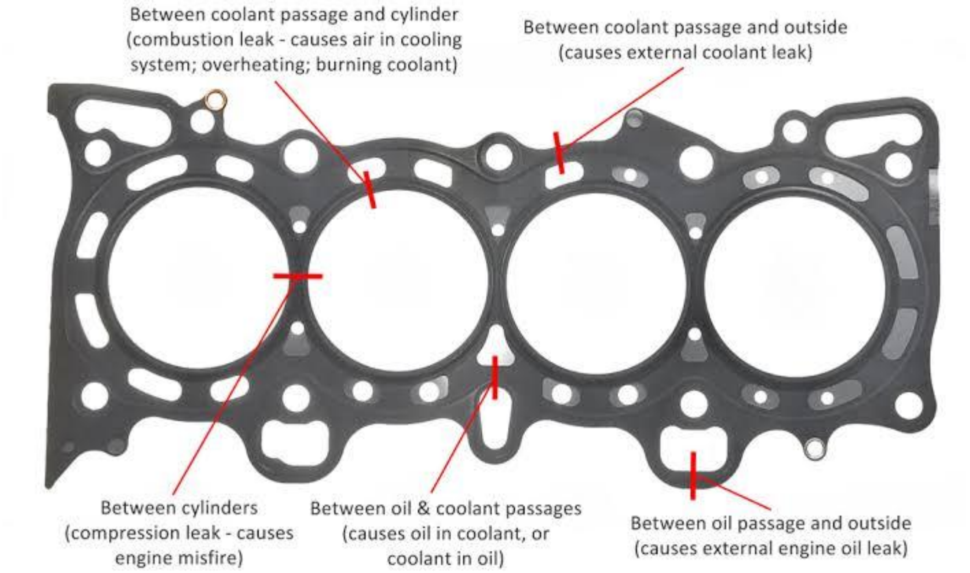


Figure 1.7. *Head Gasket*

1.4.2. Moving parts

1.4.2.1. Piston

The piston is the component that, by moving within the cylinder or cylinder liner, transmits the thrust of the gases to the crankshaft via the connecting rod. It is generally cast from a lightweight material with good thermal conductivity, such as aluminum alloys and cast iron. The head and the piston pin support, which must transmit the mechanical energy, are particularly reinforced. The piston consists of a head or base whose diameter must be smaller than the cylinder bore (including thermal expansion). Sealing is ensured by piston rings located in grooves machined around the circumference of the piston. The lower part, or skirt, of the piston must provide guidance both when cold and hot with minimal friction.

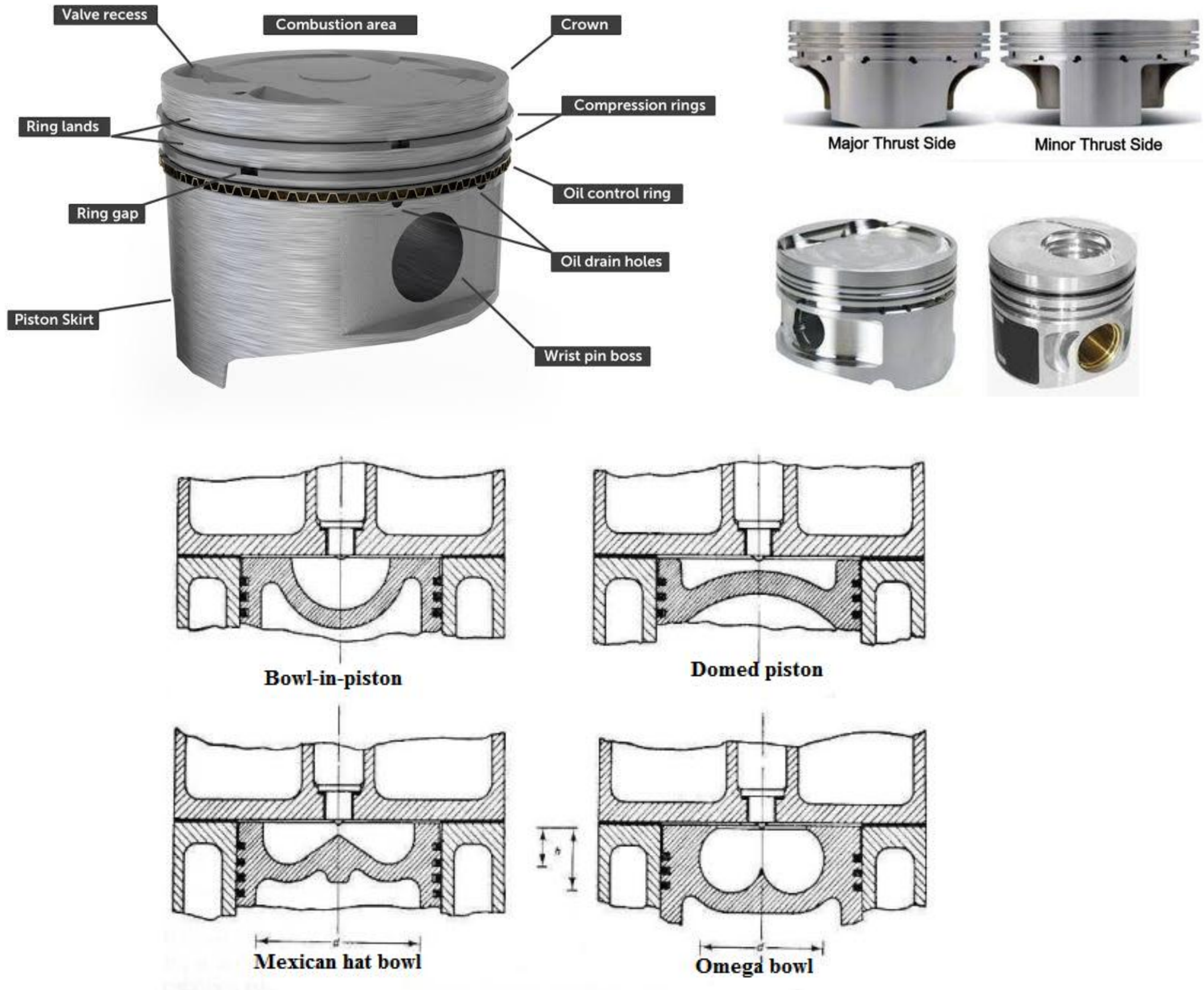


Figure 1.8. *Piston head shapes used in compression ignition engines*

To limit turbulence in a spark-ignition engine and thus prevent the spark from being extinguished, the piston head is flat. However, turbulence must be relatively significant in a compression-ignition engine because it promotes evaporation and the preparation of a homogeneous mixture after the injection of the liquid fuel, reducing the time required for combustion. To achieve this, the piston head can contain different shapes of cavities as shown in Figure 1.8.

Piston Rings

Piston rings are broken rings with a square or rectangular cross-section. They must ensure uniform radial pressure on the cylinder walls. It is a split expansion ring, placed in the groove of the piston. They are usually made of cast iron or pressed steel alloy. The function of the ring are as follows:

a. It forms a gas tight combustion chamber for all positions of piston.
b. It reduces contact area between cylinder wall and piston wall preventing friction losses and excessive wear.
c. It controls the cylinder lubrication.
d. It transmits the heat away from the piston to the cylinder walls.

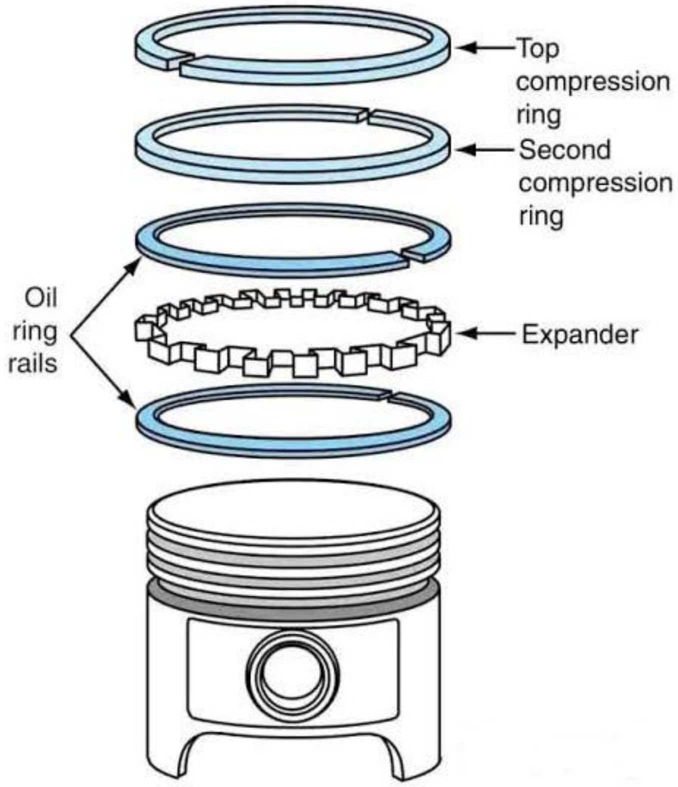


Figure 1.9. *Piston rings*

Four-stroke engines generally have three piston rings:

Top Compression Ring: This is the primary seal for combustion gases. As you mentioned, it is often treated with advanced materials (like plasma-sprayed ceramic or molybdenum) to handle the extreme heat and pressure at the top of the stroke.

Second Compression Ring (Wiper Ring): This is the "middle" ring. While it provides a secondary seal for combustion pressure, its main job is to scrape (wipe) excess oil off the cylinder wall that the oil ring might have missed, while also helping to transfer heat from the piston to the cylinder wall.

Oil Control Ring: This is the bottom-most ring assembly. It consists of two thin rails and an expander. Its sole purpose is to scrape oil back into the crankcase while leaving a microscopic film of oil to lubricate the rings above it

1.4.2.2. Connecting Rod

The connecting rod is a mechanical component whose end is connected to the piston by the piston pin and the other end to the crankshaft journal. It transforms the reciprocating linear motion of the piston into the continuous rotary motion of the crankshaft. Connecting rods are typically made of high-strength nickel-chromium steel, or sometimes medium-hard carbon steel. Aluminum alloy connecting rods are also used in high-performance engines due to their light weight.

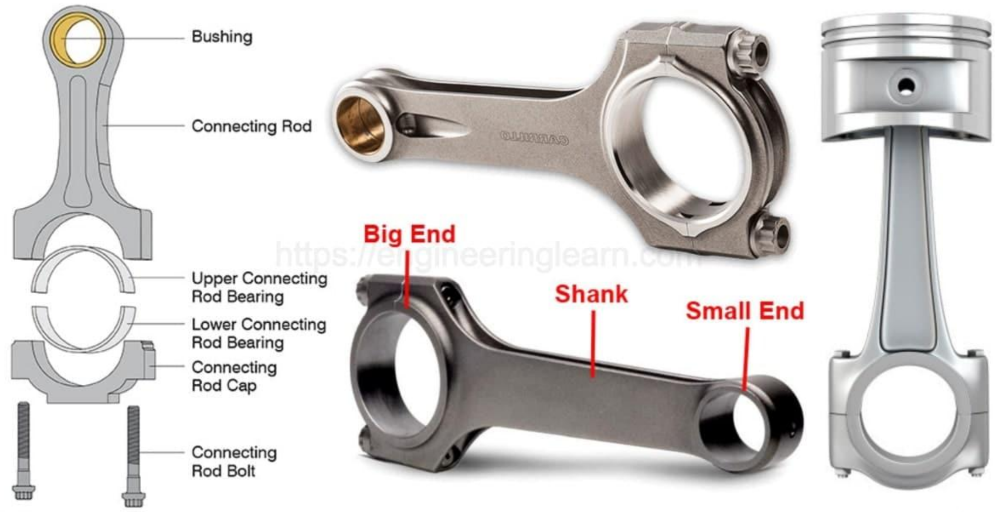


Figure 1.10. *Types and parts of connecting rod*

The connecting rod consists of three parts:

1. The small end, connected to the piston pin, usually with a press-fit bronze bushing, or in some cases with a needle bearing.
2. 2. The body is the part between the small end and the big end. It has an "H" or "I" cross-section to withstand various compressive and tensile stresses and to prevent buckling.

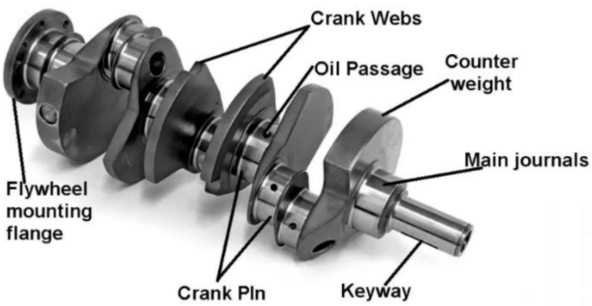
3. The connecting rod big end, which rotates on the crankshaft journal, is cut in a plane perpendicular to the connecting rod's axis to allow for the installation of the bearings and its mounting on the crankshaft journal.

The lower part that fits over the journal is called the cap. It is generally secured with bolts. The cut can be straight or angled relative to the connecting rod's axis. To allow rotation on the crankshaft, either needle bearings or thin-walled bushings can be used. In the first case, the crankshaft must be disassembled into several parts to remove the connecting rod.

The connecting rod length refers to the distance between the axis of the big end and the axis of the small end. The connecting rod's big end articulates with the crankshaft journal via a thin bearing. Thin bearings provide good durability and conductivity. They consist of a cold-rolled steel support, rolled into a semicircle, and coated with a thin layer of antifriction metal.

1.4.2.3. Crankshaft

A crankshaft is an essential part of the power transmission system. In which, the reciprocating movement of the piston is converted into a rotating movement by using the connecting rod. A crankshaft consists of crankpins, crank webs (crank arms or cheeks), balancing weights, and main journals. The large end of the connecting rod is attached to the crankpin of the crankshaft. During one stroke, the center-to-center distance between the crankpin and the crankshaft is half the piston displacement. Thus, one complete revolution of the crankshaft makes two stroke of the piston.



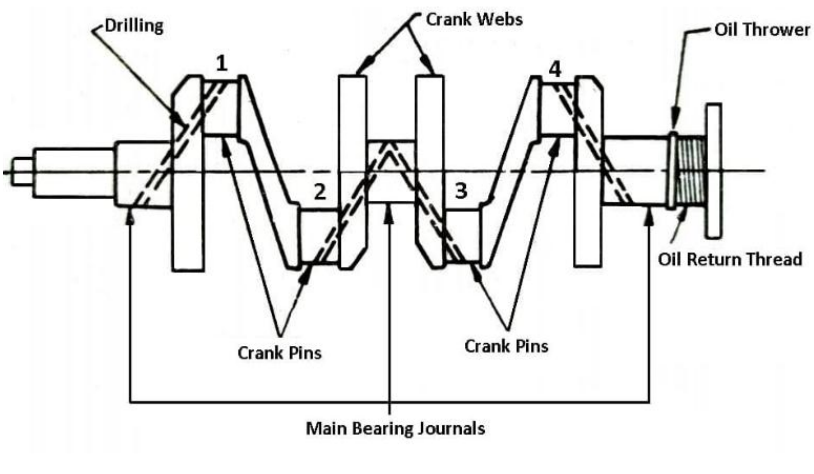


Figure 1.11. *Parts of* crankshaft

The crankshaft is the crank that receives the thrust from the connecting rod and provides rotary motion from the reciprocating motion of the piston. The force exerted by the connecting rod applies torque to the crankshaft, which is then converted to engine torque at its end. At one end of the crankshaft, this engine torque is used to propel the vehicle. At the other end, a portion of the available torque is used to power the engine's auxiliary components.

1.4.2.4. Camshaft

A camshaft is a mechanical device that synchronizes multiple movements. It consists of a shaft equipped with several cams. It transforms the continuous rotational motion of the shaft into an alternating linear motion (for example, of a valve) or an alternating rotational motion (for example, of a rocker arm). The camshaft is driven by the crankshaft and has cams that act on the valve lifters to control their opening.

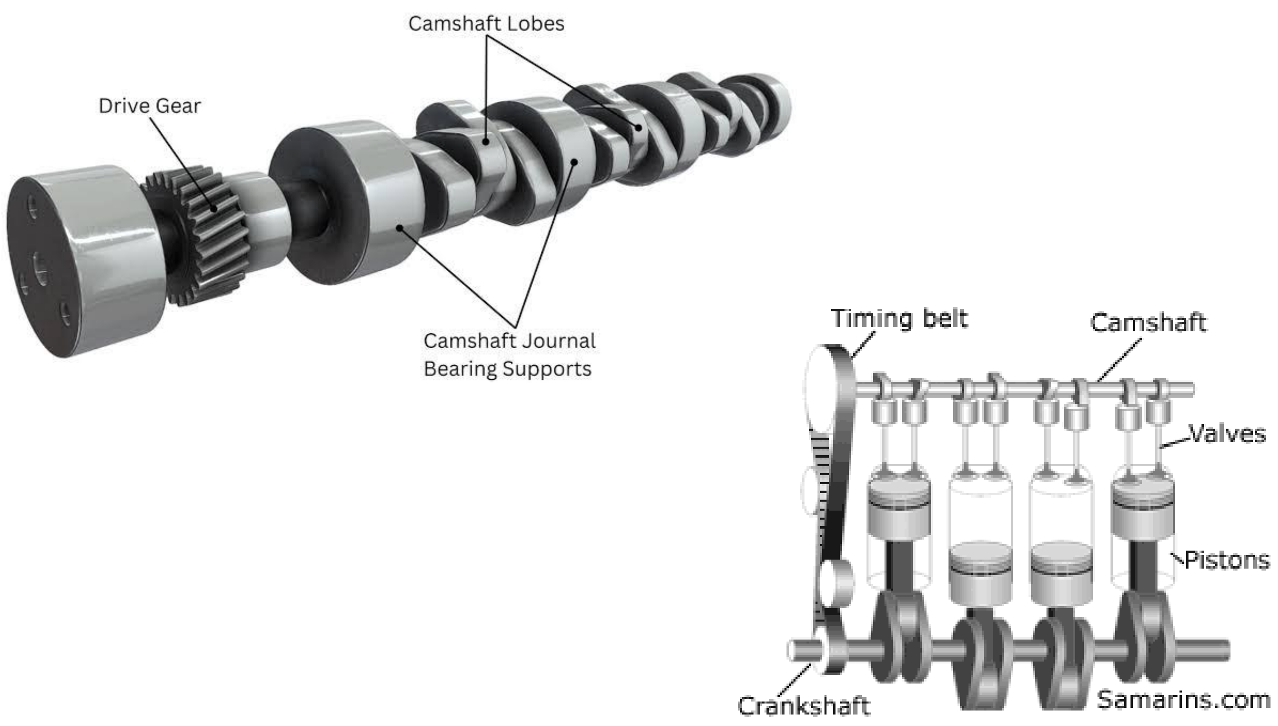


Figure 1.12. *Parts of camshaft*

1.4.2.5. Inlet and exhaust valves:

The valve is a metal plug made of nickel, chromium, or tungsten steel whose role is to open and close the intake and exhaust ports to allow compression, combustion, and the evacuation of burnt gases. It consists of a stem and a head.

The stem, also called the shank, is cylindrical and connected to the head by a large-radius fillet to reduce stress and thus reinforce the critical cross-section. The stem guides the valve during its reciprocating linear movement in a guide that is mounted either in the block or the cylinder head. The head, truncated cone-shaped, rests on a seat via a conical section to ensure the closure and sealing of the intake or exhaust port.

The head is characterized by its seat angle, which is 30° or 45° depending on the engine type, and can be flat or domed.

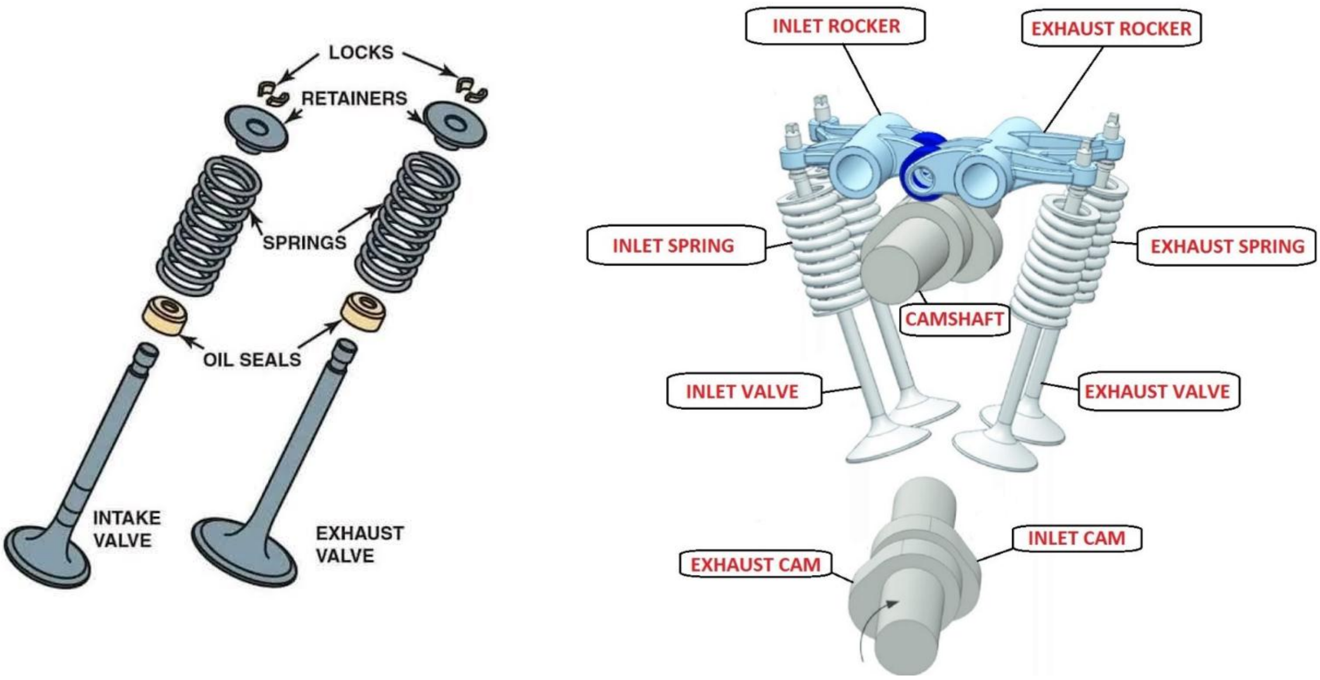


Figure 1.13. *Inlet and exhaust valves*

Inlet Valve: Allows the air or air–fuel mixture to enter the cylinder.

Exhaust Valve: Allows burnt gases to leave the cylinder.

Valves are subjected to significant thermal stress. Even though they benefit from the cooling effect of the intake gases, an intake valve can easily reach an operating temperature of 500°C. On an exhaust valve, the temperature can rise to 800°C. Exhaust valves are sometimes cooled with sodium. The sodium is incorporated into the hollow stem and dissipates heat from the valve head to the valve stem. The valve consists of a head and a stem.

The valve head seals the cylinder in conjunction with the valve seat. The valve stem is guided within the cylinder head by a valve guide. A seal at the top of the valve stem prevents oil from entering the combustion chamber. A valve spring ensures the valve closes. The sealing surface must be as narrow as possible to achieve optimal seating pressure. A sealing surface that is too narrow will cause the valve head to burn up because the heat is not adequately dissipated.

Timing gear: Timing gear is a combination of gears, one gear of which is mounted at one end of the camshaft and the other gear at the crankshaft. Camshaft gear is bigger in size than that of the crankshaft gear and it has twice as many teeth as that of the crankshaft gear. For this reason, this gear is commonly called half time gear. Timing gear controls the timing of ignition, timing of opening and closing of valve as well as fuel injection timing.

Inlet manifold: It is that part of the engine through which air or air-fuel mixture enters into the engine cylinder. It is fitted by the side of the cylinder head.

Exhaust manifold: It is that part of the engine through which exhaust gases go out of the engine cylinder. It is capable of withstanding high temperature of burnt gases. It is fitted by the side of the cylinder head.

1.5. Internal combustion engine terminologies

a. **Bore:** Bore is the diameter of the engine cylinder.
b. **Stroke:** It is the linear distance traveled by the piston from top dead center (TDC) to bottom dead center (BDC).
c. **Stroke-bore ratio:** The ratio of length of stroke (L) and diameter of bore (D) of the cylinder is called stroke-bore ratio (L/D). In general, this ratio varies between 1 to 1.45 and for tractor engines, this ratio is about 1.25.
d. **Swept volume**: It is the volume (A x l) displaced by one stroke of the piston where A is the cross sectional area of piston and l is the length of stroke ($A = Л.d^2/4$)
e. **Top dead center:** When the piston is at the top of its stroke, it is said to be at the top dead center (TDC),
f. **Bottom dead center:** when the piston is at the bottom of its stroke, it is said to be at its bottom dead center (BDC).

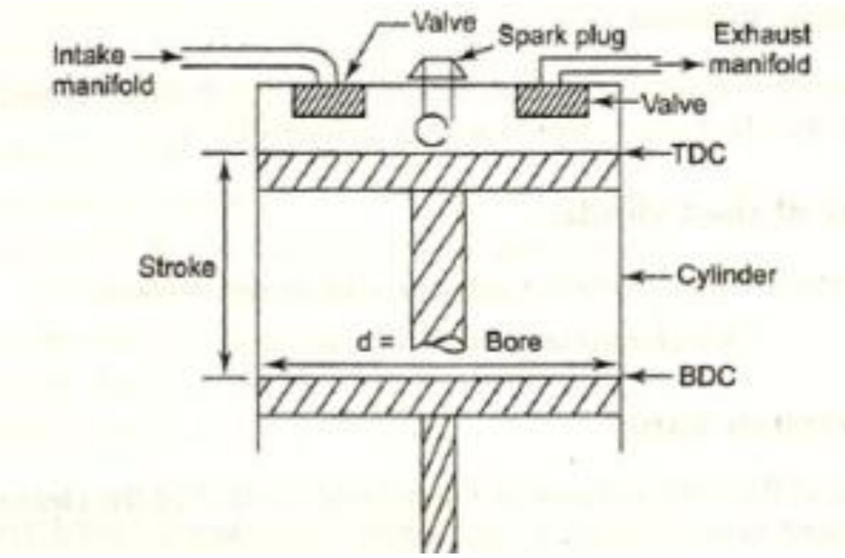


Figure 1.14. *Diagram of combustion chamber*

1.5.1. Compression ratio

It is the ratio of the volume of the cylinder at the beginning of the compression stroke to that at the end of compression stroke, i.e. ratio of total cylinder volume to clearance volume. The Compression ratio of diesel engine varies from 14:1 to 25:1 and that of carburetor type engine (spark ignition engine) varies from 4:1 to 12:1.

1.5.2. Power

It is the rate of doing work. S.I. unit of power is watt. Watt = Joule/sec. (4.2 Joules = 1 Calorie). In metric unit, the power can be expressed in kg.m/sec.

1.5.3. Horse power (HP)

It is the rate of doing work. Expressed in horse power Conversion factors from work to power 4500 kg m of work /minute = 1.0 hp 75 kg. m of work /second = 1.0 hp.

a. **Indicated horse power (IHP):** It is the power generated in the engine cylinder and received by the piston. It is the power developed in a cylinder without accounting frictional losses.
b. **Brake horse power (BHP):** It is the power delivered by the engine at the end of the crankshaft. It is measured by a dynamometer.

1.6. Cycle stroke of engine (diesel/ petrol engine)

1.6.1. Four stroke cycle engine

In four stroke cycle engines the four events namely suction, compression, power and exhaust take place inside the engine cylinder. The four events are completed in four stroke of the piston (two revolutions of the crank shaft).

This engine has got valves for controlling the inlet of charge and outlet of exhaust gases. The opening and closing of the valve is controlled by cams, fitted on camshaft. The camshaft is driven by crankshaft with the help of suitable gears or chains. The camshaft runs at half the speed of the crankshaft. The events taking place in I.C. engine are as follows:

1. Suction stroke ; 2. Compression stroke; 3. Power stroke ; 4. Exhaust stroke

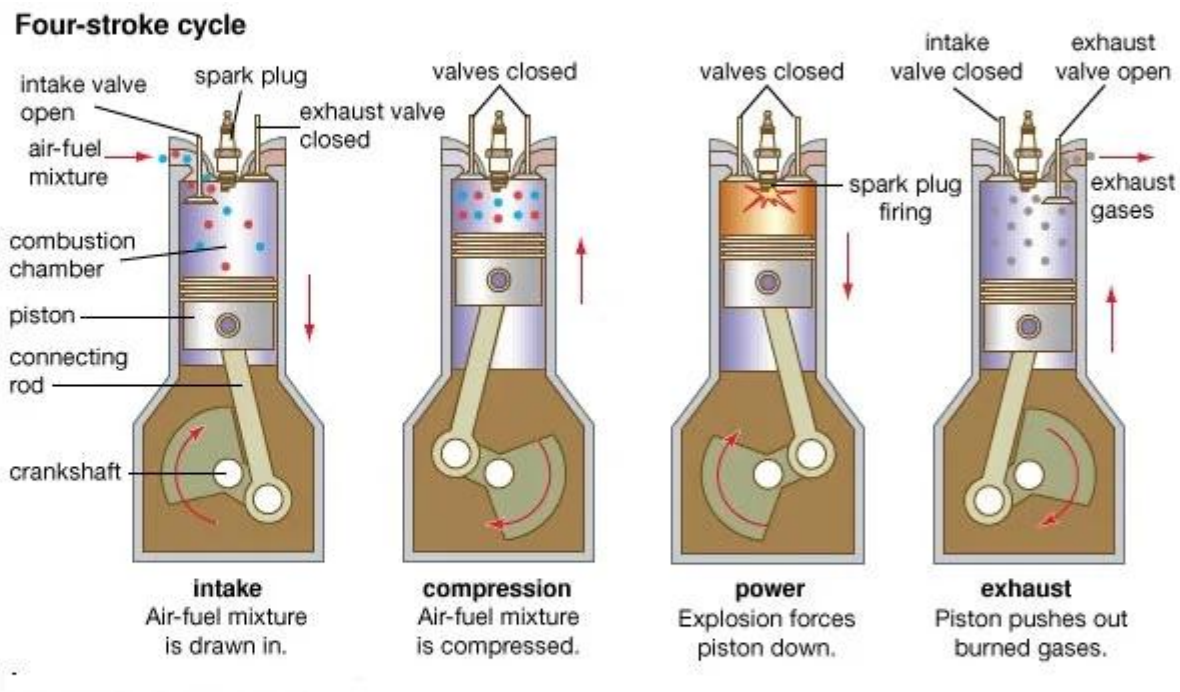


Figure 1.15. *Four stroke engine cycle*

1.6.1.1. Intake stroke (Suction stroke)

During suction stroke inlet valve opens and the piston moves downward. Only air or a mixture of air and fuel are drawn inside the cylinder. The exhaust valve remains in closed position during this stroke. The pressure in the engine cylinder is less than atmospheric pressure during this stroke.

1.6.1.2. Compression stroke

During this stroke the piston moves upward. Both valves are in closed position. The charge taken in the cylinder is compressed by the upward movement of piston. If only air is compressed, as in case of diesel engine, diesel is injected at the end of the compression stroke and ignition of fuel takes place due to high pressure and temperature of the compressed air. If a mixture of air and fuel is compressed in the cylinder, as in case of petrol engine, the mixture is ignited by a spark plug.

1.6.1.3. Power stroke

After ignition of fuel, tremendous amount of heat is generated, causing very high pressure in the cylinder which pushes the piston downward. The downward movement of the piston at this instant is called power stroke. The connecting rod transmits the power from piston to the crank shaft and crank shaft rotates. Mechanical work can be taped at the rotating crank shaft. Both valves remain closed during power stroke.

1.6.1.4. Exhaust stroke

During this stroke piston moves upward. Exhaust valve opens and exhaust gases go out through exhaust valves opening. All the burnt gases go out of the engine and the cylinder becomes ready to receive the fresh charge. During this stroke inlet valve remains closed. Thus it is found that out of four stroke, there is only one power stroke and three idle stroke in four stroke cycle engine. The power stroke supplies necessary momentum for useful work.

1.6.2. Two stroke cycle engine (petrol engine)

In two stroke cycle engines, the whole sequence of events i.e., suction, compression, power and exhaust are completed in two stroke of the piston i.e. one revolution of the crankshaft. There is no valve in this type of engine. Gas movement takes place through holes called ports in the cylinder. The crankcase of the engine is air tight in which the crankshaft rotates.

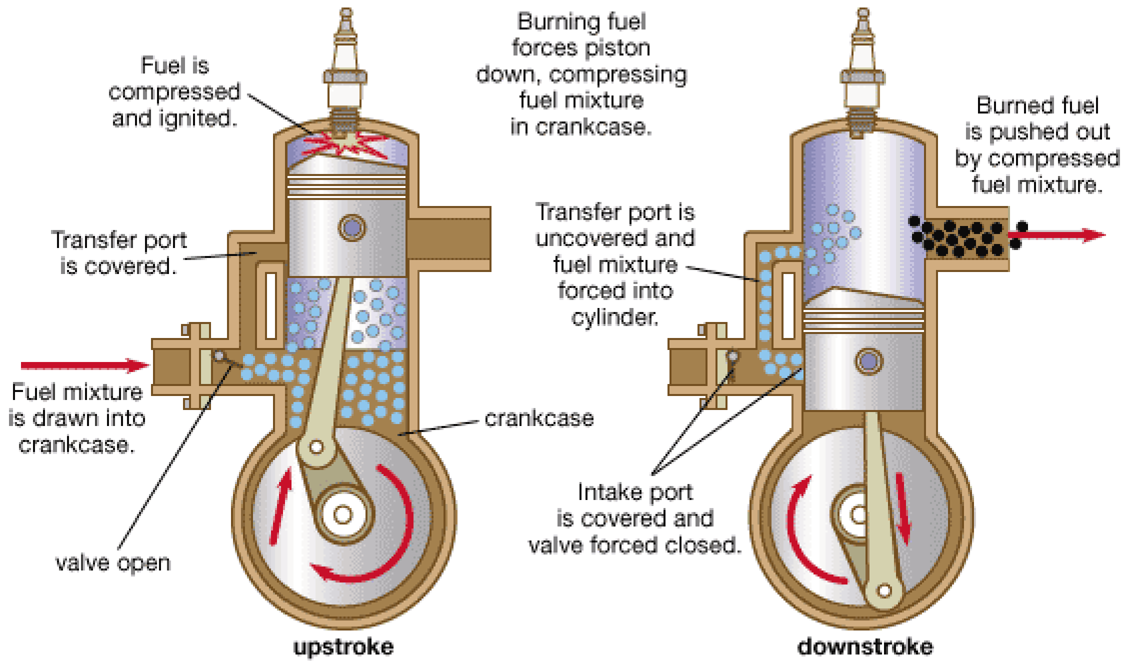


Figure 1.16. *Two stroke engine cycle*

1.6.2.1. Upward stroke of the piston (Suction + Compression)

When the piston moves upward, it covers two of the ports, the exhaust port and transfer port, which are normally almost opposite to each other. This traps the charge of air- fuel mixture drawn already in to the cylinder. Further upward movement of the piston compresses the charge and uncovers the suction port. Now fresh mixture is drawn through this port into the crankcase. Just before the end of this stroke, the mixture in the cylinder is ignited by a spark plug Thus, during this stroke both suction and compression events are completed.

16.2.2. Downward stroke (Power + Exhaust)

Burning of the fuel rises the temperature and pressure of the gases, which forces the piston to move down the cylinder. When the piston moves down, it closes the suction port, trapping the fresh charge drawn into the crankcase during the previous upward stroke. Further downward movement of the piston uncovers first the exhaust port and then the transfer port. Now fresh charge in the crankcase moves in to the cylinder through the transfer port driving out the burnt gases through the exhaust port. Special shaped piston crown deflect the incoming mixture up around the cylinder so that it can help in driving out the exhaust gases. During the downward stroke of the piston power and exhaust, events are completed.

1.6.3. Firing order of four stroke ICE

You probably already know that an engine has pistons which convert their reciprocating motion (up and down motion) to rotary (rotational motion) of the crankshaft. The power to turn the crankshaft is made available to the piston by the occurrence of combustion inside the combustion chambers (cylinders) that house the pistons. The combustion event, and therefore the movement of the pistons, must be coordinated to ensure continuous production of power as long as the ignition is on, engine is running, and all other enabling conditions are met. The sequence in which the cylinders generate power is called the **firing order**, the **order** in which the cylinders are **fired**. Most engines today are classified as four stroke engines where stroke refers to the up or down travel of a piston. The four stages/strokes are intake, compression, power and exhaust strokes. Therefore, while one cylinder is on the intake stroke, another is on the compression stroke, another on the power stroke and yet another on the exhaust stroke.

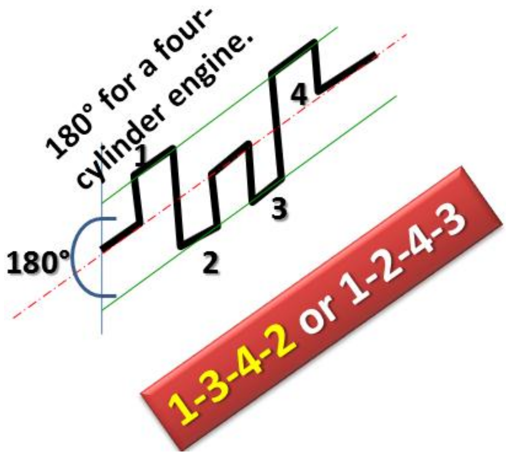
1.6.3.1. Firing order of four cylinder engine

The Firing Order of 4 cylinder engine is the sequence in which each cylinder burns the air–fuel mixture to produce power. In easy words, it explains which cylinder fires first and how the others follow. When the firing order of 4 cylinder engine is set correctly, the power strokes are evenly spaced, the engine runs smoothly, and unwanted vibrations are reduced. This sequence is important because it directly affects how stable, efficient, and refined the engine feels. The firing order of 4 cylinder engine is decided by the design of the crankshaft, the movement of the pistons, and the timing of the ignition system.

The engine firing order in internal combustion engines determines the ignition sequence for the cylinders in spark ignition. It governs the distribution of power in a multi-cylinder reciprocating engine, achieved by appropriately positioning spark plugs in gasoline engines or sequencing fuel injection in diesel engines.

Selecting an appropriate firing order is crucial during engine design to minimise vibration, ensure engine durability, enhance user comfort, and heavily influence crankshaft design. The firing order dictates the sequence of electrical impulses delivered to different cylinders, aiming to achieve balance and minimise vibration as much as possible. In the case of a radial engine, the firing order follows a specific pattern to align with the motion of the crank's throw during rotation, ensuring optimal performance and smooth operation.

Choosing the firing order is an essential part of engine design. Manufacturers carefully decide firing orders to tame vibrations and improve heat dissipation. The firing order also impacts ride quality (smoothness of ride), engine balance and engine sound. All these factors, except perhaps engine sound, decidedly play a role in extending an engine's fatigue life. However, many piston heads consider engine sound an essential part of engine design.



	C #1	C #2	C #3	C #4
0 180°	PE	Exh	Comp	Int
180° 360°	Exh	Int	PE	Comp
360° 540°	Int	Comp	Exh	PE
540° 720°	Comp	PE	Int	Exh

PE: Explosion-expansion, Exh: Exhaust, Comp: Compression, Int: Intake

Figure 1.17. *Firing order of 4 cylinder engine cycle*

Most 4-cylinder engines have a firing order of 1-3-4-2 although other firing orders such as 1-3-2-4, 1-4-3-2, 1-2-4-3 are possible. Consider the inline 4 engine in Figure 1.17.

In Figures 1.18a through 1.18e**,** the 720 degrees of crankshaft rotation has been broken into 180-degree *(720/ number of cylinders = 720/4 =180°)* intervals to aid illustration. In figures 1.18a through 1.18d, the first column contains the cylinder numbers (not in the firing order).

In Figure 1.18a, cylinder 1 starts off with the power stroke. Since the firing order is 1-3-4-2, it means the next cylinder to fire will be cylinder 3. It follows from figure 1.17 that if cylinder 1 is on the power stroke (p) and cylinder 3 is the next to fire, it should be on the stroke before the power stroke because it is preparing to fire after cylinder 1. This is the compression stroke (c) – read figure 1.17 in a direction opposite to the arrows' direction, counterclockwise.

Cylinder 4 which fires after cylinder 3 should be two strokes behind the power stroke on cylinder 1. Examining Figure 1.17 again should help deduce that cylinder 4 should be on the intake stroke.

Now cylinder 2 should be behind the power stroke on cylinder 1. That would put cylinder 2 on the exhaust stroke. All of this happens in the first 180 degrees of crankshaft rotation

	Crankshaft rotation in degrees			
Cylinder #	**180**	360	540	720
1	p			
2	e			
3	c			
4	i			

Figure a

	Crankshaft rotation in degrees			
Cylinder #	180	**360**	540	720
1	p	e		
2	e	i		
3	c	p		
4	i	c		

Figure b

	Crankshaft rotation in degrees			
Cylinder #	180	360	**540**	720
1	p	e	i	
2	e	i	c	
3	c	p	e	
4	i	c	p	

Figure c

	Crankshaft rotation in degrees			
Cylinder #	180	360	540	**720**
1	p	e	i	c
2	e	i	c	p
3	c	p	e	i
4	i	c	p	e

Figure d

	Crankshaft rotation in degrees			
Cylinder #	**180**	**360**	**540**	**720**
1	p	e	i	c
3	c	p	e	i
4	i	c	p	e
2	e	i	c	p

Figure e

Figure 1.18. *The steps of 4 cylinder engine firing order*

1.6.3.2. Firing order of three cylinder engine

A 3-cylinder engine has fewer cylinders, so maintaining balance is a bit difficult. The firing order is selected to reduce vibration and improve smoothness. In figures 1.19, the 720 degrees of crankshaft rotation has been broken into 240-degree (720/ number of cylinders = 720/3 =240°) intervals to aid illustration.

- **1-2-3 firing order**

  This is the simplest sequence where cylinders fire one after another. It is easy to design but produces more vibration.

- **1-3-2 firing order**

  This is more balanced because the middle cylinder fires between the outer ones, reducing engine shaking.

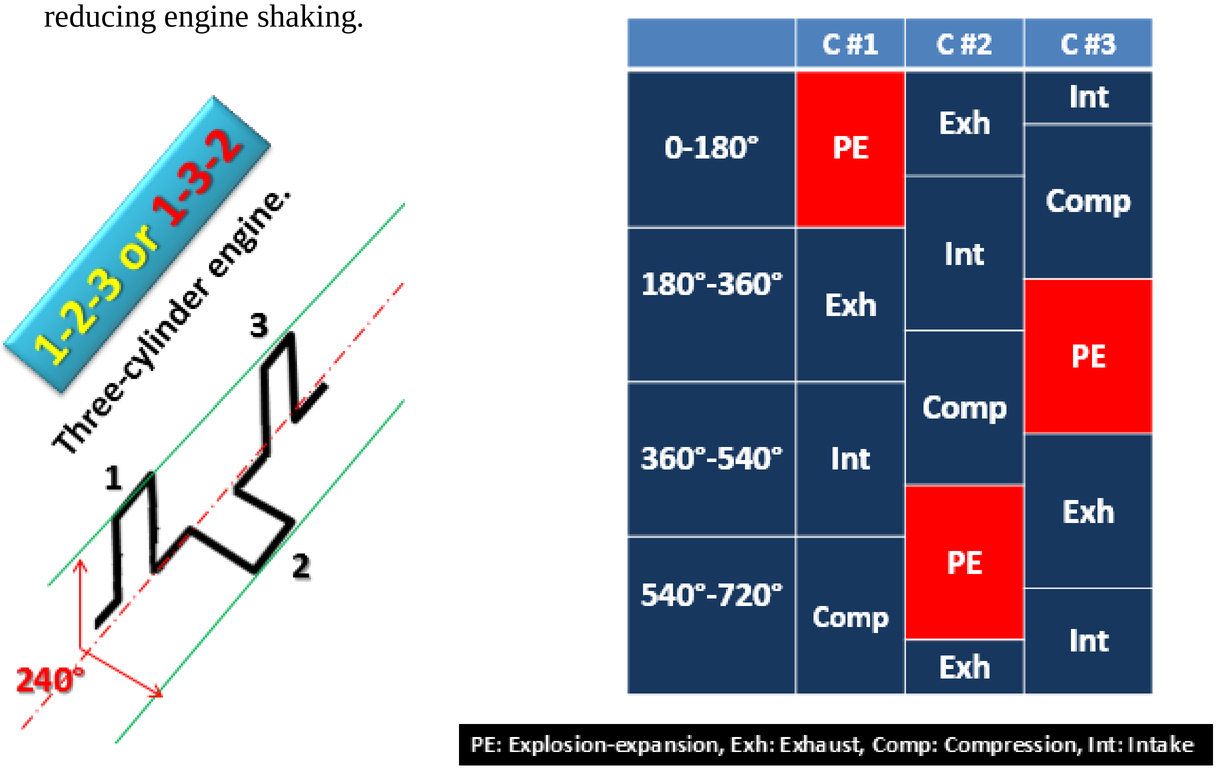


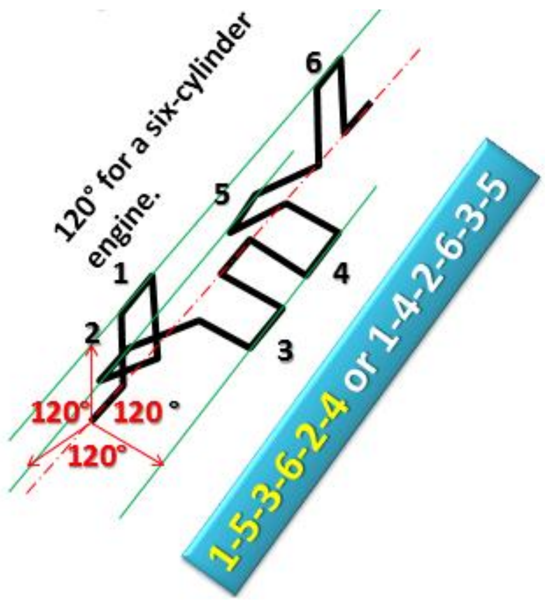
Figure 1.19. *Firing order of 3 cylinder engine*

Examples:

- Saab two-stroke engine → 1-2-3
- BMW K75 engine → 1-3-2

1.6.3.3. Firing order of six cylinder engine

Figure 1.20, illustrates the firing orders for a 6-cylinder engine with firing order 1-5-3-6-2-4. This is the firing order on the Mercedes Benz M272-E35 engine which has powered ML350 vehicles since 2006. It also powers the R350 vehicle and other Mercedes Benz vehicles.



<table>
<tr><th></th><th>C #1</th><th>C #2</th><th>C #3</th><th>C #4</th><th>C #5</th><th>C #6</th></tr>
<tr><td rowspan="3">0-180°</td><td rowspan="3">PE</td><td rowspan="2">Exh</td><td>Int</td><td>PE</td><td rowspan="2">Comp</td><td rowspan="3">Int</td></tr>
<tr><td rowspan="3">Comp</td><td rowspan="3">Exh</td></tr>
<tr><td rowspan="3">Int</td><td rowspan="3">PE</td></tr>
<tr><td rowspan="3">180°-360°</td><td rowspan="3">Exh</td><td rowspan="3">Comp</td></tr>
<tr><td rowspan="3">PE</td><td rowspan="3">Int</td></tr>
<tr><td rowspan="3">Comp</td><td rowspan="3">Exh</td></tr>
<tr><td rowspan="3">360°-540°</td><td rowspan="3">Int</td><td rowspan="3">PE</td></tr>
<tr><td rowspan="3">Exh</td><td rowspan="3">Comp</td></tr>
<tr><td rowspan="3">PE</td><td rowspan="3">Int</td></tr>
<tr><td rowspan="3">540°-720°</td><td rowspan="3">Comp</td><td rowspan="3">Exh</td></tr>
<tr><td rowspan="2">Int</td><td rowspan="2">PE</td></tr>
<tr><td>Exh</td><td>Comp</td></tr>
</table>

PE: Explosion-expansion, Exh: Exhaust, Comp: Compression, Int: Intake

Figure 1.20. *Firing order of 6 cylinder engine*

From figure 1.20 cylinder 1 fires in the first.

- In the next 120 degrees, as cylinder 1 moves from the power stroke to the exhaust stroke, cylinder 5 fires.
- In the next 120 degrees (240 degrees), as cylinder 5 moves from the power stroke to the exhaust stroke, cylinder 3 fires.
- In the next 120 degrees (360 degrees), as cylinder 3 moves from the power stroke to the exhaust stroke, cylinder 6 fires.
- In the next 120 degrees (480 degrees), as cylinder 6 moves from the power stroke to the exhaust stroke, cylinder 2 fires.
- In the next 120 degrees (600 degrees), as cylinder 2 moves from the power stroke to the exhaust stroke, cylinder 4 fires.
- In the next 120 degrees (720 degrees), as cylinder 4 moves from the power stroke to the exhaust stroke, cylinder 1 fires again.

1.7. Fuel Type in ICE

An engine burns fuel as a source of energy. Various types of fuel will burn in an engine: gasoline, diesel fuel, gasohol, alcohol, liquefied petroleum gas, and other alternative fuels. Diesel and gasoline (petrol) are the two main fuels used in internal combustion engines (ICEs). Although both types of engines convert chemical energy into mechanical energy through combustion, they differ significantly in fuel properties, ignition process, efficiency, and applications.

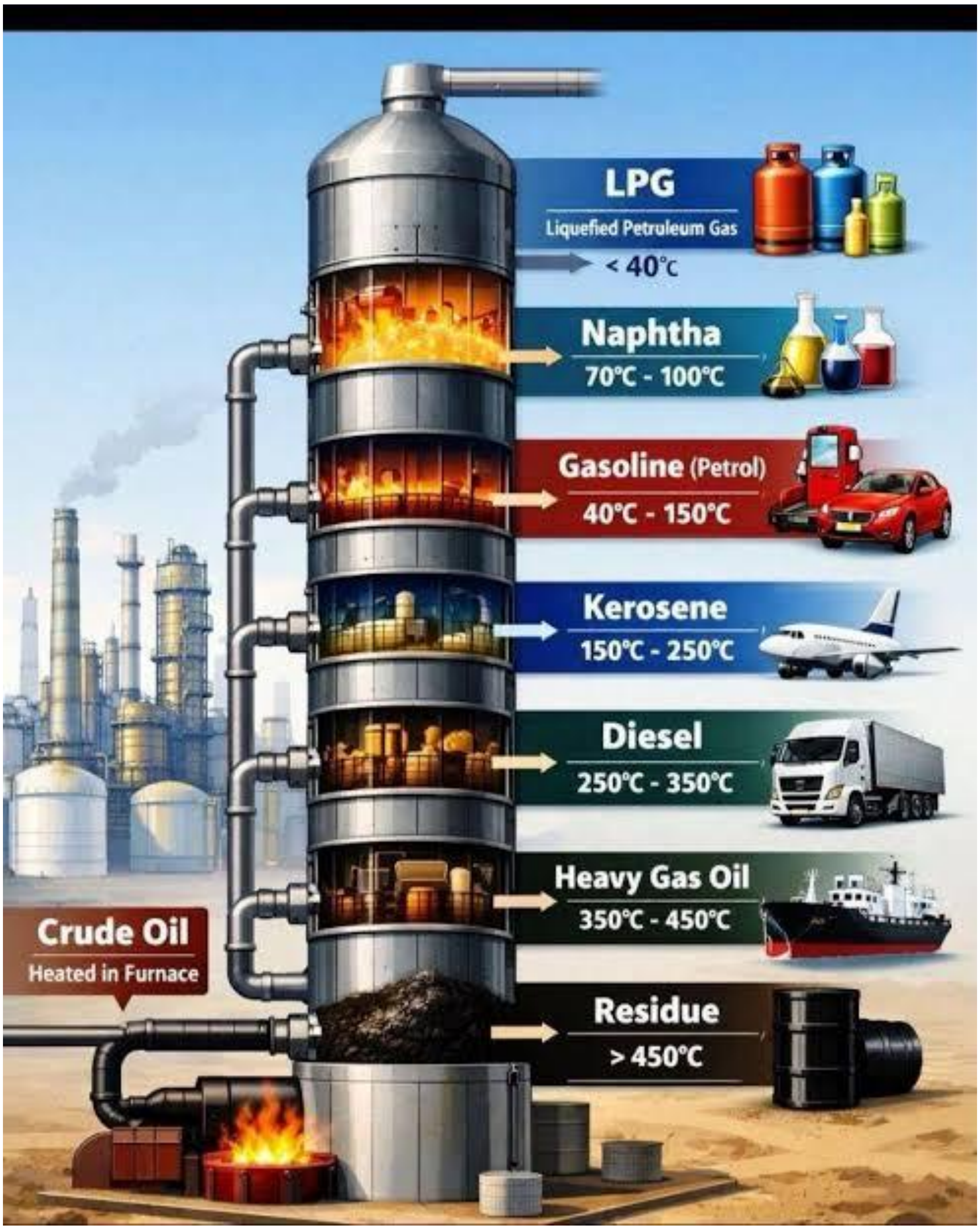


Figure 1.21. *Fuels used in ICEs*

Different types of fuels, such as gasoline, diesel, heating oil, and kerosene, are used for various applications depending on their specific chemical properties and the engines for which they are designed. Derived from the refining of crude oil, whose components are extracted in different fractions through distillation, each of these fuels has a distinct composition and use.

As a reminder, the chemical composition of crude oil varies depending on its geographical origin. Fuels are produced primarily by blending hydrocarbons in refineries . The resulting products undergo a number of transformations, notably through the use of synthetic molecules, to improve their quality and performance.

1.7.1. Petrol (Gasoline)

Gasoline is composed of a mixture of light hydrocarbons, primarily heptane (C_7H_{16}) , to which additives are often added to improve its performance and reduce the greenhouse gas emissions associated with its combustion. It is mainly used in spark-ignition engines . Cars that run on gasoline have naturally aspirated engines that generally operate between 1 and 40 bar and whose ignition is generated by an electric spark (spark plug).

It should be noted that gasoline is more volatile than other fuels, which allows for rapid and efficient combustion in the engines of passenger cars and motorcycles.

Several types of gasoline are available at the pump:

- **SP95 (Super 95):** classic fuel with an octane rating of 95, suitable for most petrol vehicles. It contains a small proportion of ethanol (up to 5%);
- **SP95-E10 (Super 95 with 10% ethanol):** similar to SP95 but with a higher ethanol content (up to 10%). It is compatible with most recent vehicles;
- **SP98 (Super 98):** fuel which has a higher octane rating than SP95 (98), offering better resistance to auto-ignition, making it ideal for high-performance engines;
- **E85 (Superethanol-E85):** fuel containing between 65% and 85% ethanol, intended for specific vehicles known as «Flex-Fuel ". This fuel is more environmentally friendly and economical but is only compatible with certain vehicles.

These different types of gasoline allow drivers to choose the fuel best suited to their vehicle at the service station, particularly based on compatibility, desired performance or environmental impact.

1.7.2. Diesel

Diesel, also called gas oil, is composed of heavier hydrocarbons distributed around ($C_{16}H_{34}$). Gas oils are improved by hydro-desulfurization, a hydrogen treatment that reduces their sulfur content.

It is used in compression-ignition engines : the fuel is injected at very high pressure (up to 2,000 bars) and explodes by compression, it is called " HDi " (High Pressure Direct Injection).

Diesel is heavier and less volatile than gasoline, with a slower combustion, which improves its fuel efficiency. These engines are more efficient than gasoline engines, making diesel more suitable for trucks, buses, and some commercial vehicles. According to IFPEN, " *under optimal operating conditions, a car engine today offers maximum efficiency: around 36% for a gasoline engine and 42% for a diesel engine*".

Several types of diesel are available at the pump:

- **Standard diesel :** conventional diesel fuel, used by the majority of diesel vehicles. It contains a small amount of biofuel (up to 7% fatty acid methyl esters, which is why it is also called "B7").
- **Premium diesel :** an enhanced version of standard diesel, often sold under specific brand names. It contains additional additives to improve engine performance, reduce emissions, and clean the injection system.
- **Non-road diesel (NRD or GNR):** used mainly for agricultural and construction equipment, this fuel is taxed less than road diesel.
- **B10 :** diesel containing up to 10% biofuel, it is used in some newer and compatible diesel vehicles.

As of January 1 , 2023, 53.4% of private vehicles in circulation in Europe ran on diesel (or with a non-rechargeable hybrid diesel engine, according to the latest figures from the Ministry of Energy in France).

1.7.3. Fuel Oil

Fuel oil is a fuel used primarily for domestic heating or to power certain types of industrial and marine engines.

It is similar to diesel but is generally of lower quality, with a higher sulfur content and fewer additives. Heating oil is less refined and, therefore, cheaper, but it is also more polluting.

1.7.4. Kerosene

Kerosene is a lighter fuel than diesel, used primarily in aviation to power jet engines . It is designed to operate efficiently at high temperatures and altitudes.

Kerosene has low volatility, making it safe for use in environments with extreme conditions, such as at high altitudes.

1.7.5 LPG

The term Liquefied Petroleum Gas or "LPG" refers to two gases in liquid form: propane and butane.

LPG is now widely used by industry, but also by individuals for their mobile activities. When blended, LPG can also be used as a vehicle fuel (LPG-c).

I.7.6. Biofuels

Biofuels are energy sources produced from biomass, the living matter of plants, or from biodegradable industrial waste. Depending on their composition, biofuels can be found in liquid or gaseous states and are primarily used in the transportation sector. The history of biofuels began in 1892 when Rudolph Diesel, the inventor of the diesel engine, first tested peanut oil in the engine he invented, the internal combustion engine, which bears his name. The major drawbacks of using vegetable oils are mainly related to their high viscosity (approximately 11–17 times higher than diesel fuel) and low volatility, which leads to the formation of deposits in engines. Therefore, they must be treated before being used as fuel. Biofuels are subdivided into: bioethanol, biodiesel, bio-oil, biogas, Fischer- Tropsch fluids , and biohydrogen. Biofuels represent a renewable energy source with little risk to the environment, thus constituting a sustainable development pathway. Indeed, they contribute to reducing greenhouse gas emissions and consequently improving air quality. Furthermore, they can replace fossil fuels.

1.8. Diesel and gasoline as Fuels in Internal Combustion Engines

Diesel and gasoline engines are both types of internal combustion engines, meaning they produce power by burning fuel inside a cylinder. However, the way they burn that fuel is fundamentally different. In a gasoline engine, air and fuel are mixed together and then ignited by a spark plug. In contrast, a diesel engine compresses only air to a very high pressure and temperature, and then fuel is injected into this hot air, causing it to ignite automatically without a spark.

Another major difference lies in the nature of the fuels themselves. Gasoline is lighter, more volatile, and burns quickly, which allows engines to produce higher speeds and faster acceleration. On the other hand, diesel fuel is denser and contains more energy per unit, meaning it releases more energy during combustion. Because of this higher energy density, diesel engines are generally more fuel-efficient and can produce more work using less fuel compared to gasoline engines.

In terms of performance, diesel and gasoline engines deliver power in different ways. Diesel engines generate higher torque, which makes them ideal for heavy-duty applications such as trucks, buses, and construction equipment where strong pulling power is needed. Gasoline engines, however, operate at higher speeds (RPM) and are better suited for light vehicles that require quick acceleration and smoother driving.

Efficiency and durability also set the two apart. Diesel engines are typically more efficient due to their higher compression ratios and better fuel utilization, often achieving significantly better mileage than gasoline engines. They are also built stronger to withstand high pressure, which makes them more durable and longer-lasting. Gasoline engines, while less efficient, are usually lighter, cheaper, and simpler in design, making them more common in everyday passenger cars.

Finally, there are differences in emissions and environmental impact. Diesel engines generally produce less carbon dioxide (CO_2) per distance traveled, but they emit more nitrogen oxides (NOx) and particulate matter. Gasoline engines, in contrast, produce fewer particulates but more carbon monoxide and unburned hydrocarbons. This means each fuel type has its own environmental advantages and drawbacks.

In summary, diesel engines are more efficient, durable, and better for heavy work, while gasoline engines are faster, lighter, and better suited for everyday driving.

Table 1.1. *Comparison between a diesel engine and a petrol engine*

Feature	**Diesel Engine**	**Gasoline Engine**
Ignition	Compression ignition, no spark plugs required	Spark ignition with spark plugs
Compression Ratio	Higher (14:1 to 25:1) witch greater thermal efficiency	Lower (4:1 to 12:1) witch less thermal efficiency
Combustion Control	Less controlled give higher vibration and noise	More controlled give smoother operation
Torque & Speed	Higher torque, lower speeds; peak torque at low RPM	Lower torque, higher speeds; peak torque at higher RPM
Engine Weight & Durability	Components must withstand higher pressures (heavier)	Lighter components; lower pressures
Maintenance	Longer intervals, higher costs	Shorter intervals, generally lower costs
Cold Start	More difficult in low temperatures	Easier in low temperatures
Overheating	Less frequent due to efficient operation	Can be more prone
Emissions	More soot and NOx, due to higher fuel sulfur/nitrogen and temperatures	More CO emissions, fewer particulates
CO_2 Emissions	Higher per liter of fuel, but lower per km due to efficiency	Lower per liter, but higher per km due to greater fuel consumption
Environmental Impact	Particulates and NOx make diesel less environmentally friendly	CO and overall CO_2 higher, but fewer particulates

1.9. CNG and LPG as Fuels in Internal Combustion Engines

1.9.1. CNG (Compressed Natural Gas)

- Main component: Methane (CH_4)
- Stored as a gas compressed to 200 to 250 bar**.**
- Much lighter than air.

1.9.1.1. Advantages

- Very clean fuel.
- Emits up to 80% less greenhouse gases compared to gasoline engines.
- Produces very little soot or carbon deposits.
- Contains no lead or benzene (keeps engine oil cleaner).
- Reduced spark plug fouling.

- In case of leakage, it disperses quickly (safer behavior).

1.9.1.2. Disadvantages

- Lower energy content compared to gasoline and diesel.
- Requires high-pressure storage cylinders (heavy and bulky).
- Higher combustion temperature (around 200°C when tuned for equal power).
- Increased valve wear due to low lubricating properties.
- Higher temperature accelerates oil oxidation and may reduce engine durability.

1.9.2. LPG (Liquefied Petroleum Gas)

- Composition: Propane (C_3H_8) ; Propylene (C_3H_6) ; Butane (C_4H_{10}) ; Butylene (C_4H_8)
- Obtained from crude oil refining (distillation process).
- Liquefied at 15°C under pressure of 1.7 to 7.5 bar**.**
- Heavier than air.

1.9.2.1. Advantages

- Produces about 25% less CO_2 than gasoline.
- Cleaner combustion than gasoline.
- Requires less storage space compared to CNG.
- Suitable for passenger vehicles.
- Lower carbon deposits inside engine.

1.9.2.2. Disadvantages

- Heavier than air, may accumulate near ground in case of leakage.
- Slight power loss (10% compared to gasoline).
- Lower lubricating properties, possible valve seat wear.
- Lower energy content than gasoline/diesel.

1.9.3. Energy Comparison

- Gasoline and diesel have higher energy content.
- LPG (especially propane) has about 2.5 times higher calorific value than CNG**.**
- Because LPG and CNG contain fewer hydrocarbon bonds, their overall energy density is lower than conventional fuels.

1.9.4. Engine Conversion

- Most gasoline (spark ignition) engines can be converted to operate on LPG or CNG.
- Converted engines experience slight power reduction due to lower energy density.
- Combustion is cleaner to less soot formation and cleaner engine internals.

Table 1.2. Comparison Between Gasoline, Diesel, LPG, and CNG

	Gasoline (Petrol)	**Diesel**	**LPG**	**CNG**
Main Composition	Hydrocarbons (C_4–C_{12})	Heavier hydrocarbons (C_{12}–C_{20})	Propane, Butane mixture	Methane (CH_4)
Engine Type	Spark Ignition (SI)	Compression Ignition (CI)	Spark Ignition (converted SI engines)	Spark Ignition (converted SI engines)
Ignition Method	Spark plug	Self-ignition (high compression)	Spark plug	Spark plug
Compression Ratio	Low (4:1–12:1)	High (14:1–25:1)	Medium	Medium–High
Fuel State in Storage	Liquid	Liquid	Liquefied gas (1.7 to 7.5 bar)	Compressed gas (200 to 250 bar)
Energy Content	High	Very High	Lower than gasoline	Lower than LPG
Thermal Efficiency	Moderate	High	Moderate	Moderate
Fuel Economy	Moderate	Good	Good	Very good
CO_2 Emissions	High	High (but efficient per km)	Around 25% less than gasoline	Up to 80% less than gasoline
Other Emissions	CO, HC, NOx	NOx, soot (particulates)	Cleaner than gasoline	Very low soot & particulates
Engine Power	High	High torque	10% less than gasoline	Slightly lower than gasoline
Engine Wear	Normal	Strong construction required	Possible valve wear	Possible valve wear
Lubrication Quality	Good	Very good	Lower	Lower
Safety in Leakage	Flammable liquid	Flammable liquid	Heavier than air (settles)	Lighter than air (disperses)
Storage Space Required	Small	Small	Moderate	Large (high-pressure cylinder)
Typical Applications	Cars, motorcycles	Trucks, buses, heavy machinery	Passenger cars, taxis	Buses, taxis, fleet vehicles

1.10. Conclusion

Internal combustion engines are heat machines that take heat from a high temperature source, convert part of it into mechanical work, and throw away the rest as waste heat. Their operating principle is simple: heat in to work out to waste heat out, and this cycle keeps repeating. Based on how combustion happens, heat engines are mainly classified into internal combustion engines, where fuel burns inside the engine (like petrol and diesel engines), and external combustion engines, where fuel burns outside (like steam engines).

For internal combustion engines, fuels like gasoline and diesel are used because they contain chemical energy that can be converted into heat during combustion. Gasoline burns quickly and is good for speed, while diesel burns slower and gives more power and efficiency. Inside the engine, both fuels do the same job produce heat that creates pressure to move the piston and generate mechanical work**.**

From the comparison between Gasoline, Diesel, LPG, and CNG, we can conclude that each fuel has advantages and disadvantages depending on application, cost, efficiency, and environmental impact. Gasoline engines provide good performance and smooth operation but produce higher emissions and moderate fuel economy. Diesel engines offer higher thermal efficiency, better fuel economy, and greater torque, making them suitable for heavy-duty and long-distance applications. However, they are heavier and may produce more NOx and particulate emissions. LPG is a cleaner alternative to gasoline, producing lower CO_2 emissions and fewer carbon deposits; though it results in slight power loss and possible valve wear. CNG is the cleanest fuel among the four, with very low greenhouse gas emissions and minimal engine deposits. However, it requires high-pressure storage tanks and may reduce engine durability due to higher combustion temperatures.

1.11. Exercises

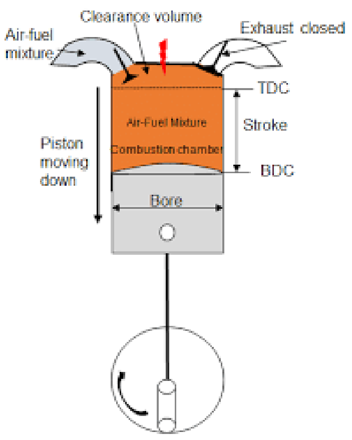
1.11.1. Exercise 01

Consider the following table, which presents the four strokes of a 4-cylinder engine cycle:

1. Complete the table.
2. Deduce the firing order.

	0 → 180 °CA	180 → 360°CA	360 → 540 °CA	540 → 720 °CA
Cylinder 1				
Cylinder 2	Intake			
Cylinder 3				
Cylinder 4	Exhaust			

1.11.2. Exercise 02



A 4-cylinder internal combustion engine has the following characteristics: a bore of 89 mm; a stroke of 95.6 mm; and a compression ratio of 18.

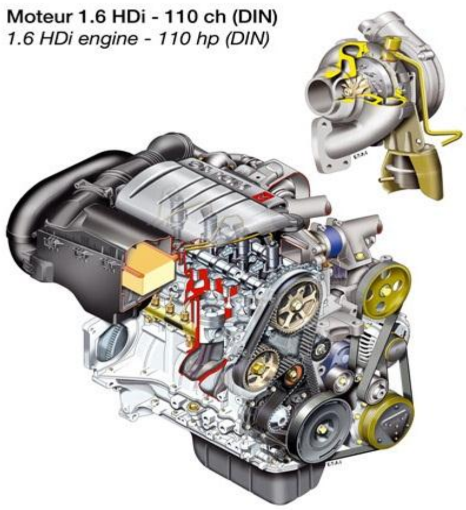
1. This engine runs on diesel or petrol fuel.
2. Calculate the clearance volume of this engine.
3. Calculate the total volume of the engine.

1.11.3. Exercise 03

A three-cylinder in-line four-stroke engine, fitted to the Peugeot 301/Citroën C-Elysée, has the following characteristics: bore = 7.5 cm, stroke = 9.05 cm, compression ratio = 10, Torque 110 N.m at 3000 rpm, power = 53 kW (72 hp) at 5000 rpm.

1. What fuel is used in this engine (Gasoline, Diesel), justify your answer.
2. Calculate the clearance volume of this engine, then the total volume.
3. What is the firing order of this engine? Justify your answer with a table.

1.11.4. Exercise 04



A Peugeot 1.6 HDI 4-cylinder engine has the following characteristics: a bore of 75 mm; a stroke of 88.6 mm; and a compression ratio of 18.3.

1. This engine runs on diesel or petrol fuel.
2. Calculate the clearance volume of this engine.
3. Calculate the total volume of this engine.
4. What is the most commonly used firing order for a 4-stroke cycle of this engine? Justify your answer with a table.

1.11.5. Exercise 05

A 6-cylinder Schweitzer-GV engine has the following characteristics: a bore of 12 cm; a stroke of 14.5 cm; combustion chamber volume $V_0 = 102\ cm^3$.

1. Calculate the compression ratio and the total volume of this engine.
2. What is the most commonly used firing order? Justify your answer with a table.
3. Determine the effective power, for a speed N = 1500 rpm and a motor torque C=22 Nm.

1.11.6. Exercise 06

Consider a 6-cylinder V-engine operating on a four- stroke cycle at an engine speed of 3750 rpm; the engine develops a torque of 200 Nm at... Mechanical efficiency of 85%. The engine specifications are as follows :

Total displacement is 3 liters. Compression ratio is 9.5.

Connecting rod length is 166 mm. The engine is a square type (stroke = bore diameter).

Calculate:

1. Clearance volume.
2. Bore diameter and stroke.
3. Average piston speed.

1.12. Solution of exercises

Solution of exercise 01

Consider the following table, which presents the four strokes of a 4-cylinder engine cycle:

1. Complete the table.

	0 → 180 °CA	180 → 360°CA	360 → 540 °CA	540 → 720 °CA
Cylinder 1	Compression	Power-Expansion	Exhaust	Intake
Cylinder 2	Intake	Compression	Power-Expansion	Exhaust
Cylinder 3	Power-Expansion	Exhaust	Intake	Compression
Cylinder 4	Exhaust	Intake	Compression	Power-Expansion

2. Deduce the firing order: **3, 1, 2, 4**

Solution of exercise 02

1. Type of fuel (diesel or petrol). The compression ratio is 18, which is quite high.

For the petrol engines $05 \leq CR \leq 12$; CR : compression ration

For the diesel engines $14 \leq CR \leq 25$

So this engine runs on **diesel fuel**.

2. Clearance volume

First compute the swept volume (displacement per cylinder):

Bore ; D=89 mm=0.089 m

Stroke s =95.6 mm=0.0956 m

$$V_d = \frac{\pi . D^2}{4} . S = \frac{\pi * 89^2}{4} * 95.6 = 594.74\ cm^3$$

$$CR = \frac{V_d + V_c}{V_c} \Rightarrow V_c = \frac{V_d}{CR - 1} = \frac{594.74}{18 - 1} = 34.984 \approx 35\ cm^3$$

3. Total engine volume

$$V_T = n.(V_d) = 4 * (594.74) = 2378.96\ cm^3 \approx 2.4\ L$$

Solution of exercise 03

1. Type of fuel (diesel or petrol). The compression ratio is 10, which is quite high.

For the petrol engines 05 $\leq CR \leq$ 12 ; CR : compression ration

For the diesel engines 14 $\leq CR \leq$ 25

Since **10** falls squarely in the gasoline range, this engine uses gasoline (petrol). Also, small 3-cylinder engines like those in the Peugeot 301 and Citroën C-Elysée are commonly **spark-ignition (gasoline) engine.**

2. Clearance volume

First compute the swept volume (displacement per cylinder):

Bore ; D= 7.5 cm

Stroke s = 9.05 cm

$$V_d = \frac{\pi . D^2}{4} . S = \frac{\pi * 7.5^2}{4} * 9.05 = 399.82\ cm^3$$

$$CR = \frac{V_d + V_c}{V_c} \Rightarrow V_c = \frac{V_d}{CR - 1} = \frac{399.82}{10 - 1} = 44.42\ cm^3$$

Total engine volume

$$V_T = n.(V_d) = 4 * (399.82) = 1199.46\ cm^3 \approx 1.2\ L$$

3. The firing order is **1, 2 ,3** (3- cylinder engine)

<table>
<tr><td></td><td colspan="12">0 60 120 180 240 300 360 420 480 540 600 660 720</td></tr>
<tr><td>Cylinder 1</td><td colspan="3">Power-Expansion</td><td colspan="3">Exhaust</td><td colspan="3">Intake</td><td colspan="3">Compression</td></tr>
<tr><td>Cylinder 2</td><td>Int</td><td colspan="3">Compression</td><td colspan="3">Power-Expansion</td><td colspan="3">Exhaust</td><td colspan="2">Intake</td></tr>
<tr><td>Cylinder 3</td><td colspan="2">Exhaust</td><td colspan="3">Intake</td><td colspan="3">Compression</td><td colspan="3">Power-Expansion</td><td>Exhaust</td></tr>
</table>

Solution of exercise 04

1. Fuel type

The engine is labeled 1.6 HDi.

"HDi" stands for High-pressure Direct Injection, a technology used by diesel engines.This engine runs on diesel fuel (CR = 18.3 ; $14 \leq CR \leq 25$)

2. Clearance volume

First compute the swept volume (displacement per cylinder):

Bore ; D= 7.5 cm

Stroke s = 8.86 cm

$$V_d = \frac{\pi . D^2}{4} . S = \frac{\pi * 7.5^2}{4} * 8.86 = 391.42\ cm^3$$

$$CR = \frac{V_d + V_c}{V_c} \Rightarrow V_c = \frac{V_d}{CR - 1} = \frac{391.42}{18.3 - 1} = 22.62\ cm^3$$

3. Total engine volume

$$V_T = n.(V_d) = 4 * (391.42) = 1565.69\ cm^3 \approx 1.56\ L$$

4. Firing order

The most common firing order for inline 4-cylinder 4-stroke engines is: **1, 3, 4, 2**

	0 → 180 °CA	180 → 360°CA	360 → 540 °CA	540 → 720 °CA
Cylinder 1	Power-Expansion	Exhaust	Intake	Compression
Cylinder 2	Exhaust	Intake	Compression	Power-Expansion
Cylinder 3	Compression	Power-Expansion	Exhaust	Intake
Cylinder 4	Intake	Compression	Power-Expansion	Exhaust

Solution of exercise 05

1. Compression ration

First compute the swept volume (displacement per cylinder):

Bore ; D= 12 cm

Stroke S= 14.5 cm

$$V_d = \frac{\pi . D^2}{4} . S = \frac{\pi * 12^2}{4} * 14.5 = 1639.91\ cm^3$$

$$CR = \frac{V_d + V_c}{V_c} \Rightarrow CR = \frac{V_d}{V_c} + 1 = \frac{1639.91}{102} + 1 = 17.077 \approx 17.1$$

2. Firing order

The most common firing order for an inline 6-cylinder engine is: **1, 5, 3, 6, 2, 4**

With 6 cylinders: 720°/6=120° ; One power stroke every 120°, giving:

	0 60 120 180 240 300 360 420 480 540 600 660 720				
Cylinder 1	Power-Expansion	Exhaust	Intake	Compression	
Cylinder 2	Exhaust	Intake	Compression	Power-Expansion	Exhaust
Cylinder 3	Intake	Compression	Power-Expansion	Exhaust	Intake
Cylinder 4	P-E	Exhaust	Intake	Compression	Power-Expansion
Cylinder 5	Comp	Power-Expansion	Exhaust	Intake	Compression
Cylinder 6	Intake	Compression	Power-Expansion	Exhaust	

3. Step 1: Angular speed

$w = 2.\pi.\frac{1500}{60} = 157.1\ rad/s$; P=22×157.1≈3456 W ; P≈3.46 kW

Solution of exercise 06

1. Clearance volume

$$CR = \frac{V_d + V_c}{V_c} \Rightarrow V_c = \frac{V_d}{CR - 1}$$

$$V_d = \frac{3000}{6} = 500\ cm^3$$

$$CR = \frac{V_d + V_c}{V_c} \Rightarrow V_c = \frac{V_d}{CR - 1} = \frac{500}{9.5 - 1} = 58.82 \approx 35\ cm^3$$

2. Bore diameter and stroke: the engine is square, so: D=L

$$V_d = \frac{\pi . D^2}{4}.D \Rightarrow D = \left(\frac{4 * V_d}{\pi}\right)^{1/3} = \left(\frac{4 * 500}{\pi}\right)^{1/3} = 8.6\ cm^3$$

3. Average piston speed

L=0.086m

N=3750 rpm

$$C_m = \frac{2 \times S \times N}{60} = \frac{2 \times 0.086 \times 3750}{60} = 10.75\ m/s$$

Average piston speed = 10.75 m/s

Chapter II: Thermodynamic of engine cycles

2.1. Introduction

Most power producing devices operate on cycles, and the study of power cycles is an exciting and important part of thermodynamics. The cycles encountered in actual devices are difficult to analyze because of the presence of complicating effects, such as friction, and the absence of sufficient time for establishment of the equilibrium conditions during the cycle. To make an analytical study of a cycle feasible, we have to keep the complexities at a manageable level and utilize some idealizations. When the actual cycle is stripped of all the internal irreversibilities and complexities, we end up with a cycle that resembles the actual cycle closely but is made up totally of internally reversible processes. Such a cycle is called an ideal cycle.

The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the air- standard assumptions:

1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
2. All the processes that make up the cycle are internally reversible.
3. The combustion process is replaced by a heat-addition process from an external source.
4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

Another assumption that is often utilized to simplify the analysis even more is that air has constant specific heats whose values are determined at room temperature (25°C). When this assumption is utilized, the air-standard assumptions are called the cold-air-standard assumptions. A cycle for which the air-standard assumptions are applicable is frequently referred to as an air-standard cycle.

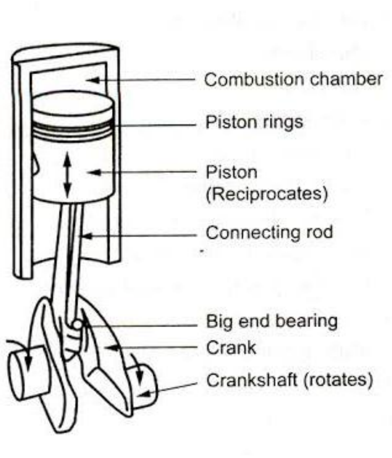
2.2. Overview of Reciprocating Engines

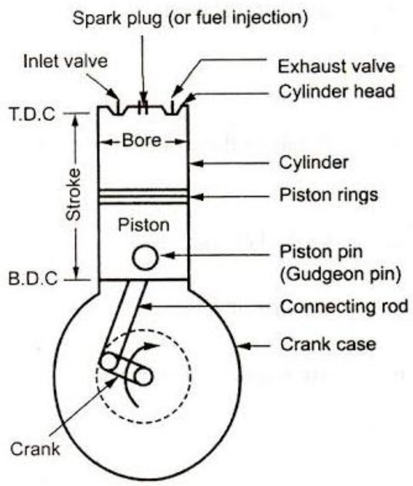
Despite its simplicity, the reciprocating engine (basically a piston-cylinder device) is one of the rare inventions that have proved to be very versatile and to have a wide range of applications. It is the powerhouse of the vast majority of automobiles, trucks, light aircrafts, ships and electric power generators, as well as many other devices.

The basic components of a reciprocating engine are shown in figure below. The piston reciprocates in the cylinder between two fixed positions called the top dead center (TDC) ‘the position of the piston when it forms the smallest volume in the cylinder’ and the bottom dead center (BDC) ‘the position of the piston when it forms the largest volume in the cylinder’. The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction, and it is called the stroke of the engine. The diameter of the piston is called the bore. The air or air-fuel mixture is drawn into the cylinder through the intake valve, and the combustion products are expelled from the cylinder through the exhaust valve.

The minimum volume formed in the cylinder when the piston is at TDC is called the clearance volume. The volume displaced by the piston as it moves between TDC and BDC is called the displacement volume. Figure below shoes both the clearance and displacement volumes. The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is called the compression ratio *CR* of the engine:

$$CR = r_v = \varepsilon = \frac{V_T}{V_c} = \frac{V_d}{V_c} + 1 = \frac{V_{TDC}}{V} \qquad (2.1)$$





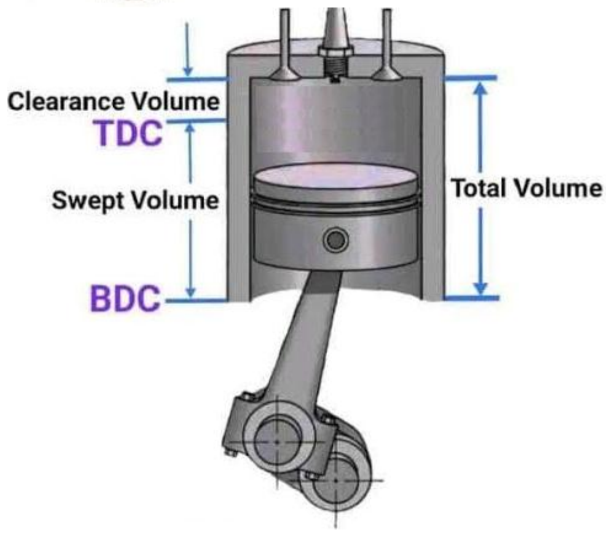


Figure 2.1. *Internal components and basic terminology of a reciprocating ICE*

Another term frequently used in conjunction with reciprocating engines is the mean effective pressure (MEP). It is a fictitious pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle:

$$\boldsymbol{MEP} = \frac{W_{net}}{V_{max}-V_{min}} = \frac{W_{net}}{V_1-V_2} \tag{2.2}$$

The mean effective pressure can be used as a parameter to compare the performances of reciprocating engines of equal size. The engine with a larger value of MEP delivers more net work per cycle and thus performs better.

Reciprocating engines are classified as spark- ignition (SI) engines or compression-ignition (CI) engines, depending on how the combustion process in the cylinder is initiated. In SI engines, the combustion of the air-fuel mixture is initiated by a spark plug. In CI engines, the air-fuel mixture is self-ignited as a result of compressing the mixture above its self-ignition temperature. Otto cycle and Diesel cycle are the ideal cycles for the SI and CI reciprocating engines, respectively.

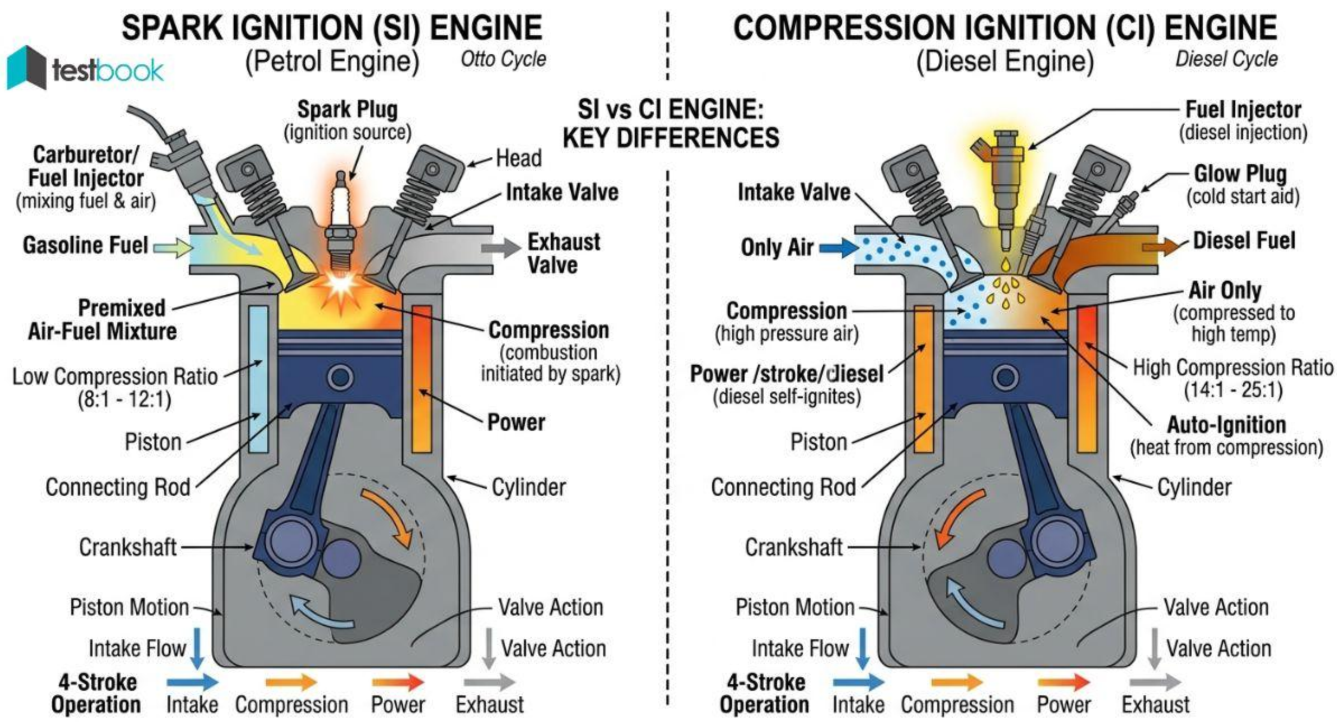


Figure 2.2. *Spark ignition engine and compression ignition engine*

2.3. Engine cycles

2.3.1. Thermodynamic principals

The First Law of Thermodynamics is essentially the law of conservation of energy. It states that energy cannot be created or destroyed, only transformed from one form to another.
In a closed system, The change in the system's internal energy (ΔU)is expressed with this formula:

$$\Delta U = \delta W + \delta Q \tag{2.3}$$

With δW : The work done by the system on its surroundings ; $\delta W_{i-f} = -\int_i^f PdV$

δQ: The heat added to the system ;

$$\delta Q = \begin{cases} m.C_v.dT + PdV \text{ (for constant volume transformation)} \\ m.C_p.dT - VdP \text{ (for constant pressure transformation)} \end{cases}$$

$$P.V = n.R.T\ ; \quad P.V = m.r.T \tag{2.4}$$

Where :

P = Gas pressure (*Pa*) ; V = Gas volume (m^3) ; n = Number of mole (*mole*);
R = Gas constant (*J/mole.K*); T= Gas temperature (*K*) ; r = Massic gas constant (*J/kg.K*)
m = Mass of gas in cylinder (*kg*) ; C_p & C_v = Specific heat at constant pressure and constant volume

2.3.1.1 Isentropic process ($\boldsymbol{\delta Q = 0}$; $\boldsymbol{P.V^{\gamma} = C^{te}}$)

$$P_1.V_1{}^{\gamma} = P_2.V_2{}^{\gamma} \Rightarrow P_2 = P_1.\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1.(r_v)^{\gamma} \tag{2.5}$$

$$T_1.V_1{}^{\gamma-1} = T_2.V_2{}^{\gamma-1} \Rightarrow T_2 = T_1.\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1.(r_v)^{\gamma-1} \tag{2.6}$$

2.3.1.2 Constant volume process (dV=0)

$$\begin{cases} P_2V_2 = n.R.T_2 \\ P_3V_3 = n.R.T_3 \end{cases} \Rightarrow \frac{P_3}{P_2} = \frac{T_3}{T_2} \tag{2.7}$$

2.3.1.3 Constant pressure process (dP=0)

$$\begin{cases} P_2V_2 = n.R.T_2 \\ P_3V_3 = n.R.T_3 \end{cases} \Rightarrow \frac{V_3}{V_2} = \frac{T_3}{T_2} \tag{2.8}$$

2.3.2. Spark Ignition Engines and Otto Cycle

In most spark-ignition engines, the piston executes four complete strokes (two mechanical cycles) within the cylinder, and the crankshaft completes two revolutions for each thermodynamic cycle. These engines are called four-stroke internal combustion engines. In two-stroke engines, all four functions described above are executed in just two strokes: the power stroke and the compression stroke. The thermodynamic analysis of the actual four- stroke or two-stroke cycles is not a simple task. However, the analysis can be simplified significantly if the air-standard assumptions are utilized. The resulting cycle, which closely resembles the actual operating conditions, is the ideal Otto cycle. It consists of four internally reversible processes as shown on the (*P-V*) diagram:

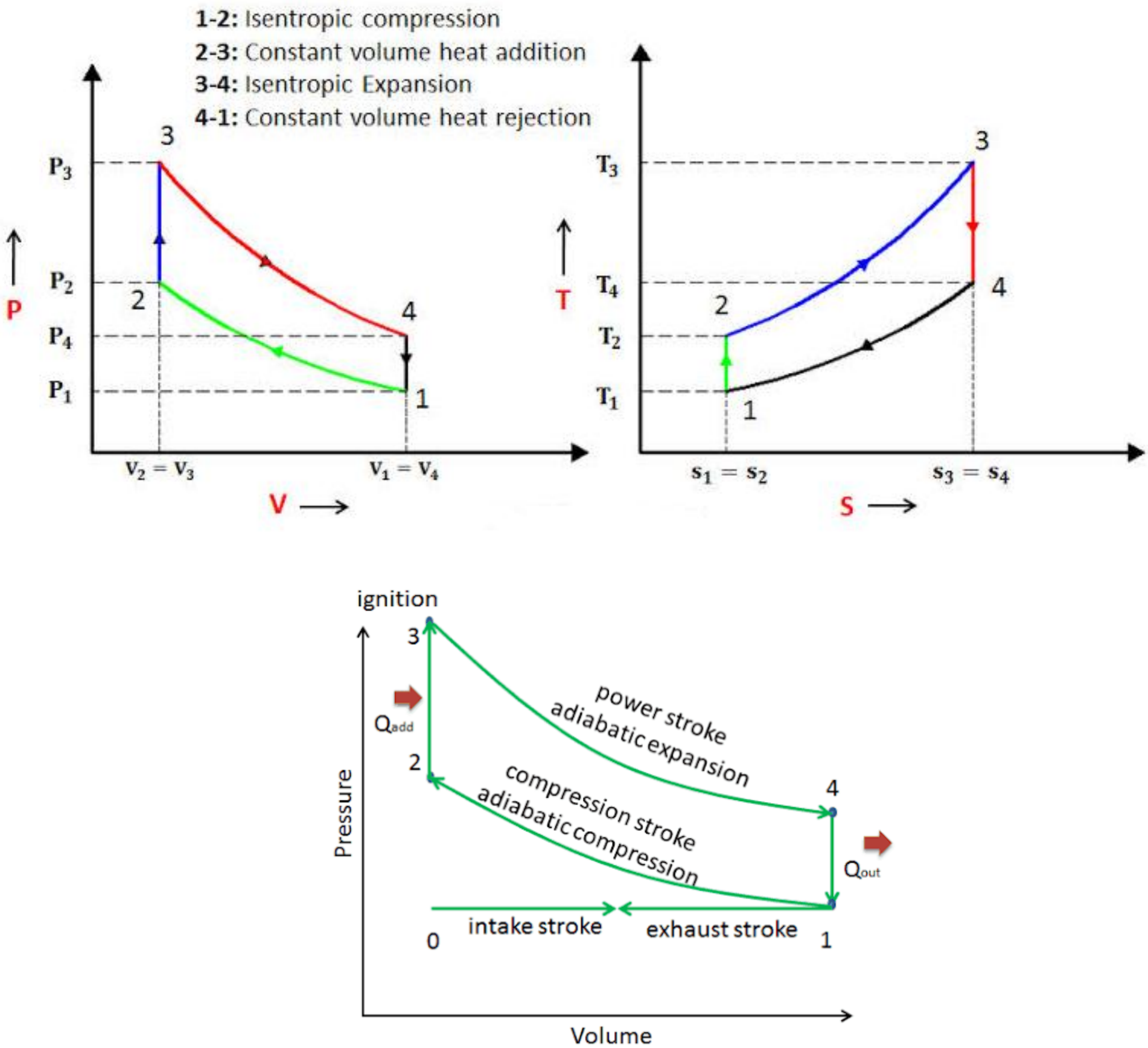


Figure 2.3. *P-V and T-S diagram of Otto cycle (Spark ignition engine cycle)*

The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit mass basis, as:

$$(Q_{in} - Q_{out}) + (W_{in} - W_{out}) = \Delta U \tag{2.9}$$

No work is involved during the two heat transfer processes since both take place at constant volume. Therefore, heat transfer to and from the working fluid can be expressed as:

$$Q_{in} = U_3 - U_2 = C_v(T_3 - T_2) \tag{2.10}$$

$$Q_{out} = U_4 - U_1 = C_v(T_4 - T_1) \tag{2.11}$$

The net work of the cycle is:

$$W_{net} = W_{out} - W_{in} = Q_{in} - Q_{out} \tag{2.12}$$

2.3.2.1. The thermal efficiency

The the thermal efficiency of the ideal Otto cycle under cold-air-standard assumptions is expressed as:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{|m.C_v.(T_1-T_4)|}{|m.C_v.(T_3-T_2)|} = 1 - \frac{m.C_v.(T_4-T_1)}{m.C_v.(T_3-T_2)} \tag{2.13}$$

$$\eta_{th} = 1 - \frac{T_1(\frac{T_4}{T_1} - 1)}{T_2(\frac{T_3}{T_2} - 1)} ; \begin{cases} T_1.V_1{}^{\gamma-1} = T_2.V_2{}^{\gamma-1} \\ T_4.V_4{}^{\gamma-1} = T_3.V_3{}^{\gamma-1} \end{cases} \Rightarrow \frac{T_4}{T_1} = \frac{T_3}{T_2} \text{ witch give } \frac{T_4}{T_1} - 1 = \frac{T_3}{T_2} - 1$$

Processes 1-2 and 3-4 are isentropic, and $V_2 = V_3$ and $V_4 = V_1$, $\boldsymbol{CR} = \frac{\boldsymbol{V_1}}{\boldsymbol{V_2}}$ thus:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{1}{CR}\right)^{\gamma-1} = 1 - CR^{1-\gamma} \tag{2.14}$$

$$T_1.V_1{}^{\gamma-1} = T_2.V_2{}^{\gamma-1} \Rightarrow \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{1}{CR}\right)^{\gamma-1} = CR^{1-\gamma}$$

2.3.2.2. The mean effective pressure

The mean effective pressure MEP is given by:

$$MEP = \frac{W_{net}}{V_c} = \frac{Q_{in} - Q_{out}}{V_1 - V_2} = \frac{m.C_v.(T_3-T_2) - m.C_v.(T_4-T_1)}{V_1(1-\frac{V_2}{V_1})} = \frac{m.C_v.(T_3-T_2) - m.C_v.(T_4-T_1)}{\frac{m.r.T_1}{P_1}(1-\frac{1}{CR})} \tag{2.15}$$

$$\frac{C_v}{r} = \frac{1}{\gamma-1} \quad ; \quad CR = r_v = \frac{V_1}{V_2} = \frac{V_4}{V_3} \quad ; \quad r_c = \frac{T_3}{T_2} = \frac{P_3}{P_2} \quad ; \quad \frac{T_4}{T_3} = \frac{T_1}{T_2}$$

$$MEP = P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{\left(\frac{T_3}{T_1}-\frac{T_2}{T_1}\right)-\left(\frac{T_4}{T_1}-1\right)}{CR-1}\right] = P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{\left(\frac{T_3}{T_2}.\frac{T_2}{T_1}-CR^{\gamma-1}\right)-\left(\frac{T_4}{T_3}.\frac{T_3}{T_2}.\frac{T_2}{T_1}-1\right)}{CR-1}\right]$$

$$= P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{(r_c.CR^{\gamma-1}-CR^{\gamma-1})-(CR^{-\gamma+1}.r_c.CR^{\gamma-1}-1)}{CR-1}\right]$$

$$= P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{CR^{\gamma-1}.(r_c-1)-(r_c-1)}{CR-1}\right]$$

$$MEP = P_1.\left(\frac{1}{\gamma-1}\right).(r_c-1).\left[\frac{CR^{\gamma}-CR}{CR-1}\right] \qquad (2.16)$$

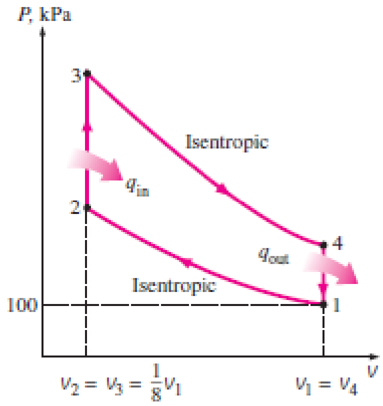
Example 2.1

The compression ratio in an air-standard Otto cycle is 8. At the beginning of the compression stroke, the pressure is 0.1 MPa and the temperature is 15°C. The heat transferred to the air per cycle is 1800 kJ/kg. Draw the (P-V) diagram of the cycle and determine:

1) The pressure and temperature at the end of each process of the cycle.

2) The thermal efficiency of the cycle.

3) The mean effective pressure.



Solution:

1) The compression ratio $CR = \frac{V_1}{V_2} = \frac{V_4}{V_3} = 8$

Process 1-2 (isentropic compression):

$$\boldsymbol{T_1.{V_1}^{\gamma-1} = T_2.{V_2}^{\gamma-1} \Rightarrow T_2 = T_1.\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1.(r_v)^{\gamma-1}}$$

$$\Rightarrow T_2 = (15+273).(8)^{1.4-1}$$

$\boldsymbol{T_2 = 662}$ **K**

$$P_1.{V_1}^{\gamma} = P_2.{V_2}^{\gamma} \Rightarrow P_2 = P_1.\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1.(r_v)^{\gamma} = 0.1\times10^3\times8^{1.4} = 1837.9\ kPa$$

$\boldsymbol{P_2 = 1837.9}$ **kPa**

Process 2-3 (constant-volume heat addition):

$Q_{in} = (T_3 - T_2) \Rightarrow 1800 = 0.718\times(T_3 - 662)$

$\boldsymbol{T_3 = 3169}$ **K**

$$\frac{P_3}{P_2} = \frac{T_3}{T_2} \rightarrow P_3 = 1837.9\times\frac{3169}{662}$$

$\boldsymbol{P_3}$ = **8798 kPa**

Process 3-4 (isentropic expansion):

$$\Rightarrow \frac{\boldsymbol{T_4}}{\boldsymbol{T_3}} = \left(\frac{\boldsymbol{V_3}}{\boldsymbol{V_4}}\right)^{\boldsymbol{\gamma-1}} = \boldsymbol{CR^{1-\gamma}} \ ; \ \mathbf{T_4} = (\mathbf{3169}) \times (\mathbf{8})^{\mathbf{1-1.4}}$$

$\boldsymbol{T}_4 = \mathbf{1380\ K}$

$$P_3.V_3{}^{\gamma} = P_4.V_4{}^{\gamma} \Rightarrow P_4 = P_3.\left(\frac{V_3}{V_4}\right)^{\gamma} = P_3.(r_v)^{-\gamma} = 8798 \times 8^{-1.4} = 479\ kPa$$

$\boldsymbol{P}_4 = \mathbf{479\ kPa}$

2) The thermal efficiency of the cycle:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - CR^{1-\gamma} = 1 - 8^{-0.4} = 0.5647 = \mathbf{56.47}\%$$

3) From the equation of state:

$$P_1.V_1 = R.T_1 \Rightarrow V_1 = \frac{R.T_1}{P_1} = \frac{0.287 \times 288}{0.1 \times 10^3} = 0.827\ m^3/kg$$

$$CR = \frac{V_1}{V_2} \Rightarrow V_2 = \frac{V_1}{CR} = \frac{0.827}{8} = 0.1034\ m^3/kg$$

$$Q_{out} = C_v(T_4 - T_1) = 0.718 \times (1380 - 288) = 784 \ \ kJ/kg$$

$$W_{net} = Q_{in} - Q_{out} = 1800 - 784 = 1016 \ \ kJ/kg$$

$$MEP = \frac{W_{net}}{V_1 - V_2} = \frac{1016}{0.827 - 0.1034} = \boldsymbol{1404\ kP}$$

2.3.3. Compression Ignition Engines and Diesel Cycle

The Diesel cycle is the ideal cycle for compression ignition (CI) reciprocating engines. The CI engine is very similar to the SI engine, differing mainly in the method of initiating combustion. In spark- ignition engines (also known as gasoline engines), the air–fuel mixture is compressed to a temperature that is below the auto-ignition temperature of the fuel, and the combustion process is initiated by firing a spark plug. In CI engines (also known as diesel engines), the air is compressed to a temperature that is above the auto-ignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines.

The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. Therefore, the combustion process in these engines takes place over a longer interval. Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat- addition process. In fact, this is the only process where the Otto and the Diesel cycles differ. The remaining three processes are the same for both ideal cycles. The four processes comprising the cycle are shown on the (P-V) diagram:

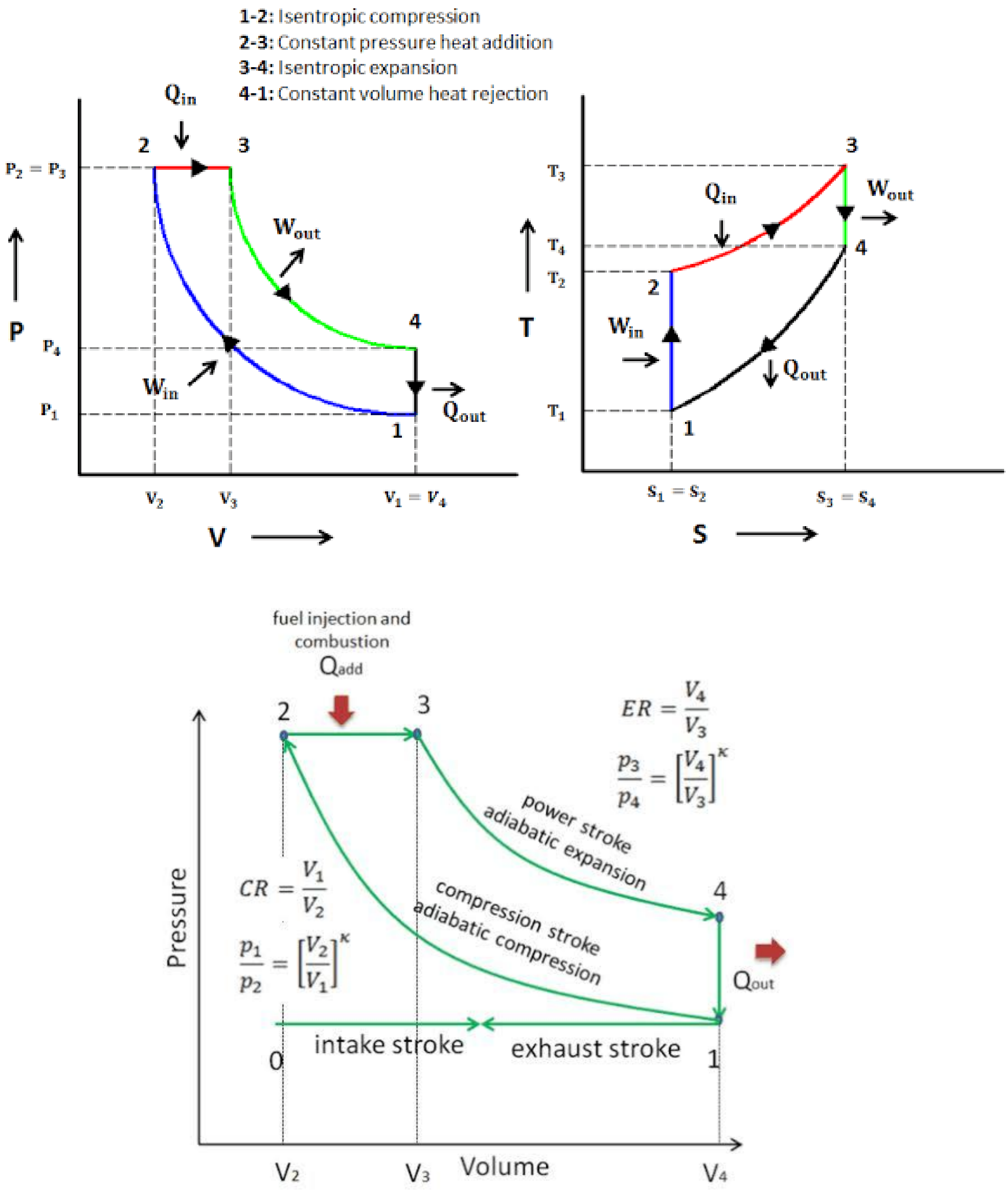


Figure 2.4. *P-V and T-S diagram of Diesel engine cycle*

Noting that the Diesel cycle is executed in a piston-cylinder device, which forms a closed system, the amount of heat transferred to the working fluid at constant pressure and rejected from it at constant volume can be expressed as:

$$Q_{in} = h_3 - h_2 = C_p(T_3 - T_2) \quad (2.17)$$

$$Q_{out} = U_4 - U_1 = C_v(T_4 - T_1)$$
(2.18)

The net work of the cycle is:

$$W_{net} = W_{out} - W_{in} = Q_{in} - Q_{out} \quad (2.19)$$

2.3.3.1. The thermal efficiency

Then the thermal efficiency of the ideal Diesel cycle under the cold-air-standard assumptions is expressed as:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{|m.C_v.(T_1 - T_4)|}{|m.C_p.(T_3 - T_2)|} = 1 - \frac{m.C_v.(T_4 - T_1)}{m.C_p.(T_3 - T_2)} \quad (2.20)$$

$$\eta_{th} = 1 - \frac{(\frac{T_4}{T_2} - \frac{T_1}{T_2})}{\gamma(\frac{T_3}{T_2} - 1)} = 1 - \frac{(\frac{T_4}{T_3}\frac{T_3}{T_2} - \frac{T_1}{T_2})}{\gamma(\frac{T_3}{T_2} - 1)};$$

Processes 1-2 and 3-4 are isentropic, $V_4 = V_1$, $\boldsymbol{r_c = \frac{T_3}{T_2} = \frac{V_3}{V_2}}$ ***and*** $\boldsymbol{CR = \frac{V_1}{V_2}}$ thus:

$$T_1.V_1^{\gamma-1} = T_2.V_2^{\gamma-1} \Rightarrow \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{1}{CR}\right)^{\gamma-1} = CR^{1-\gamma}$$

$$T_3.V_3^{\gamma-1} = T_4.V_4^{\gamma-1} \Rightarrow \frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{\frac{V_3}{V_2}}{\frac{V_4}{V_2}}\right)^{\gamma-1} = = \left(\frac{r_c}{CR}\right)^{\gamma-1}$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{\left[\left(\frac{r_c}{CR}\right)^{\gamma-1}.r_c - (CR)^{1-\gamma}\right]}{\gamma(r_c - 1)} = 1 - (CR)^{1-\gamma}.\frac{[r_c^{\gamma} - 1]}{\gamma(r_c - 1)} \quad (2.21)$$

Where *CR* is the compression ratio defined by equation (2.1), r_c is the combustion ration. Looking at equation (2.21) carefully, one would notice that under the cold-air-standard assumptions, the efficiency of a Diesel cycle differs from the efficiency of an Otto cycle by the quantity in the brackets. Therefore:

$$\eta_{th,otto} > \eta_{th,Diesel} \quad (2.22)$$

2.3.3.2. The mean effective pressure

The mean effective pressure MEP is given by:

$$MEP = \frac{W_{net}}{V_c} = \frac{Q_{in} - Q_{out}}{V_1 - V_2} = \frac{m.C_p.(T_3 - T_2) - m.C_v.(T_4 - T_1)}{V_1\left(1 - \frac{V_2}{V_1}\right)}$$

$$= \frac{m.C_p.(T_3 - T_2) - m.C_v.(T_4 - T_1)}{\frac{m.r.T_1}{P_1}(1 - \frac{1}{CR})} \tag{2.23}$$

$$\begin{cases} \gamma = \frac{C_P}{C_v} \\ r = C_P - C_v \end{cases} \Rightarrow \frac{C_v}{r} = \frac{1}{\gamma - 1} \quad ; \quad CR = r_v = \frac{V_1}{V_2}; \quad r_c = \frac{T_3}{T_2} = \frac{V_3}{V_2} \quad ;$$

$$\boldsymbol{MEP = P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{\left(\frac{T_3}{T_1} - \frac{T_2}{T_1}\right).\gamma - \left(\frac{T_4}{T_1} - 1\right)}{CR-1}\right]}$$

$$\boldsymbol{MEP = P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{\gamma.CR^{\gamma-1}(r_c - 1) - (r_c{}^{\gamma} - 1)}{CR-1}\right]} \tag{2.24}$$

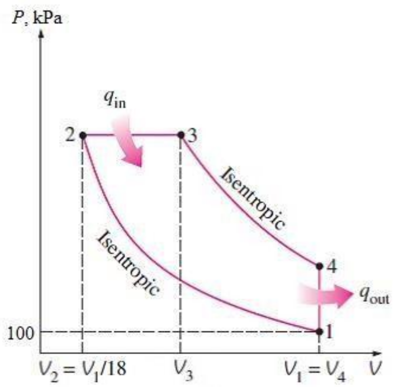
Example (2.2):

An air-standard Diesel cycle has a compression ratio of 18, and the heat transferred to the working fluid per cycle is 1800 kJ/kg. At the beginning of the compression process, the pressure is 0.1 MPa and the temperature is 15°C. Draw the (P-V) diagram of the cycle and determine:

1) The pressure and temperature at the end of each process of the cycle.

2) The thermal efficiency of the cycle.

3) The mean effective pressure.



Solution:

1) The compression ratio CR = 18

$$P_1.V_1 = r.T_1 \Rightarrow V_1 = \frac{r.T_1}{P_1} = \frac{0.287 \times 288}{0.1 \times 10^3} = 0.827\ m^3/kg$$

$V_1 = V_4 = 0.827\ m^3/kg$

$$CR = \frac{V_1}{V_2} \Rightarrow V_2 = \frac{V_1}{CR} = \frac{0.827}{18} = 0.0459\ m^3/kg$$

Process 1-2 (isentropic compression):

$$\boldsymbol{T_1.V_1}^{\gamma-1} = \boldsymbol{T_2.V_2}^{\gamma-1} \Rightarrow \boldsymbol{T_2} = \boldsymbol{T_1}.\left(\frac{\boldsymbol{V_1}}{\boldsymbol{V_2}}\right)^{\gamma-1} = \boldsymbol{T_1}.(\boldsymbol{r_v})^{\gamma-1}$$

$\Rightarrow T_2 = (15 + 273.15).(18)^{1.4-1}$

$\boldsymbol{T_2} = \mathbf{915.64}$ **K**

$$P_1.V_1{}^{\gamma} = P_2.V_2{}^{\gamma} \Rightarrow P_2 = P_1.\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1.(r_v)^{\gamma} = 0.1 \times 10^3 \times 18^{1.4} = \mathbf{5720}\ \boldsymbol{kPa}$$

Process 2-3 (constant-pressure heat addition):

$\boldsymbol{P_3} = \boldsymbol{P_2} = \mathbf{5720}$ **kPa**

$Q_{in} = C_p(T_3 - T_2) \Rightarrow 1800 = 1.005 \times (T_3 - 915.62)$

$\boldsymbol{T_3} = \mathbf{2706.68}$ **K**

$$\frac{V_3}{V_2} = \frac{T_3}{T_2} \Rightarrow V_3 = V_2.\frac{T_3}{T_2} = 0.0459.\frac{2706.68}{915.64} = 0.1357\ m^3/kg$$

Process 3-4 (isentropic expansion):

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_2}\right)^{\gamma-1} = \mathrm{CR}^{1-\gamma} \Rightarrow;\ T_4 = (2706.68) \times (0.827/0.1357)^{1-1.4}$$

$\boldsymbol{T_4} = \mathbf{1313.61}$ **K**

$$P_3.V_3{}^{\gamma} = P_4.V_4{}^{\gamma} \Rightarrow P_4 = P_3.\left(\frac{V_3}{V_4}\right)^{\gamma} = P_3.(r_v)^{-\gamma} = 5720 \times \left(\frac{0.827}{0.1357}\right)^{-1.4} = 456\ kPa$$

$\boldsymbol{P_4} = \mathbf{456}$ **kPa**

2) $Q_{out} = C_v(T_4 - T_1) = 0.718 \times (1313.61 - 288.15) = 736.28$ kJ/kg

$W_{net} = Q_{in} - Q_{out} = 1800 - 736.28 = 1063.72$ kJ/kg

The thermal efficiency of the cycle:

$$\boldsymbol{\eta_{th}} = \frac{\boldsymbol{W_{net}}}{\boldsymbol{Q_{in}}} = \frac{\mathbf{1063.72}}{\mathbf{1800}} = \mathbf{0.5909} = \mathbf{59\%}$$

4) The mean effective pressure:

$$\boldsymbol{MEP} = \frac{\boldsymbol{W_{net}}}{\boldsymbol{V_c}} = \frac{\boldsymbol{W_{net}}}{\boldsymbol{V_1} - \boldsymbol{V_2}} = \frac{\mathbf{1063.72}}{\mathbf{0.827 - 0.0459}} = \mathbf{1361.82}\ \boldsymbol{kPa}$$

2.3.4. Dual Cycle

Approximating the combustion process in internal combustion engines as a constant-volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic. Probably a better (but slightly more complex) approach would be to model the combustion process in both gasoline and diesel engines as a combination of two heat-transfer processes, one at constant volume and the other at constant pressure. The ideal cycle based on this concept is called the dual cycle, and a (*P-V*) diagram for it is shown below. The relative amounts of heat transferred during each process can be adjusted to approximate the actual cycle more closely. Note that both the Otto and the Diesel cycles can be obtained as special cases of the dual cycle.

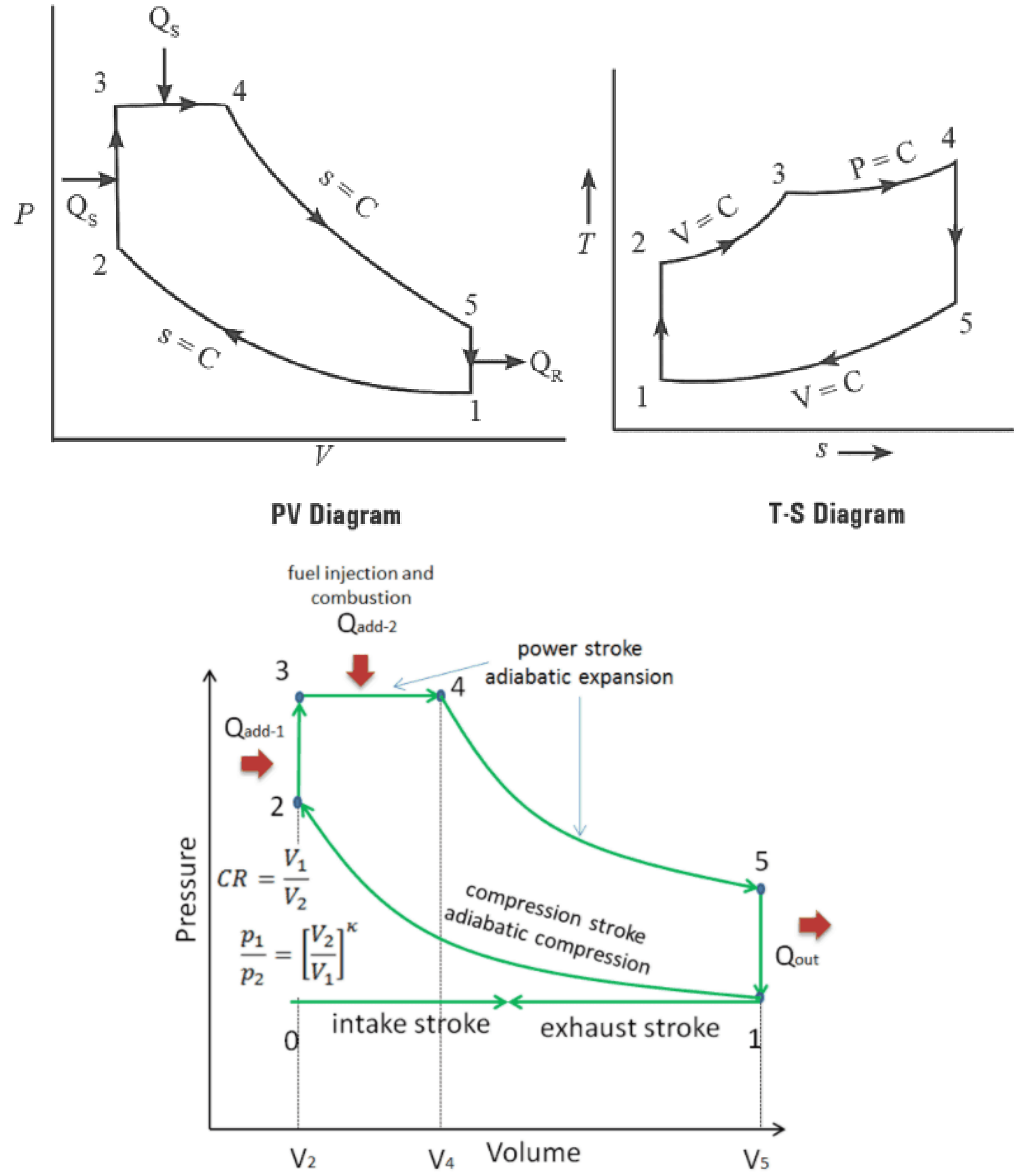


Figure 2.5. *P-V and T-S diagram of Dual (Sabathé) engine cycle*

2.3.3.1. The thermal efficiency

Then the thermal efficiency of the ideal Dual cycle under the cold-air-standard assumptions is expressed as:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{|m.C_v.(T_1 - T_5)|}{|m.C_v.(T_3 - T_2) + m.C_p.(T_4 - T_3)|} = 1 - \frac{C_v.(T_5 - T_1)}{C_v.(T_3 - T_2) + C_p.(T_4 - T_3)} \quad (2.25)$$

2.4. Exercises

Exercise 01

A mixture of fuel and air has a combustion enthalpy Hc= 2560kJ/kg. To simulate a spark-ignition internal combustion engine (gasoline), let us assume a Beau de Rochas cycle (Otto-cycle) with Qin = 2560 kJ/kg. The compression ration CR = 9 . At the beginning of compression: pressure P_1 = 100 kPa , temperature T_1 = 20 °C , Cv = 0.7167 kJ/kg.K

1. Determine the pressure and the temperature maximum of the cycle.
2. Determine the thermal efficiency of the cycle and the amount of rejected heat.
3. Determine the mean effective pressure.

Exercise 02 :

A heat engine using a perfect diatomic fluid describes the reversible gasoline (ABCDA) cycle. composed of two isochores connected by two adiabatic processes . The gas is admitted at a pressure P_A = 1 bar into the cylinder of an engine with a volume V_A = 0.1 liter at a temperature of 300 K. It undergoes the following reversible transformations:

- A----B: Adiabatic compression up to volume V_B = 0.01 liters of the CC;
- B----C: Heating at constant volume until a temperature T_C = 2120K is reached;
- C-----D: Adiabatic expansion back to the initial volume;
- D-----A: Cooling of the gas at constant volume to the initial pressure.

1. Provide a schematic representation of this cycle in a (P-V) diagram.
2. Complete table 1; an explanation of the results is required.

State	P(bar)	V (liter)	T(K)
A			
B			
C			
D			

Exercise 03 :

A diesel engine in which the combustion of fuel (diesel) is replaced by the isobaric heating of a gas whose, the composition remains constant during the cycle. A diatomic ideal gas ($\gamma = 1.4$) is admitted at a pressure of 1 bar into the cylinder of engine with volume $V_A = 2$ liter at a temperature of 293 K. It undergoes the following reversible transformations:

- A-----B: Adiabatic compression up to volume $V_B = 0.1$ liter of the chamber combustion;
- B-----C: Heating at constant pressure up to volume $V_C = 0.3$ liter;
- C-----D: Adiabatic expansion back to the initial volume;
- D-----A: Cooling of the gas at constant volume to the initial pressure.

1. Provide a schematic representation of this cycle in a (P-V) diagram.
2. Complete table 1; an explanation of the results is required.

State	P(bar)	V (liter)	T(K)
A			
B			
C			
D			

Exercise 04 :

A diesel engine has a compression ratio of 20. The pressure and temperature at the beginning of the compression stroke are 95 kPa and 25°C respectively. The maximum temperature reached is 2800 K.

1. Calculate the thermal efficiency of the cycle and the mean effective pressure.
2. What compression ratio can a spark-ignition engine have in order to achieve the same efficiency as this diesel cycle?

Exercise 05:

In a compression-ignition engine operating on a reversible dual cycle (Sabathé cycle). The fuel is injected at point 2. The combustion begins (starts) from point 2 to point 4. Transformations 1-2 and 4-5 are isentropic, while transformation 2-3 is isochoric and transformation 3-4 is isobaric. At point 1, the pressure is 1 bar and the temperature is 25 °C. At point 4, the pressure is 70 bar and the temperature is 2100 K. The volumetric compression ratio is CR = 19. Both compression and expansion are adiabatic. $\gamma = 1.4$ and $r = 287$ J/ kg·K .

1. Provide a representation of this cycle in a (P,V) diagram and then in a (T,S) diagram.
2. Calculate the temperatures and pressures in the C.C. at points 2, 3, and 5 of the cycle.
3. Calculate the heat exchanged per unit mass.
4. Calculate the thermal efficiency of the cycle.

Exercise 06:

An engine, operating on a reversible double combustion cycle (1-2-3-4-5), receives air at 1 bar and 20°C. The maximum cycle pressure is 70 bar and the maximum temperature is 2000°C. The volumetric compression ratio CR is 20. Compression and expansion are adiabatic. γ=1.4; R= 287 J/ kg.K .

1. Represent this cycle in in a (P,V) diagram and then in a (T,S) diagram.
2. Determine the temperatures and pressures at each point in the cycle.
3. Determine the quantities of heat and work exchanged for each transformation.
4. Determine the thermal efficiency of this cycle.

Exercise 7:

An internal combustion engine has a compression ratio of = 10.1 and uses gasoline as fuel. At an engine speed of 1500 rpm, the combustion of the air-fuel mixture releases a heat Q_{in} equal to 4300 kJ/kg, under intake conditions of 0.89 bar and 25 °C. Given, C_v = 1.62 kJ/(kg·K),

If the engine operates according to the Otto cycle, we are asked to:

1. Provide a schematic representation of the engine cycle using the P-V and T-S diagrams.
2. Calculate the pressure and temperature at each peak of the engine cycle.
3. Calculate the work done by the engine (W_{net}), and the amount of heat released rejected (Q_f)

In the same engine, it was experimentally verified that the expansion of the burnt gases follows a polytropic evolution with an exponent η of 1.28.

Determine:

4. The pressure and temperature at the end of the expansion.
5. The net work of the cycle.

2.5. Solution of exercises

Solution of exercise 01 :

1.

$$Q_{in} = C_v(T_3 - T_2) \Rightarrow T_3 = \frac{Q_{in}}{C_v} + T_2 \qquad \text{(a)}$$

Process 1-2 (isentropic compression):

$$\boldsymbol{T_1 . V_1{}^{\gamma-1} = T_2 . V_2{}^{\gamma-1} \Rightarrow T_2 = T_1 . \left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1 . (r_v)^{\gamma-1}}$$

$\Rightarrow T_2 = (20 + 273.15).(9)^{1.4-1}$

$\boldsymbol{T}_2 = 705.97$ **K**

From equation (a) : $T_3 = \frac{Q_{in}}{C_v} + T_2 = \frac{2560}{0.7167} + 705.97 = \boldsymbol{4277.89688\ K}$

Process 2-3 (constant-volume heat addition):

$V_3 = V_2 \Rightarrow \frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow P_3 = P_2 \times \frac{T_3}{T_2}$; $\quad P_1 . V_1{}^{\gamma} = P_2 . V_2{}^{\gamma} \Rightarrow P_2 = P_1 . \left(\frac{V_1}{V_2}\right)^{\gamma}$

$$P_3 = P_1 . \left(\frac{V_1}{V_2}\right)^{\gamma} \times \frac{T_3}{T_2} = P_1 . (CR)^{\gamma} \times \frac{T_3}{T_2} = 1 \times (9)^{1.4} \times \frac{4277.89}{705.97} = \boldsymbol{131.33\ bar}$$

2. The thermal efficiency of the cycle:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} == 1 - CR^{1-\gamma} = 1 - 9^{-0.4} = 0.5847 = \boldsymbol{58.47}\%$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} \Rightarrow Q_{out} = (1 - \eta_{th}) \times Q_{in}$$

$$Q_{out} = (1 - 0.5847) \times 2560 = \boldsymbol{1063.168\ kJ/kg}$$

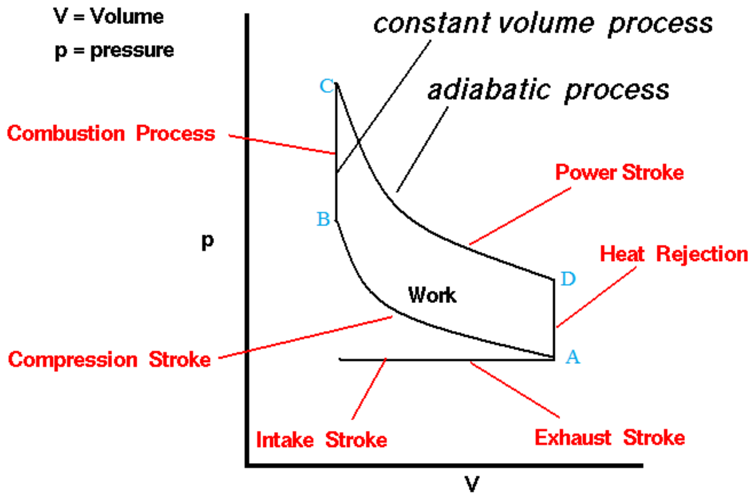
3. Mean effective pressure

$$MEP = P_1 . \left(\frac{1}{\gamma - 1}\right) . (r_c - 1) . \left[\frac{CR^{\gamma} - CR}{CR - 1}\right] = 1 \times \left(\frac{1}{1.4 - 1}\right) . \left(\frac{4278.89}{705.97} - 1\right) . \left[\frac{9^{1.4} - 9}{9 - 1}\right]$$

$MEP = \boldsymbol{20.04\ bar}$

Solution of exercise 02 :

1. Diagram (P-V)



2

State	P(bar)	V (liter)	T(K)
A	1	0.1	300
B	25.118	0.01	753.566
C	70.664	0.01	2120
D	2.813	0.1	843.987

$$T_A.V_A{}^{\gamma-1} = T_B.V_B{}^{\gamma-1} \Rightarrow T_B = T_A.\left(\frac{V_A}{V_B}\right)^{\gamma-1} \Longrightarrow T_B = (300).\left(\frac{0.1}{0.01}\right)^{1.4-1} = \mathbf{753.566}\ \mathrm{K}$$

$$P_A.V_A{}^{\gamma} = P_B.V_B{}^{\gamma} \Rightarrow P_B = P_A.\left(\frac{V_A}{V_A}\right)^{\gamma} = P_A.(r_v)^{\gamma} = 1 \times \left(\frac{0.1}{0.01}\right)^{1.4} = \mathbf{25.118\ bar}$$

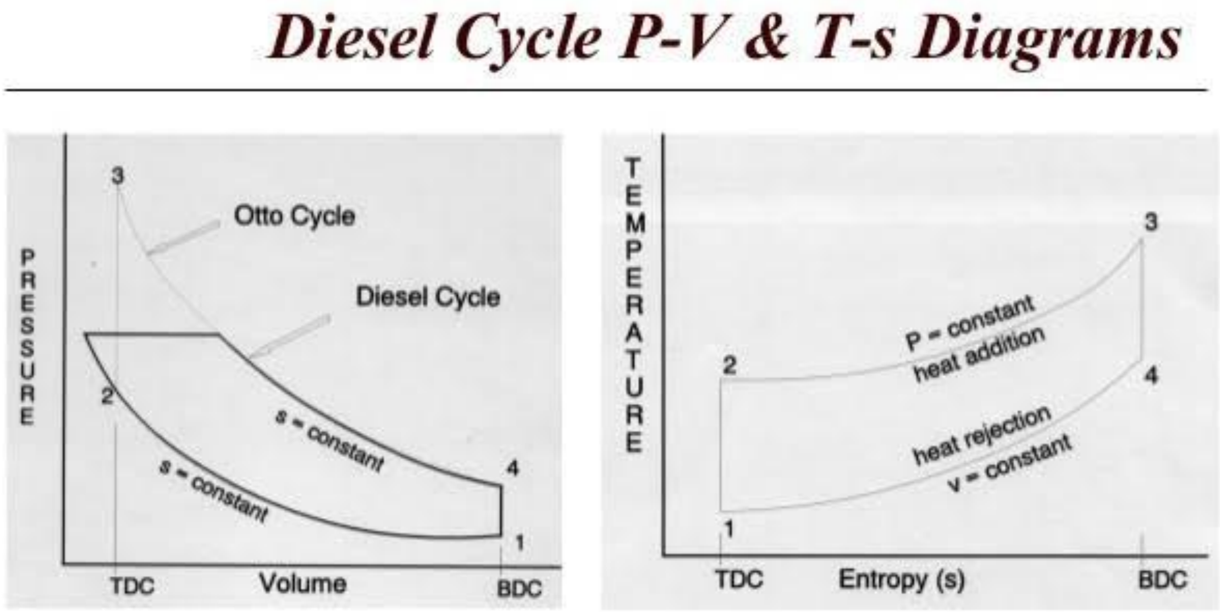
$$\frac{P_C}{P_B} = \frac{T_C}{T_B}\ (V_C = V_B) \Rightarrow\ P_C = P_B \times \frac{T_C}{T_B} = 25.118 \times \frac{2120}{753.566} = \mathbf{70.664\ bar}$$

$$T_c.V_c{}^{\gamma-1} = T_D.V_D{}^{\gamma-1} \Rightarrow T_D = T_C.\left(\frac{V_C}{V_D}\right)^{\gamma-1} \Longrightarrow T_D = (2120).\left(\frac{0.01}{0.1}\right)^{1.4-1} = \mathbf{843.987}\ \mathrm{K}$$

$$P_C.V_C{}^{\gamma} = P_D.V_D{}^{\gamma} \Rightarrow P_D = P_C.\left(\frac{V_C}{V_D}\right)^{\gamma} = 70.664 \times \left(\frac{0.01}{0.1}\right)^{1.4} = \mathbf{2.813\ bar}$$

Solution of exercise 03 :

1.



2

State	P(bar)	V (liter)	T(K)
A	1	2	293
B	66.289	0.1	971.135
C	66.289	0.3	2913.405
D	4.655	2	1364.07

$$T_A.{V_A}^{\gamma-1} = T_B.{V_B}^{\gamma-1} \Rightarrow T_B = T_A.\left(\frac{V_A}{V_B}\right)^{\gamma-1} \Rightarrow T_B = (293).\left(\frac{2}{0.1}\right)^{1.4-1} = \mathbf{971.135}\ \mathrm{K}$$

$$P_A.{V_A}^{\gamma} = P_B.{V_B}^{\gamma} \Rightarrow P_B = P_A.\left(\frac{V_A}{V_A}\right)^{\gamma} = P_A.(r_v)^{\gamma} = 1 \times \left(\frac{2}{0.1}\right)^{1.4} = \boldsymbol{66.289\ bar}$$

$$\frac{V_C}{V_B} = \frac{T_C}{T_B}\ (P_C = P_B) \Rightarrow\ T_C = T_B \times \frac{V_C}{V_B} = 971.135 \times \frac{0.3}{0.1} = \boldsymbol{2913.405\ K}$$

$$T_C.{V_C}^{\gamma-1} = T_D.{V_D}^{\gamma-1} \Rightarrow T_D = T_C.\left(\frac{V_C}{V_D}\right)^{\gamma-1} \Longrightarrow T_D = (2913.405).\left(\frac{0.3}{2}\right)^{1.4-1} = \mathbf{1364.07}\ \mathrm{K}$$

$$P_C.{V_C}^{\gamma} = P_D.{V_D}^{\gamma} \Rightarrow P_D = P_C.\left(\frac{V_C}{V_D}\right)^{\gamma} = 66.289 \times \left(\frac{0.3}{2}\right)^{1.4} = \boldsymbol{4.655\ bar}$$

Solution of exercise 04

1. The thermal efficiency of the cycle:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{\left[\left(\frac{r_c}{CR}\right)^{\gamma-1}.r_c - (CR)^{1-\gamma}\right]}{\gamma(r_c - 1)} = 1 - (CR)^{1-\gamma}.\frac{[r_c{}^{\gamma} - 1]}{\gamma(r_c - 1)}$$

$$CR = r_v = \frac{V_1}{V_2} = 20 \quad ; \quad r_c = \frac{T_3}{T_2} = \frac{T_3}{T_{1\,.CR^{\gamma-1}}} = \frac{2800}{298.15\times 20^{0.4}} = 2.833$$

$$\eta_{th} 1 - (CR)^{1-\gamma}.\frac{[r_c{}^{\gamma} - 1]}{\gamma(r_c - 1)} = 1 - 20^{-0.4} \times \frac{2.83^{1.4}-1}{1.4\times(2.83-1)} = \mathbf{0.6125 = 61.25\%}$$

Mean effective pressure

$$MEP = \frac{W_{net}}{V_c} = \frac{Q_{in} - Q_{out}}{V_1 - V_2} = P_1.CR.\left(\frac{1}{\gamma-1}\right).\left[\frac{\gamma.CR^{\gamma-1}(r_c-1)-(r_c{}^{\gamma}-1)}{CR-1}\right]$$

$$MEP = 0.95 \times 20 \times \left(\frac{1}{1.4-1}\right).\left[\frac{1.4\times 20^{1.4-1}(2.83-1)-(2.83^{1.4}-1)}{20-1}\right] = \mathbf{13.003 bar}$$

2. The thermal efficiency of cycle (Diesel = Gasoline) :

$$\eta_{th_{Petrol}} = \eta_{th_{Diesel}} = \frac{W_{net}}{Q_{in}} = 1 - \frac{1}{CR^{\gamma-1}} = 0.6125 \Rightarrow \frac{1}{CR^{\gamma-1}} = 1 - \eta_{th} \Rightarrow CR^{\gamma-1} = \frac{1}{1-\eta_{th}}$$

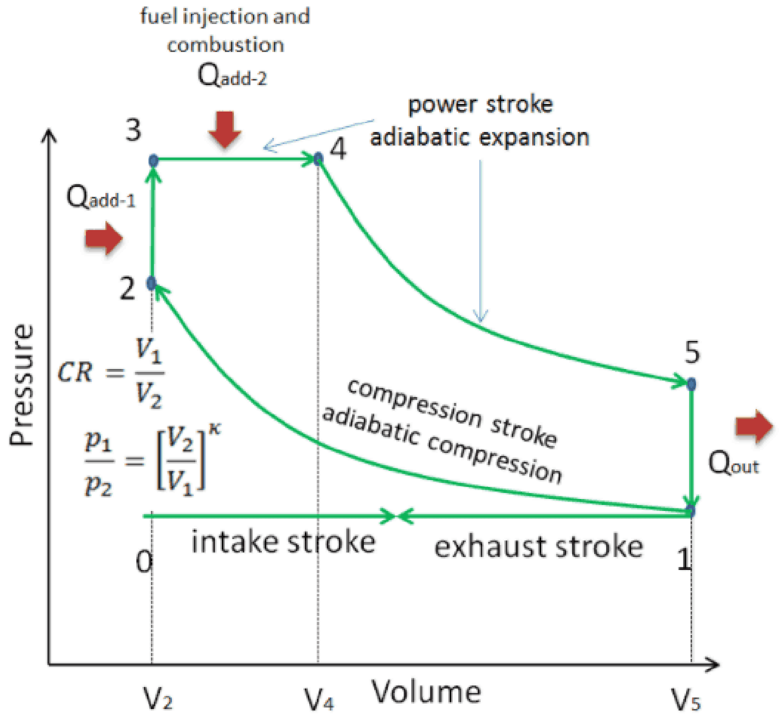
$$CR = \sqrt[1/0.4]{\frac{1}{1-0.6125}} = \mathbf{10.698}$$

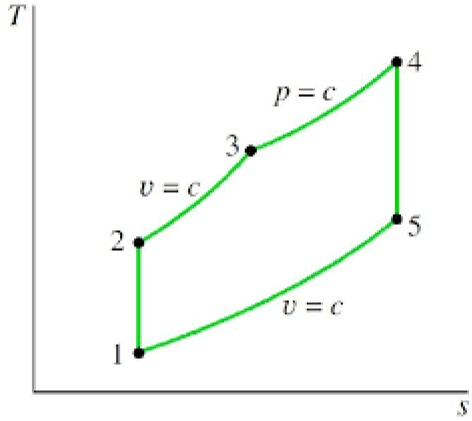
$$\eta_{th_{Petrol}} = 10.698 \quad ; \quad \eta_{th_{Diesel}} = 20$$

The volumetric compression ratio of the spark-ignition engine is half that of the diesel engine, in order to operate at the same thermal efficiency. Furthermore, this engine can be manufactured without the risk of detonation because the compression ratio is less than 12.

Solution of exercise 05

1. Diagram (P-V) and (T-S)





2. Pressure and temperature

Point 1

$P_1 = 1\ bar$; $T_1 = 298.15\ K$

Point 2

1-2 isentropic compression

$$\boldsymbol{T_1.V_1{}^{\gamma-1} = T_2.V_2{}^{\gamma-1} \Rightarrow T_2 = T_1.\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1.(r_v)^{\gamma-1} = 298.15 \times 19^{1.4-1}}$$

$$\boldsymbol{= 968.135\ K}$$

$$P_1.V_1{}^{\gamma} = P_2.V_2{}^{\gamma} \Rightarrow P_2 = P_1.\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1.(r_v)^{\gamma} = 1 \times 19^{1.4} = \boldsymbol{61.69\ bar}$$

Point 3

2-3 constant volume evolution ($V_2=V_3$)

$P_3 =$ **70 bar** ;

$$\begin{cases} P_2V_2 = n.R.T_2 \\ P_3V_3 = n.R.T_3 \end{cases} \Rightarrow \frac{V_3}{V_2} = \frac{P_3}{P_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 968.135 \times \frac{70}{61.69} = \boldsymbol{1098.549\ K}$$

Point 4

3-4 constant volume evolution (**P_4**=P_3 = **70 bar**)

T4 =2100 K

Point 5

5-1 constant volume expansion

$$\frac{P_5}{P_1}=\frac{T_5}{T_1}\Rightarrow T_5=T_1.\frac{P_5}{P_1}$$

4-5 isentropic expansion

$$T_4.P_4^{\frac{\gamma-1}{\gamma}}=T_5.P_5^{\frac{\gamma-1}{\gamma}}$$

$$P_5^{1-\gamma}.T_5^{\gamma}=P_4^{1-\gamma}.T_4^{\gamma}\Rightarrow P_5^{1-\gamma}.T_1^{\gamma}.P_5^{\gamma}.P_1^{-\gamma}=P_4^{1-\gamma}.T_4^{\gamma}\Rightarrow P_5=P_4^{1-\gamma}.P_1^{\gamma}.\left(\frac{T_4}{T_1}\right)^{\gamma}$$

$$P_5=70^{1-1.4}\times 1^{1.4}\times\left(\frac{2100}{298.15}\right)^{1.4}=\boldsymbol{2.81\ bar}$$

$$T_5=298.15\times\frac{2.81}{1}=\boldsymbol{837.80\ K}$$

3.

$$Q_{2-3}=C_V(T_3-T_2)=\frac{r}{\gamma-1}(T_3-T_2)=\frac{287}{1.4-1}(1098.549-968.135)=\boldsymbol{93.572\ kJ/kg}$$

$$Q_{3-4}=C_P(T_4-T_3)=\frac{\gamma.r}{\gamma-1}(T_3-T_2)=\frac{1.4\times 287}{1.4-1}(2100-1098.549)=\boldsymbol{1005.957\ kJ/kg}$$

$$Q_{5-1}=C_V(T_1-T_5)=\frac{r}{\gamma-1}(T_1-T_5)=\frac{287}{1.4-1}(298.15-837.8)=\boldsymbol{-387.199\ kJ/kg}$$

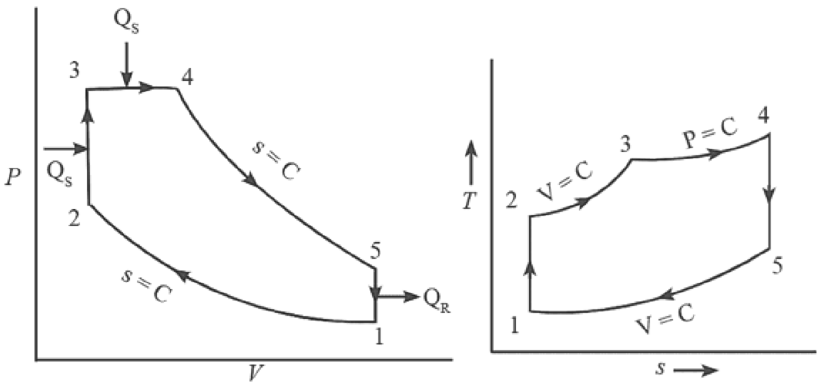
4.

$$\eta_{th}=\frac{W_{net}}{Q_{in}}=1-\frac{Q_{out}}{Q_{in}}=1-\frac{|m.C_v.(T_1-T_5)|}{|m.C_v.(T_3-T_2)+m.C_p.(T_4-T_3)|}=1-\frac{C_v.(T_5-T_1)}{C_v.(T_3-T_2)+C_p.(T_4-T_3)}$$

$$\eta_{th}=1-\frac{387.199}{93.572+1005.957}=0.6478=\boldsymbol{64.78}\%$$

Solution of exercise 06

1.



PV Diagram T-S Diagram

2.

Point 1

P_1 = **1 bar** ; $T_1 = (20 + 273.15) =$ **293.15 K**

$$P_1.V_1 = R.T_1 \Rightarrow V_1 = \frac{R.T_1}{P_1} = \frac{287 \times 293.15}{10^5} = \mathbf{0.84134\ m^3/kg}$$

Point 2

$$\boldsymbol{T_1.V_1{}^{\gamma-1} = T_2.V_2{}^{\gamma-1} \Rightarrow T_2 = T_1.\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1.(r_v)^{\gamma-1}}$$

$$\Rightarrow T_2 = (20 + 273.15).(20)^{1.4-1} = \mathbf{971.63\ K}$$

$$P_1.V_1{}^{\gamma} = P_2.V_2{}^{\gamma} \Rightarrow P_2 = P_1.\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1.(r_v)^{\gamma} = 1 \times 20^{1.4} = \mathbf{66.289\ bar}$$

$$P_2.V_2 = R.T_2 \Rightarrow V_2 = \frac{R.T_2}{P_2} = \frac{287 \times 971.63}{66.289 \times 10^5} = \mathbf{0.04206\ m^3/kg}$$

Point 3

P_3 = **70 bar** ; $V_3 = V_2 = \mathbf{0.04206\ m^3/kg}$

$$\begin{cases} P_2V_2 = n.R.T_2 \\ P_3V_3 = n.R.T_3 \end{cases} \Rightarrow \frac{V_3}{V_2} = \frac{P_3}{P_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 971.63 \times \frac{70}{66.289} = \mathbf{1026.024\ K}$$

Point 4

$P_4 = P_3$ = **70 bar** ; $T_4 = (2000 + 273.15) =$ **2273.15 K**

$$\begin{cases} P_4V_4 = n.R.T_4 \\ P_3V_3 = n.R.T_3 \end{cases} \Rightarrow \frac{T_4}{T_3} = \frac{V_4}{V_3} \Rightarrow V_4 = V_3 \times \frac{T_4}{T_3} = 0.04206 \times \frac{2273.15}{1026.024} = \mathbf{0.09318\ K}$$

Point 5

$V_5 = V_1 = \mathbf{0.84134\ m^3/kg}$

$$\boldsymbol{T_4.V_4}^{\gamma-1} = \boldsymbol{T_5.V_5}^{\gamma-1} \Rightarrow \boldsymbol{T_5} = \boldsymbol{T_4}.\left(\frac{\boldsymbol{V_4}}{\boldsymbol{V_5}}\right)^{\gamma-1} = \mathbf{2273.15} \times \left(\frac{\mathbf{0.09318}}{\mathbf{0.84134}}\right)^{\mathbf{0.4}} = \mathbf{942.7}\ \boldsymbol{K}$$

$$P_5.V_5{}^{\gamma} = P_4.V_4{}^{\gamma} \Rightarrow P_5 = P_4.\left(\frac{V_4}{V_5}\right)^{\gamma} = 70 \times \left(\frac{0.09318}{0.84134}\right)^{1.4} = \mathbf{3.216}\ \boldsymbol{bar}$$

3. 1-2 Isentropic

$$\boldsymbol{Q_{1-2} = 0\ J}$$

$$W_{1-2} = \frac{1}{\gamma-1}(P_2V_2 - P_1V_1) = \frac{1}{1.4-1}(66.289 \times 0.042 - 1 \times 0.84134) \times 10^5 = \mathbf{485057}\ \boldsymbol{kJ}$$

2-3 Constant –Volume

$\boldsymbol{W_{2-3} = 0\ J}$ ***(dV = 0)***

$$Q_{2-3} = C_V(T_3 - T_2) = \frac{r}{\gamma-1}(T_3 - T_2) = \frac{287}{1.4-1}(1026.024 - 971.63) = \mathbf{39.0277}\ \boldsymbol{kJ/kg}$$

3-4 Constant –Pressure

$$W_{3-4} = \int P.dV = P(V_4 - V_3) = 70 \times (0.09318 - 0.04206) = \mathbf{357.84}\ \boldsymbol{kJ}$$

$$Q_{3-4} = C_P(T_4 - T_3) = \frac{\gamma.r}{\gamma-1}(T_3 - T_2) = \frac{1.4 \times 287}{1.4-1}(2273.15 - 1026.024) = \mathbf{1252.783}\ \boldsymbol{kJ/kg}$$

4-5 Isentropic

$$\boldsymbol{Q_{4-5} = 0\ J}$$

$$W_{4-5} = \frac{1}{\gamma-1}(P_5V_5 - P_4V_4) = \frac{1}{1.4-1}(3.216 \times 0.84134 - 70 \times 0.09318) \times 10^5 = \mathbf{-954.21}\ \boldsymbol{kJ}$$

5-1 Constant –Volume

$\boldsymbol{W_{5-1} = 0\ J}$ ***(dV = 0)***

$$Q_{5-1} = C_V(T_1 - T_5) = \frac{r}{\gamma-1}(T_1 - T_5) = \frac{287}{1.4-1}(293.15 - 942.7) = \mathbf{-466.052}\ \boldsymbol{kJ/kg}$$

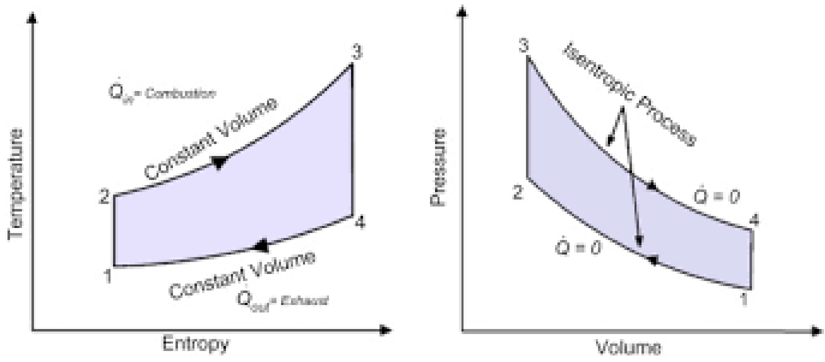
4. The thermal efficiency

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{|m.C_v.(T_1-T_5)|}{|m.C_v.(T_3-T_2) + m.C_p.(T_4-T_3)|} = 1 - \frac{C_v.(T_5-T_1)}{C_v.(T_3-T_2) + C_p.(T_4-T_3)}$$

$$\eta_{th} = 1 - \frac{466.052}{39.0277 + 1252.783} = 0.6392 = \mathbf{63.92}\%$$

Solution of exercise 7 :

1/



2/ Calculation of pressure and temperature at the peaks of the cycle:

Point 1 :

P_1 = 0.89 bars **T_1 = 25 °C = 298.15 K**

Point 2 : isentropic compression, which allows us to write:

$$T_1.V_1{}^{\gamma-1} = T_2.V_2{}^{\gamma-1} \Rightarrow T_2 = T_1.\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1.(r_v)^{\gamma-1}$$

$$\boldsymbol{T_2} = 298.15 \times (10.1)^{1.4-1} = \mathbf{751.905}\ \boldsymbol{K}$$

$$P_1.V_1{}^{\gamma} = P_2.V_2{}^{\gamma} \Rightarrow P_2 = P_1.\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1.(r_v)^{\gamma} \Rightarrow \boldsymbol{P_2} = 0.89 \times (10.1)^{1.4} = \mathbf{22.669}\ \boldsymbol{bars}$$

Point 3 : end of combustion

$$Q_{in} = Q_{23} = C_v.(T_3 - T_2) \Rightarrow T_3 = \frac{Q_{in}}{c_v} + T_2 \Rightarrow \boldsymbol{T_3} = \frac{4300}{1.62} + 751.905 = \mathbf{3406.226}\ \boldsymbol{K}$$

Since combustion occurs according to an isochoric process, then:

$$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow P_3 = P_2 \times \frac{T_3}{T_2} \Rightarrow \boldsymbol{P_3} = 22.669 \times \frac{3406.226}{751.905} = \mathbf{102.693}\ \boldsymbol{bars}$$

Point 4 : The burnt gases undergo isentropic expansion, which allows us to write:

$$T_3.V_3{}^{\gamma-1} = T_4.V_4{}^{\gamma-1} \Rightarrow T_4 = T_3.\left(\frac{V_3}{V_4}\right)^{\gamma-1} = T_3.\left(\frac{V_1}{V_2}\right)^{1-\gamma} = T_3.(r_v)^{1-\gamma}$$

$$\boldsymbol{T_4} = 3406.226 \times (10.1)^{-0.4} = \mathbf{1350.656}\ \boldsymbol{K}$$

$$P_3.V_3{}^{\gamma} = P_4.V_4{}^{\gamma} \Rightarrow P_4 = P_3.\left(\frac{V_3}{V_4}\right)^{\gamma} = P_3.\left(\frac{V_1}{V_2}\right)^{-\gamma} = P_3.(r_v)^{-\gamma}$$

$$\Rightarrow \boldsymbol{P_4} = 102.693 \times (10.1)^{-1.4} = \mathbf{4.031}\ \boldsymbol{bars}$$

3/ The work done is determined from the expression of output:

$$\eta_{th} = 1 - \frac{1}{r_v{}^{\gamma-1}} = \frac{W_{net}}{Q_{in}} \Rightarrow W_{net} = Q_{in} \times \left(1 - \frac{1}{r_v{}^{\gamma-1}}\right)$$

$$\boldsymbol{W_{net}} = 4300 \times \left(1 - \frac{1}{10.1^{1.4-1}}\right) = \mathbf{2594.939}\ \boldsymbol{kJ/kg}$$

$$W_{net} = Q_{in} - Q_{out} \Rightarrow \boldsymbol{Q_{out}} = Q_{in} - W_{net} = 4300 - 2594.939 = \mathbf{1705.061}\ \boldsymbol{kJ/kg}$$

4/ The motor cycle maintains the same states 1, 2, and 3. The only differing state is the one corresponding to the end of the expansion, that is, 4, which changes to 4': Therefore, the expansion is 3-4':

$$P_3.V_3{}^{\eta} = P_{4'}.V_{4'}{}^{\eta} \Rightarrow P_{4'} = P_3.\left(\frac{V_3}{V_{4'}}\right)^{\eta} = P_3.(r_v)^{-\eta}$$

$$\Rightarrow \boldsymbol{P_{4'}} = 102.693 \times (10.1)^{-1.28} = \boldsymbol{5.321\ bars}$$

$$T_3.V_3{}^{\eta-1} = T_{4'}.V_{4'}{}^{\eta-1} \Rightarrow T_{4'} = T_3.\left(\frac{V_3}{V_{4'}}\right)^{\eta-1} = T_3.\left(\frac{V_1}{V_2}\right)^{1-\eta} = T_3.(r_v)^{1-\eta}$$

$$\boldsymbol{T_{4'}} = 3406.226 \times (10.1)^{-0.28} = \boldsymbol{1782.639\ K}$$

5/ The work generated by the cycle:

$$\boldsymbol{W_{net}} = Q_{in} - Q_{out} = Q_{in} - C_v(T_{4'} - T_1) = 4300 - 1.62 \times (1782.639 - 298.15)$$

$$= \boldsymbol{1895.127\ kJ/kg}$$

2.7. **Exercises with solution**

Problem (2.7.1): The compression ratio of an air-standard Otto cycle is 9.5. Prior to the isentropic compression process, the air is at 100 kPa, 35°C and 600 cm^3. The temperature at the end of the isentropic expansion process is 800 K. Determine (a) the highest temperature in the cycle (b) the highest pressure in the cycle (c) the amount of heat transferred in (d) the thermal efficiency (e) the mean effective pressure.

Ans. (1969 K, 6072 kPa, 0.59 kJ, 59.4%, 652 kPa)

Problem (2.7.2): An ideal diesel engine has a compression ratio of 20 and uses air as the working fluid. The state of air at the beginning of the compression process is 95 kPa and 20°C. If the maximum temperature in the cycle is not to exceed 2200 K, determine (a) the thermal efficiency (b) the mean effective pressure.

Ans. (63.5%, 933 kPa)

Chapter III: Analyzes of real cycle of a diesel engine

3.1. Actual cycle of the internal combustion engine

It is undeniable that an internal combustion engine cannot operate according to the theoretical cycle. In fact, theoretical cycles disregard several important and crucial elements in the correct functioning of the engine:

1. Heat exchange through the walls cannot be eliminated. This results in pressure variations inside the engine cylinder.
2. Pressure losses during the flow of fresh or exhaust gases through the pipes and manifolds are not negligible.
3. Pumping losses due to flow through the throttle and valves are not negligible.
4. The engine cylinder does not fill properly, especially when the rotational speed is very high.
5. The exhaust gas scavenging is not done properly, especially when the rotational speed is very high.
6. Combustion is very rapid but not instantaneous, and it lasts for a significant fraction of the piston's stroke. Therefore, ignition or injection exactly at the end of the compression stroke is not appropriate.

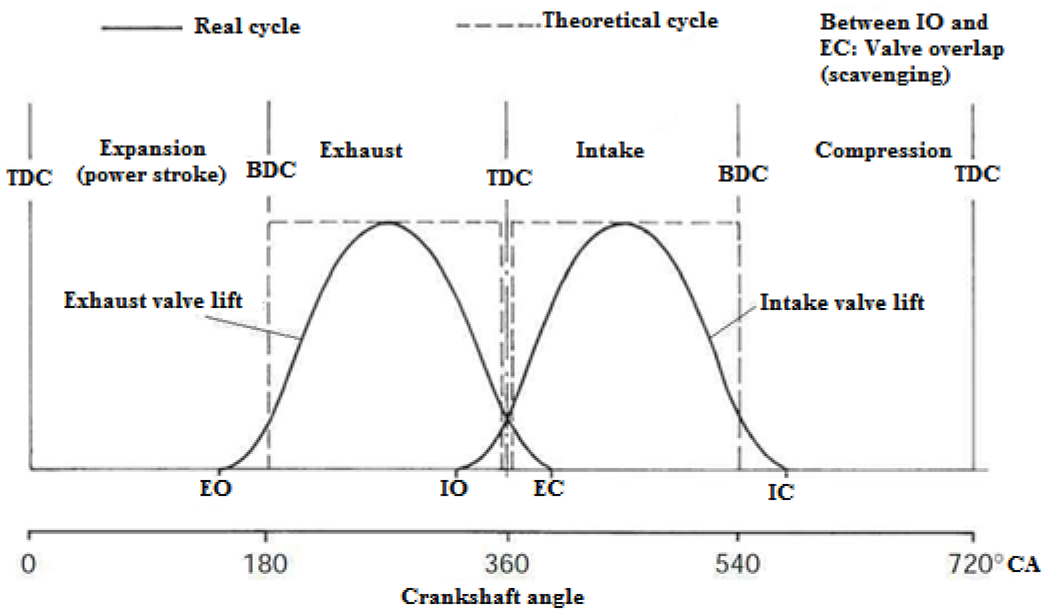


Figure 3.1. *Valve stroke during a cycle.*

According to Table 3.1, if we refer to the theoretical cycles, the time allocated to the intake and exhaust phases decreases from 30 milliseconds to 7.5 milliseconds when the speed increases from 1000 rpm to 4000 rpm. Therefore, if we maintain a time equivalent to 180 degrees crankshaft, the draining and filling of the cylinders cannot be done correctly at very high engine speeds.

Table 3.1. *Intake and exhaust durations at different speeds.*

N (rpm)	In degrees Crankshaft	In milliseconds
1000	180°CA	30
2000	180°CA	15
2800	180°CA	10.7
3000	180°CA	10
4000	180°CA	7.5

In order to take into account all the elements mentioned, we were led to:

1. Modify the valve opening and closing times. The intake valve begins to open before top dead center (IOA or intake valve opening advance) and closes after bottom dead center (ICD or intake valve closing delay). The exhaust valve begins to open before bottom dead center (EOA or exhaust valve opening advance) and closes after top dead center (ECD or exhaust valve closing delay). The intake and exhaust phases will be longer. Thus, proper scavenging and cylinder filling are insured.

2. In a spark-ignition engine, the spark is triggered a few degrees before the end of the compression stroke (IA or ignition advance), which allows the fresh gases in the cylinder.

3. In a compression-ignition engine, fuel injection is initiated a few degrees before the end of the compression stroke (IA – injection advance). This allows the fuel spray to atomize, evaporate, and mix properly with the air, resulting in more complete and smoother combustion.

3.2. Intake Phase

The piston moves from TDC to BDC. The intake valve is already open and the exhaust valve is closing. The motion of the piston creates a vacuum of about 0.1 to 0.2 bar. Pressure losses due to the restricted flow area of the intake valve affect cylinder filling and therefore engine power. The higher the engine speed, the shorter the intake valve opening duration. Complete filling of the cylinders is not possible. In addition, a small percentage of the burned gases from the previous cycle remains in the cylinder. The actual volumetric efficiency of the engine is about 80%.

To address the issues mentioned, the following solutions are generally used:

- Intake pipes and manifolds that are smooth, minimally curved, and as short as possible.
- Intake valves with larger diameters than exhaust valves.
- Combustion chambers that are as compact as possible.
- Turbocharging or supercharging when possible.

The limit values for the opening advance and closing delay of each valve are as follows:

IOA is between 10 and 30°CA,

ICD is between 40 and 60 °CA,

EOA is between 35 and 60 °CA,

ECD is between 05 and 20 °CA.

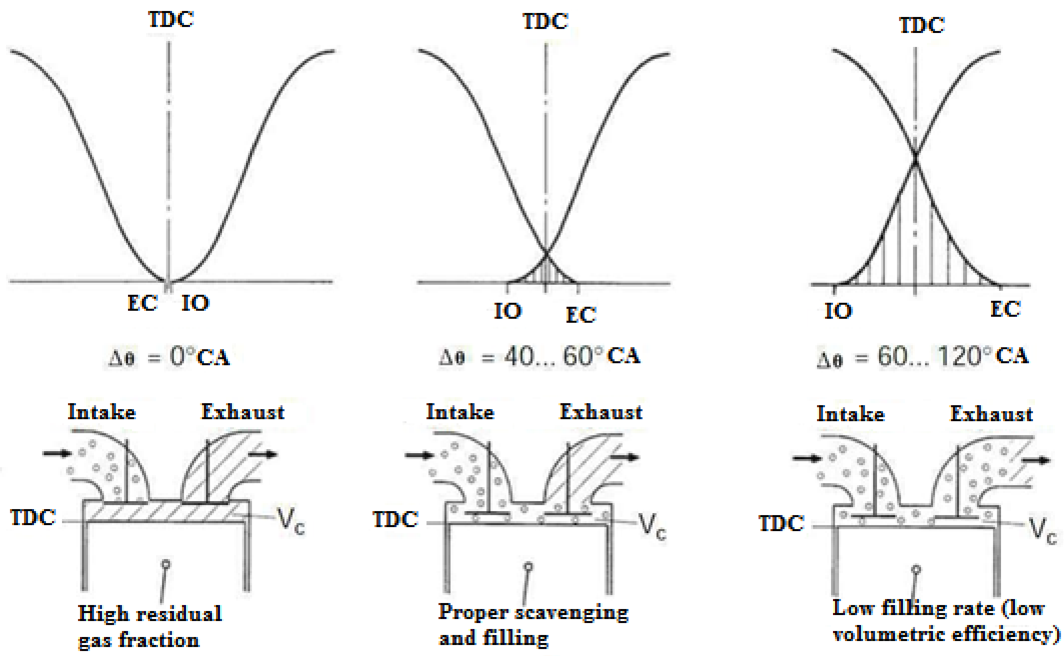


Figure 3.2. *Influence of valve crossing (Scavenging) on filling.*

Figure 3.2, shows the influence of the choice of valve opening and closing timing on the volumetric efficiency and the amount of residual gases in the cylinder. If the intake valve begins to open at the same time as the exhaust valve closes, a high level of residual gases is obtained. In addition, the valve seats will become coated with soot, and improper sealing will occur when the valves close, leading to a loss of compression.

Thus, the amount of fresh charge introduced is reduced. However, if the valve overlap phase is too prolonged, poor cylinder filling will result because the fresh gases leave the cylinder prematurely. Judicious selection of the appropriate angles for opening and closing each valve is essential to ensure proper scavenging and correct cylinder filling.

Table 3.2. *Engine technical specifications.*

	Engine 1	**Engine 2**	**Engine 3**
Kind	Experimental	Automobile	Industrial
Designation	Lister-Petter SR1	K9K766	MKDIR 620-145
Number of cylinders	1	4	6
Engine displacement [cm^3]	631	1461	9840
D [cm]	9,525	7.6	12
S [cm]	8.85	8.05	14.5
L_{CR} [cm]	16.5	13,375	22.7
CR or **ε**	18:1	17.9:1	17:1
Turbocharger	Without	kkk - Wastegate	Schweitzer - GV
Injection system	Unit pump	DCi injection	VR distribution pump
Static AI	18°CA before TDC	8°CA before TDC	8°CA before TDC
IOA	-36 °CA	+09 °CA	-06 °CA
ICD	+69 °CA	+20 °CA	+36 °CA
EOA	-76 °CA	-27 °CA	-50 °CA
ECD	+32 °CA	+07 °CA	+16 °CA
Maximum power	5.5 kW at 1800 rpm	60 kW at 3750 rpm	264 kW at 2400 rpm

Table 3.3 presents some technical data on three direct injection diesel engines from different manufacturers. The engines chosen as examples are a marine engine, an automotive engine, and an industrial engine.

Exercise 3.1: From the timing angles shown in Table 3.2, give the duration of intake, compression, power-expansion, and exhaust phases of each engine.

Solution of exercise 3.1:

Engine 1:

Intake duration = 180 – IOA + ICD = 180 –(-36) +69 = **285 °CA**

Compression duration = 180 – ICD – AI = 180 -69 – 18 = **93 °CA**

Power – Expansion duration =180 + AI +EOA = 180 +18 + (-76) =**122 °CA**

Exhaust duration = 180 –EOA + ECD = 180 –(-76) + 32 = **288 °CA**

Valve(s) overlap duration = -IOA + ECD = -(-36) + 32 = **68 °CA**

Engine 2:

Intake duration = 180 – IOA + ICD = 180 –(+9) +20 = **191 °CA**

Compression duration = 180 – ICD – AI = 180 -20 – 8 = **152 °CA**

Power – Expansion duration= 180 + AI +EOA = 180 +8 + (-27) =**161 °CA**

Exhaust duration = 180 –EOA + ECD = 180 –(-27) + 7 = **214 °CA**

Valve(s) overlap duration = -IOA + ECD = -(9) + 7 = -2 (no overlap) = **00 °CA**

Engine 3:

Intake duration = 180 – IOA + ICD = 180 –(-6) +36 = **222 °CA**

Compression duration = 180 – ICD – AI = 180 -36 – 8 = **136 °CA**

Power – Expansion duration= 180 + AI +EOA = 180 +8 + (-50) =**138 °CA**

Exhaust duration = 180 –EOA + ECD = 180 –(-50) + 16 = **246 °CA**

Valve(s) overlap duration = -IOA + ECD = -(-6) + 16 = **22 °CA**

Table 3.3. *Intake and exhaust durations of the engines (in Table 3.2).*

	Engine 1	**Engine 2**	**Engine 3**
Duration of Admission	285 °CA	191 °CA	222 °CA
Exhaust Duration	288 °CA	214 °CA	246 °CA
Valve overlap duration	68°CA	00 °CA	22 °CA

3.2. Compression Phase

With both valves closed, the piston compresses the fresh gases in the combustion chamber. The pressure and temperature of the mixture increase. These conditions promote rapid evaporation and homogenization of the mixture in diesel engines. The higher the compression ratio, the higher the power output because the contact time of the burnt gases with the cylinder walls decreases (less heat loss). The higher the end-compression pressure, the higher the maximum pressure of the cycle and the longer the expansion phase (power stroke) . Therefore, more mechanical work is recovered. However, the compression ratio should not be increased arbitrarily. In diesel engines, a high compression ratio leads to pre-ignition detonation of the gases (knocking). This creates pressure waves on the cylinder and piston. This can result in high fuel consumption and a drop in power, accompanied by noise, piston seizure, and cylinder wear. In diesel engines, a high compression ratio leads to larger component dimensions and consequently higher mechanical losses.

Taking as an example the case of a diesel engine that runs at a speed of 2800 rpm with an injection advance of 20 degrees crankshaft. By examining the curve showing the variation of the average pressure in the cylinder as a function of the crankshaft rotation angle.

It can be observed in Figure 3.3 that, for this engine, rapid combustion only begins after a certain delay composed of:

- A physical delay of approximately 10 degrees crankshaft injection followed by 7 degrees crankshaft vaporization.
- A chemical delay corresponding to the flameless oxidation combustion phase of approximately 3 degrees crankshaft.
- A very rapid main combustion phase with a flame lasting approximately 5 crankshaft degrees.
- A diffusion flame combustion phase.

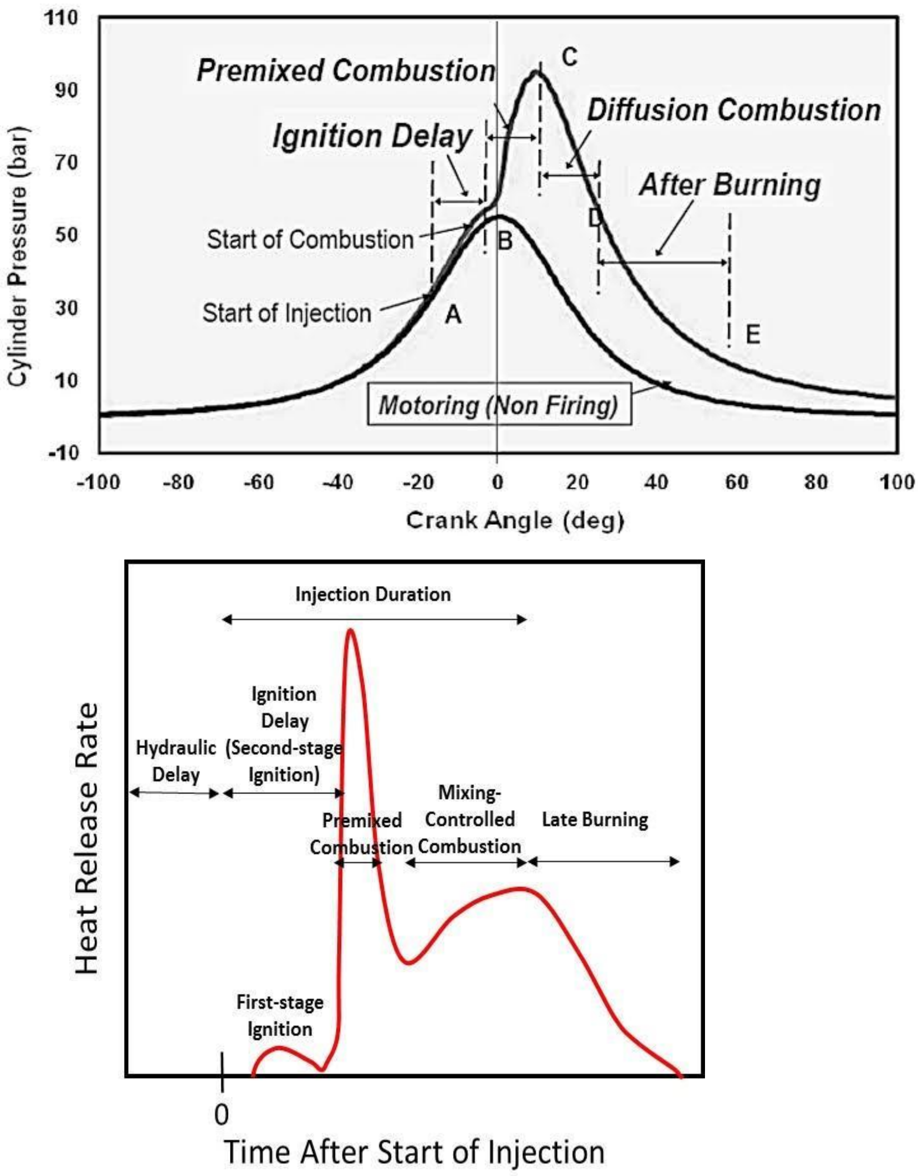


Figure 3.3. *Typical cylinder pressure and heat release curve in a diesel engine*

The injection phase lasts 40 crankshaft degrees, while combustion lasts only 20 crankshaft degrees. Combustion is significantly delayed, but with advanced injection timing, it begins precisely when the piston is at the end of its compression stroke. Table 3.4 illustrates a comparative analysis of the injection timing, duration, and combustion duration at different engine speeds. The higher the engine rotational speed, the shorter the delays become.

For this purpose, injection timing advance devices are used in diesel engine injection systems, capable of adjusting the injection angle according to engine speed. Proper engine operation is thus ensured regardless of the rotational speed.

Table 3.4. *Auto-ignition time and injection and combustion durations.*

N (rpm)	Auto-ignition delay (milliseconds)	Injection time (milliseconds)	Combustion time (milli-seconds)
1000	3.33	6.66	3.33
2000	1.67	3.34	1.67
2800	1.19	2.38	1.19
3000	1.00	2.00	1.00
4000	0.83	1.66	0.83

3-3. Combustion expansion phase

The expansion phase corresponds to a power stroke of the piston. During this phase, combustion and expansion of the gases partially occur. With both valves closed, combustion is initiated either by a spark with a certain advance or by auto-ignition or by self-ignition of the two-phase fuel jet after its evaporation and homogenization with the air. The flammability conditions are at a temperature of approximately 900 K. In an engine has ignition order there maximum combustion pressure is approximately 40 has 60 bar. In a diesel engine, the maximum pressure can exceed 100 bar. The maximum temperature is in the range of 2200 K to 2700 K, while the flame front speed is 15 to 30 m/s. Under the effect of gas pressure, the piston is forced downwards and the crankshaft completes half a revolution.

3.4. Exhaust phase

This is the exhaust (discharge) phase of the exhaust gases. The intake valve is closed while the exhaust valve opens. The pressure in the cylinder rapidly drops to 4 to 7 bar. The combustion products are expelled by the piston until a pressure of about 1.1 bar is reached, with temperatures ranging from 500 K to 800 K at idle and from 900 K to 1300 K at full load.

Valve overlap may occur during the end of the exhaust phase. The overlap duration is very short in order to prevent gases from flowing back into the engine cylinder. This results in better filling with fresh gases, especially at high engine speeds.

Figure 3.4 shows the Clapeyron diagram of the real cycle of a compression ignition engine. The real cycle is far from the ideal diesel cycle. Indeed, the intake and exhaust phases do not occur at constant pressure. Combustion takes place neither instantaneously nor at constant pressure.

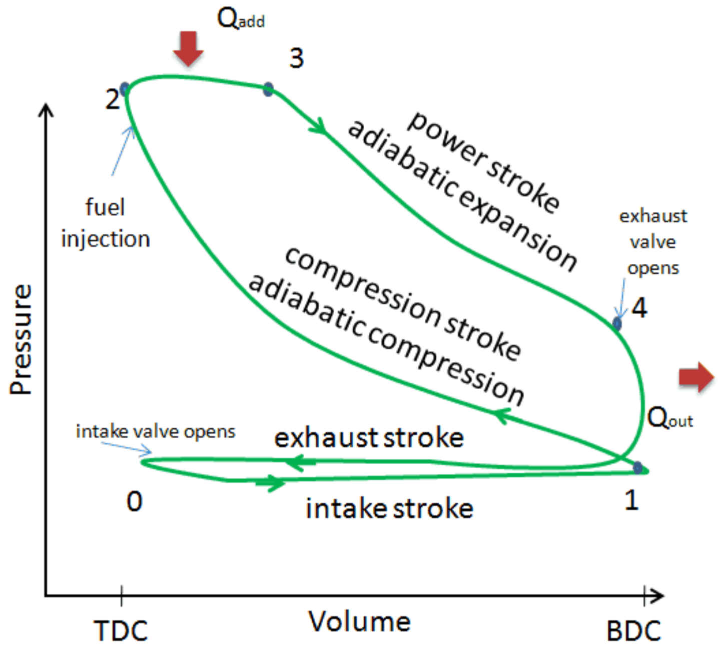


Figure 3.4. *Diagram (PV) of the actual cycle of a diesel engine.*

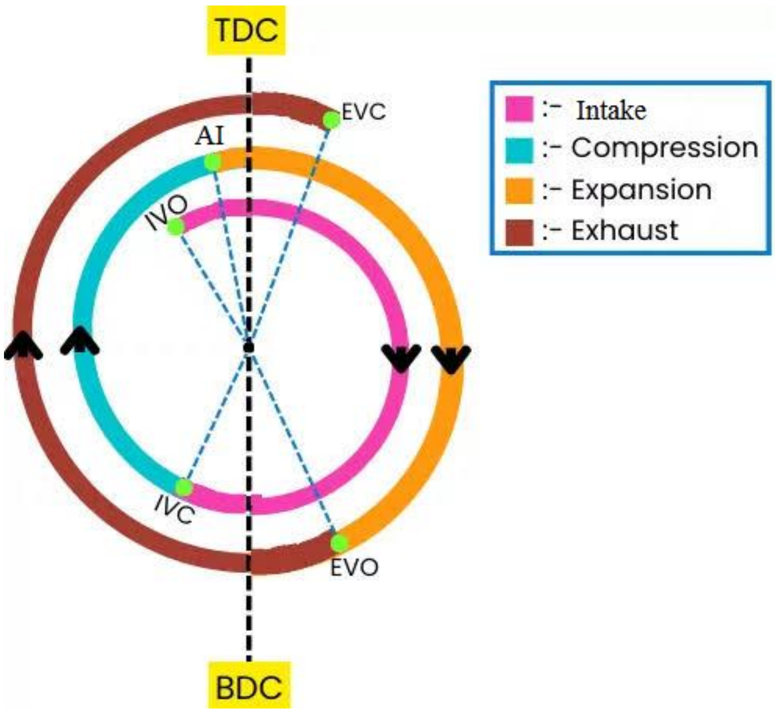
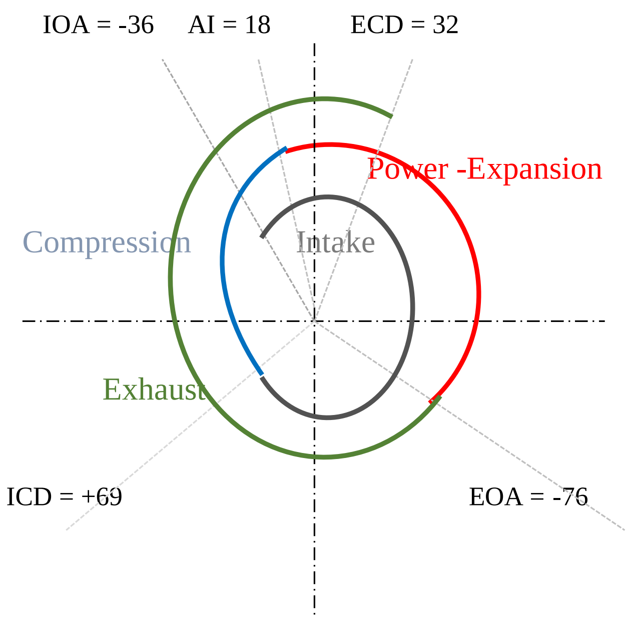


Figure 3.5. Valve Timing Diagram for a 4-stroke internal compression ignition engine.

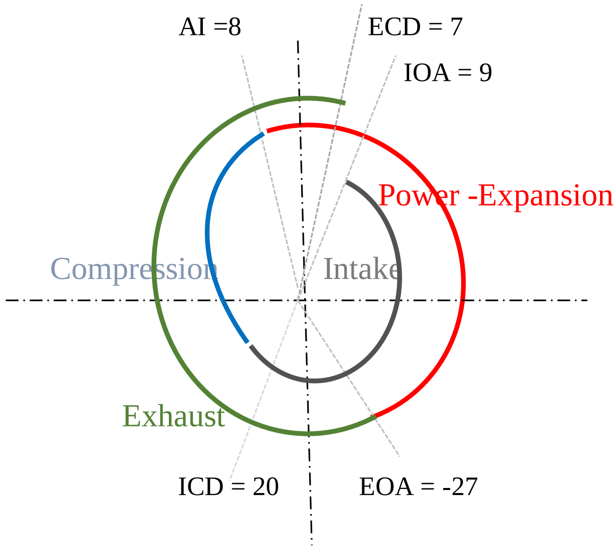
This figure illustrates the valve timing of a four-stroke engine over a complete 720° crankshaft cycle, highlighting the overlap of the intake, compression, combustion (expansion), and exhaust phases, unlike the theoretically perfectly separated phases. It shows that the intake valve opens before top dead center (TDC) and closes after bottom dead center (TDC), while the exhaust valve also opens before the end of the power stroke and closes slightly after TDC, creating a short overlap period. This valve timing improves cylinder filling and exhaust gas evacuation, especially at high engine speeds. The diagram also underscores that the actual engine cycle differs from the ideal theoretical cycle, as valve opening and closing are advanced or retarded to optimize performance and efficiency.

Exercise 3.2: From the timing angles shown in exercise 3.1, give the circular Timing Diagram of intake, compression, power-expansion, and exhaust stroke of each engine.

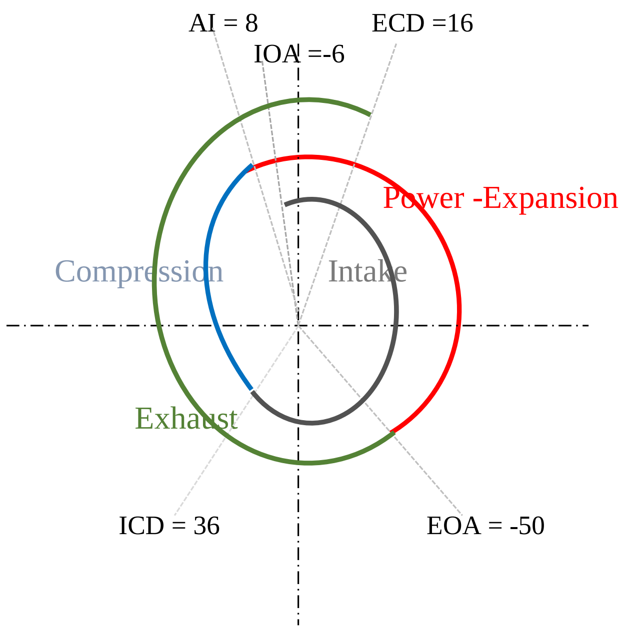
Engine 1



Engine 2



Engine 3



Exercise 3.3 :

An internal combustion engine has the following characteristics:

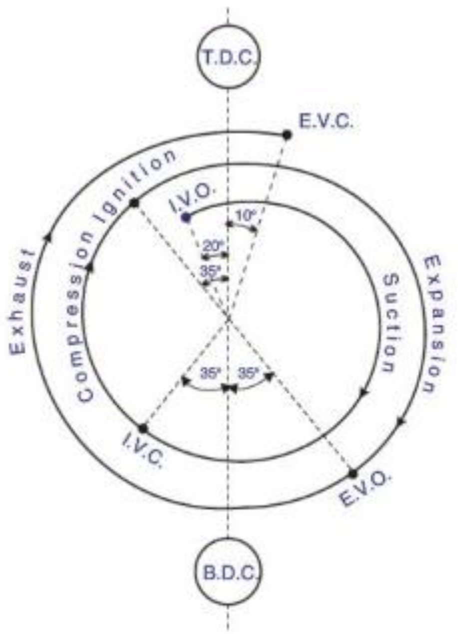
- IOA (intake opening advance) = -20°CA
- ICD (intake closing delay) = 35°CA
- EOA (exhaust opening advance) = -35°CA
- ECD (exhaust closing delay) = 10°CA
- AI (ignition advance) = 35°BTDC

1. Draw the circular timing diagram (real and theoretical)of this engine

2. Calculate the angle travelled by the crank of each stroke during one engine cycle.

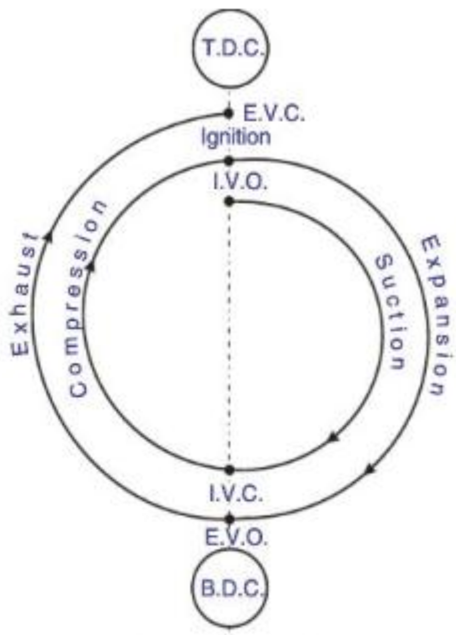
Solution of exercise 3.3

1. *Valve timing diagram:*



2.

. Actual valve timing diagram



Theoretical valve timing diagram

Intake duration = 180 – IOA + ICD = 180 –(-20) +35 = **235 °CA**

Compression duration = 180 – ICD – AI = 180 -35 – 35 = **110 °CA**

Power – Expansion duration =180 + AI +EOA = 180 +35 + (-35) =**180 °CA**

Exhaust duration = 180 –EOA + ECD = 180 –(-35) + 10 = **225 °CA**

Valve(s) overlap duration = -IOA + ECD = -(-20) + 10 = **30 °CA**

Example 3.3 :

In actual practice it is difficult to open &close the valve instantaneously so to get better perform of the engine the valve timing diagram are modified.

- The inlet valve open 10°-30°before TDC to enable fresh charge into the cylinder & help to sent the burnt gases at the same time.
- The inlet valve closes 30°-40°after BDC in order to allow additional time for air-fuel mixture to flow into the cylinder .It increases volumetric efficiency.
- The spark plug produces a spark 30°-40° before TDC thus fuel gets more time to burn.
- Exhaust valve opens 30°-60° before TDC and closes 25° after TDC in order to give more time to start leaving the cylinder

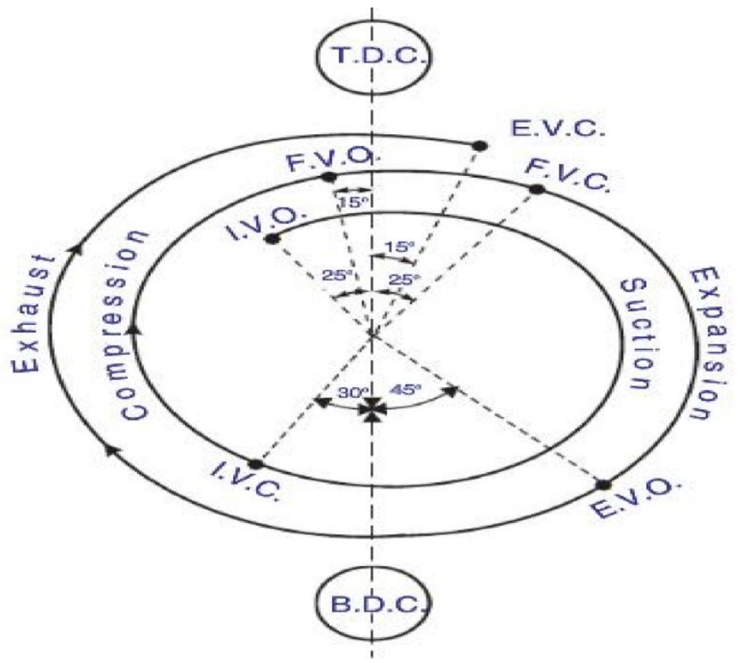
Draw the circular timing diagram (real and theoretical)of this engine

Solution of exercise 3.3

We consider that :

IVO = -25°CA; IVC = 30°CA ; EVO = -45°CA ; EVC = 25°CA ; FVO = 15°BTDC ;

Injection Duration = FVC – FVO = 25 –(-15) = 40 °CA



. Actual valve timing diagram (four-stroke Diesel cycle engines).

Exercise 3.4:

Consider a four-stroke, four-cylinder in-line engine, characterized by:

- Bore, D = 6.4 cm
- Stroke, S = 7 cm
- Combustion chamber volume (clearance volume), V_0 = 13.25 cm³
- IOA (intake opening advance) = -20°CA
- ICD (intake closing delay) = 28°CA
- EOA (exhaust opening advance) = -32°CA
- ECD (exhaust closing delay) = 5°CA
- AI (ignition advance) = 10°BTDC

1. Draw the circular timing diagram of this engine and calculate the angle travelled by the camshaft during each stroke of one engine cycle.
2. Determine the total engine displacement.
3. Draw the crankshaft configuration diagram of this engine.
4. What is the most commonly used firing order? Justify your answer with a table.
5. Calculate the compression ratio of this engine.

For a speed N = 1500 rpm, the engine torque C = 22 N.m:

6. Determine the effective power.
7. Determine the camshaft speed.

This engine operates according to the Diesel cycle.

Given:

- Heat supplied per cycle: $Q_{in} = 600\ kJ/kg$
- $T = 25°C, P = 1\ bar$ (initial temperature and pressure)

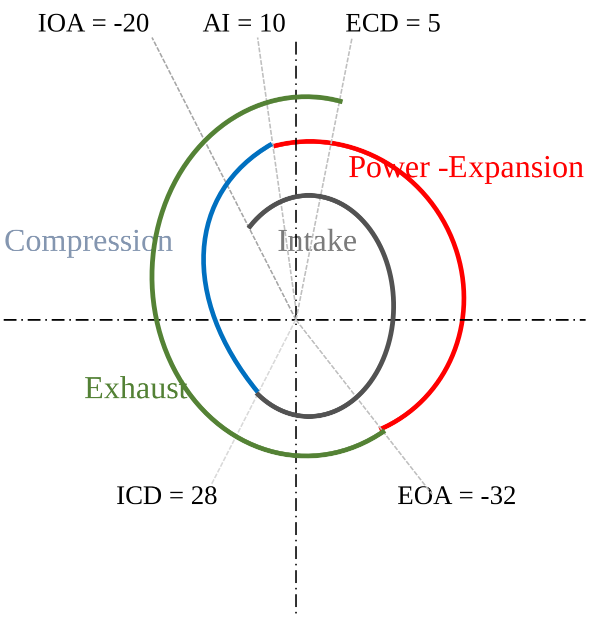
8. Construct the cycle on a P–V diagram and calculate P and T for the different processes.
9. Calculate the thermal efficiency.
10. Compare the obtained thermal efficiency with that of an engine operating on the Otto cycle with the same compression ratio.

Given:

- $\gamma = 1.4$
- Cp = 1 kJ/kg·K
- Cv = 0.715 kJ/kg·K

Solution of exercise 3.4 :

1. Circular diagram of the crankshaft showing the different engine strokes



2 roads of crank give **1** road of camshaft

Intake duration = (180 – IOA + ICD)/2 = (180 – (-20) +28)/2 = **114 degree of Camshaft**

Compression duration = (180 – ICD – AI)/2 = (180 -28 – 10)/2 = **71 degree of Camshaft**

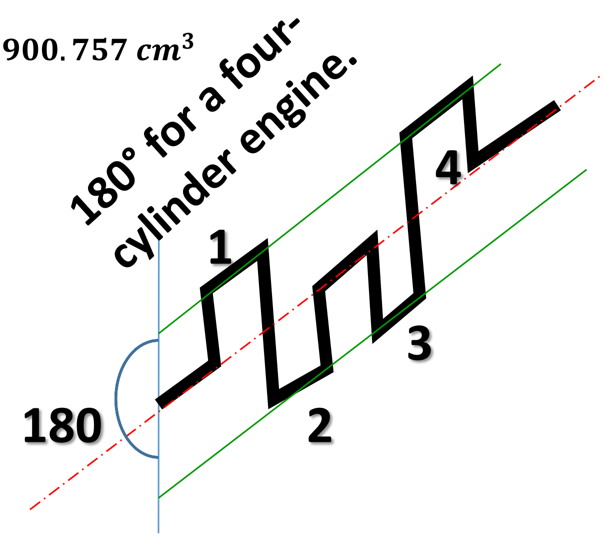
Power – Expansion duration =(180 + AI +EOA)/2 = (180 +10 + (-32))/2 =**79 degree of Camshaft**

Exhaust duration = (180 –EOA + ECD)/2 = (180 –(-32) + 5)/2 = **108.5 degree of Camshaft**

2. The total cylinder displacement is given by :

$$\boldsymbol{V_T = n.\frac{\pi.D^2}{4}.s = 4.\frac{\pi.6.4^2}{4}.7 = 900.757\ cm^3}$$

3. The crankshaft diagram



4. The engine firing order : 1, 3, 4, 2

	C #1	C #2	C #3	C #4
0 180°	PE	Exh	Comp	Int
180° 360°	Exh	Int	PE	Comp
360° 540°	Int	Comp	Exh	PE
540° 720°	Comp	PE	Int	Exh

PE: Explosion-expansion, Exh: Exhaust, Comp: Compression, Int: Intake

5. The compression ratio

$$CR = \varepsilon = \frac{V_d}{V_0} + 1 = \frac{\frac{\pi . D^2}{4} . s}{V_0} = \frac{\frac{\pi * 6.4^2}{4} * 7}{13.25} = 17.9954 \approx 18$$

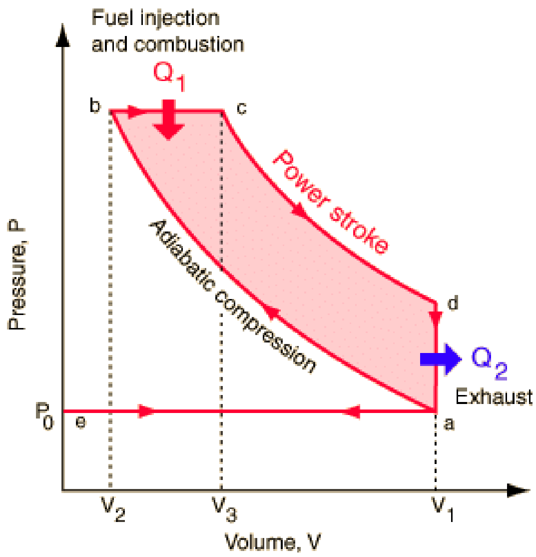
6. Brake power (effective power)

$$P_e = C.\omega = C.\frac{2.\pi.N}{60} = 22 * \frac{2 * \pi * 1500}{60} = 3455.75\ W$$

7. The camshaft rotation speed

$$N_{Camshaft} = \frac{N_{crank}}{2} = \frac{1500}{2} = 750\ rpm$$

8. Diagram (P-V) , CR = 18 so the engine is an internal compression engine



Calculation of pressure and temperature at each state of the cycle

P_1 = 1 bar ; T_1 = 25 + 273.15 = 298.15 K

State (1-2) Adiabatic compression

$$P_1 {V_1}^{\gamma} = P_2 {V_2}^{\gamma} \Rightarrow P_2 = P_1 . CR^{\gamma} = 1 * 18^{1.4} = 57.198\ bar$$

$$T_1 {V_1}^{\gamma-1} = T_2 {V_2}^{\gamma-1} \Rightarrow T_2 = T_1 . CR^{\gamma-1} = 298.15 * 18^{0.4} = 947.42\ K$$

State (2-3) Constant pressure evolution

$$P_3 = P_2 \quad = 57.198\ bar$$

$$Q_{in} = C_P (T_3 - T_2) \Rightarrow T_3 = \frac{Q_{in}}{C_p} + T_2 = \frac{600}{1} + 947.42 = 1547.42\ K$$

State (3-4) Adiabatic evolution

$$P_3 {V_3}^{\gamma} = P_4 {V_4}^{\gamma} \Rightarrow P_4 = P_3 . \left(\frac{V_3}{V_1}\right)^{\gamma} = P_3 . \left(\frac{\frac{T_3}{T_2}}{CR}\right)^{\gamma} = 57.198 * \left(\frac{\frac{1547.42}{947.42}}{18}\right)^{1.4} = 1{,}98\ bar$$

$$T_3 {V_3}^{\gamma-1} = T_4 {V_4}^{\gamma-1} \Rightarrow T_4 = T_3 . \left(\frac{V_3}{V_1}\right)^{\gamma-1} = T_3 . \left(\frac{\frac{T_3}{T_2}}{CR}\right)^{\gamma-1} = 1547.42 * \left(\frac{\frac{1547.42}{947.42}}{18}\right)^{0.4}$$

$$= 592.55\ K$$

9. Thermal efficiency of the engine

$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{C_v (T_4 - T_1)}{Q_{in}} = 1 - \frac{0.715 * (592.55 - 298.15)}{600} = 0.6491$$

$$= 64.91\%$$

10. Comparison with the Otto cycle (gasoline engine)

$$\eta_{th} = 1 - \frac{1}{CR^{\gamma-1}} = 1 - CR^{1-\gamma} = 1 - 18^{-0.4} = 0.6853 = 68.53\%$$

$\eta_{Otto}\ or\ \eta_{Gasoline} > \eta_{Diesel}$ **(For the same compression ratio)**

Chapter IV: Dynamic study of an ICE

CHAPTER: DYNAMICS OF RECIPROCATING ENGINES

4.1. Introduction

The dynamics of reciprocating engines deals with the study of forces, torques, accelerations, vibrations, and balancing associated with the moving parts of internal combustion engines. These engines transform thermal energy into mechanical work through the reciprocating motion of pistons inside cylinders.

4.2. Instantaneous Piston Position (cm)

Consider the piston, connecting rod and crank system as shown in figure (4-1). We are asked to find the instantaneous position of the piston (x) as a function of the crankshaft angle β, the connecting rod-crank ratio (λ= r / L), with: r is the crank radius and L is length of connection rod

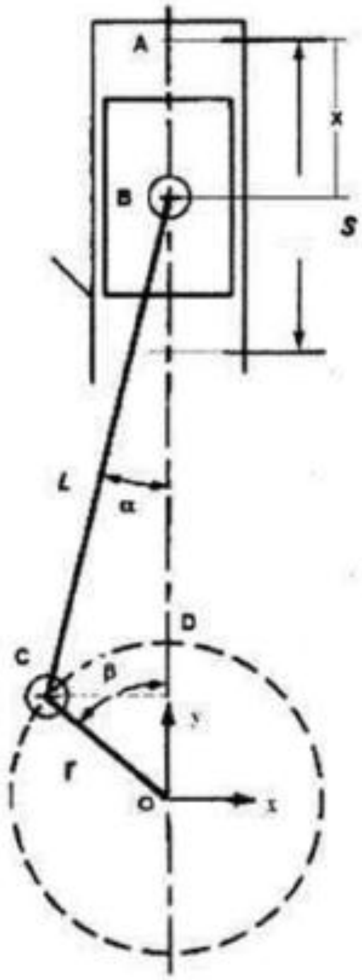
$$\boldsymbol{x} = \overline{\boldsymbol{OA}} - \overline{\boldsymbol{OB}} = (\boldsymbol{r} + \boldsymbol{L}) - \overline{\boldsymbol{OB}}$$

$$\overline{OB} = \overline{OD} + \overline{DB}$$

With $\overline{\boldsymbol{OD}} = \boldsymbol{r}.\cos\boldsymbol{\beta}$ and $\overline{\boldsymbol{DB}} = \boldsymbol{L}.\cos\boldsymbol{\alpha}$

Hence:

$$\boldsymbol{x}(\boldsymbol{\beta}) = (\boldsymbol{r} + \boldsymbol{L}) - \boldsymbol{r}.\cos\boldsymbol{\beta} - \boldsymbol{L}.\cos\boldsymbol{\alpha} \quad (4.1)$$



Moreover:

Figure 4.1: piston-connecting rod-crank system

$$Sin\,\beta = \frac{\overline{CD}}{r} \text{ and } Sin\,\propto = \frac{\overline{CD}}{L} \Rightarrow r.Sin\,\beta = L.Sin\,\propto \Rightarrow Sin\propto = \frac{r}{L}Sin\,\beta$$

$$\Rightarrow Sin\,\propto = \lambda.Sin\,\beta$$

On the other hand we have : $sin^2\alpha + cos^2\alpha = 1$; $sin\,\alpha = \sqrt{1 - cos^2\alpha}$

$$1 - cos^2\alpha = \lambda^2 sin^2\beta \Rightarrow \cos\alpha = \sqrt{1 - \lambda^2 sin^2\beta} \quad (4.2)$$

Substituting (2) into (1):

$$x = r - r.\cos\beta + L - L.\cos\alpha = r.(1-\cos\beta) + L.(1-\cos\alpha) \quad (4.3)$$

$$x(\beta) = r\left[(1-\cos\beta) + \frac{L}{r}\left(1 - \sqrt{1-\lambda^2 sin^2\beta}\right)\right] \quad (4.4)$$

Given that the Taylor series expansion of $(1-f)^m$ in the neighborhood of zero is given by:

$$(1-f)^m = 1 - \frac{m.f}{1!} + \frac{m.(m-1).f^2}{2!} + o(f^3) \quad (4.5)$$

If we set: m = 1/2 and $f = \lambda^2.sin^2\beta$, we can therefore write:

$$\sqrt{1-\lambda^2 sin^2\beta} = 1 - \frac{\frac{1}{2}\lambda^2.sin^2\beta}{1!} + \frac{\frac{1}{2}.\left(\frac{1}{2}-1\right).\lambda^4.sin^4\beta}{2!} + o(f^3) \quad (4.6)$$

The term $\frac{\frac{1}{2}.\left(\frac{1}{2}-1\right).\lambda^4.sin^4\beta}{2!} \rightarrow 0 \; because \; f^2 \rightarrow 0 \; when \quad f \rightarrow 0$

$$\sqrt{1-\lambda^2 sin^2\beta} = 1 - \frac{1}{2}\lambda^2.sin^2\beta \quad (4.7)$$

Substituting (7) into (4):

$$x(\beta) = r\left[(1-\cos\beta) + \frac{L}{r}\left(1 - 1 + \frac{1}{2}\lambda^2.sin^2\beta\right)\right] \quad (4.8)$$

$$x(\beta) = r\left[(1-\cos\beta) + \frac{1}{\lambda}\left(\frac{1}{2}\lambda^2.sin^2\beta\right)\right] \quad (4.9)$$

$$\boldsymbol{x(\beta) = r\left[1-\cos\beta + \frac{\lambda}{2}.sin^2\beta\right]} \quad (4.10)$$

Since $sin^2\beta = \frac{1}{2}(1-\cos 2\beta)$

$$\boldsymbol{x(\beta) = r\left[1-\cos\beta + \frac{\lambda}{4}.(1-\cos 2\beta)\right]} \quad (4.11)$$

4.3. Instantaneous piston speed (cm/s)

The instantaneous piston speed is the speed of a piston at a specific crank angle during the engine cycle. It varies continuously because the piston motion is driven by the rotating crankshaft and connecting rod.

For a crank-slider mechanism, piston velocity is approximately:

$$\dot{x}(\beta) = \frac{dx(\beta)}{dt} = \frac{d\beta}{dt}\frac{dx(\beta)}{d\beta} = \dot{\beta}.\frac{dx(\beta)}{d\beta} \tag{4.12}$$

Where:

$\dot{\beta} = \omega = \frac{2.\pi.N}{60}$, Is the angular velocity in radians per second.

$$\boldsymbol{\dot{x}(\beta) = r.\omega\left[\sin\beta + \frac{\lambda}{2}(\sin 2\beta)\right]} \tag{4.13}$$

- $\dot{x}(\beta)$ = instantaneous piston speed
- r = crank radius
- ω = angular speed of crankshaft
- β = crank angle from top dead center
- λ = the connecting rod-crank ratio

Key points:

- Piston speed is zero at top dead center (TDC) and bottom dead center (BDC).
- Maximum piston speed occurs roughly around 70 [deg.] -to- 75 [deg.] of crank angle after TDC.
- The piston does not move with simple harmonic motion because of the connecting rod geometry.

The mean piston speed (often used in engine design) is different:

$$\overline{v} = \frac{2.s.N}{60} \tag{4.14}$$

Where:

- $\overline{v}$ = mean piston speed
- s = stroke length
- N = crankshaft speed (rpm)

Example

An internal combustion engine of : Stroke =100 cm ; Crank radius: r= 50 mm ; Connecting rod length: L=0.20 m ; Engine speed: N=3000 rpm ; Crank angle: 60°

1. Give the instantaneous piston speed of this engine.
2. Calculate the mean piston speed.

Solution:

1. $\omega = \frac{2.\pi.N}{60} = \frac{2\times\pi\times3000}{60} = 314.16 \text{ rad/s}$
   - r = 0.05
   - L =0.20
   - θ = 60°
   - ω= 314.16

$$\dot{x}(\beta) = 0.05 \times 314.16 \left[\sin 60 + \frac{0.05}{2 \times 0.2}(\sin 120)\right] = 15.3\ m/s$$

2. Mean piston speed

$$\overline{v} = \frac{2 \times 0.1 \times 3000}{60} = 10m/s$$

4.4. Instantaneous piston acceleration (cm²/s)

The instantaneous piston acceleration is the acceleration of the piston at a specific crank angle during the motion of a reciprocating engine mechanism. Unlike mean acceleration, it continuously changes throughout the engine cycle because the piston motion depends on both the crank rotation and the connecting rod geometry. The acceleration is maximum near the top dead center (TDC) and bottom dead center (BDC), where the piston changes direction. For a slider–crank mechanism, the approximate instantaneous piston acceleration is given by the equation:

$$\ddot{x}(\beta) = \frac{dv(\beta)}{dt} = \frac{d\beta}{dt}\frac{dv(\beta)}{d\beta} = \dot{\beta}.\frac{dv(\beta)}{d\beta} \tag{4.15}$$

$$\boldsymbol{\ddot{x}(\beta) = r.\omega^2[\cos\beta + \lambda\,(\cos 2\beta)]} \tag{4.16}$$

If the crank radius and connecting rod length are expressed in centimeters, the acceleration unit becomes cm/s². This parameter is very important in engine design because high piston acceleration produces large inertia forces, which affect vibration, stresses, wear, and the overall mechanical performance of the engine.

4.5. Instantaneous combustion chamber volume

This is the graphical representation of the actual cycle through the average pressure curve in cylinder P_{cyl} as a function of the crankshaft angle β, or the average pressure diagram in cylinder P_{cyl} as a function of the volume $V(\beta)$. D, is the diameter of the cylinder

For each value of the crankshaft angle β, we can measure or predict by calculation P_{cyl} then we derive the position of the piston in the cylinder using the following relationship:

$$x(\beta) = r\left[1 - \cos\beta + \frac{\lambda}{4}.(1 - \cos 2\beta)\right]$$

The volume $V(\beta)$ is the product of the position $x(\beta)$ and V_{cyl}, i.e.:

$$\boldsymbol{V(\beta) = r\left[1 - \cos\beta + \frac{\lambda}{4}.(1 - \cos 2\beta)\right].\frac{\pi.D^2}{4}} \tag{4.17}$$

The total volume of combustion chamber is given by equation (4.18)

$$\boldsymbol{V_T(\beta) = V_c + V(\beta) = V_c + r\left[1 - \cos\beta + \frac{\lambda}{4}.(1 - \cos 2\beta)\right].\frac{\pi.D^2}{4}} \tag{4.18}$$

4.6. Dynamics of the connecting rod-crank mechanism

In physics, the action of a force about an axis of rotation is called a moment. Engine engineers use the term "torque" for the same quantity. At any given moment, the value of the torque applied to the crankshaft is:

$$\mathbf{C\ (N.m)\ =\ r\ (m)\ x\ F_T\ (N)} \tag{4.19}$$

Engine torque is often expressed in m.daN or m.kg. F_T is the decomposition of the action of the piston on the connecting rod (F1).

The work produced by the torque is:

$$\mathbf{W\ (Joule)\ =\ C\ (N.m)\ x\ (radian)} \tag{4.20}$$

The force F1 depends on:

- The value of the force generated by the pressure on the piston;
- The angle (α).

Mechanical transformation of gas thrust into engine torque via a connecting rod/crank system, is presented in figure 4.2 :

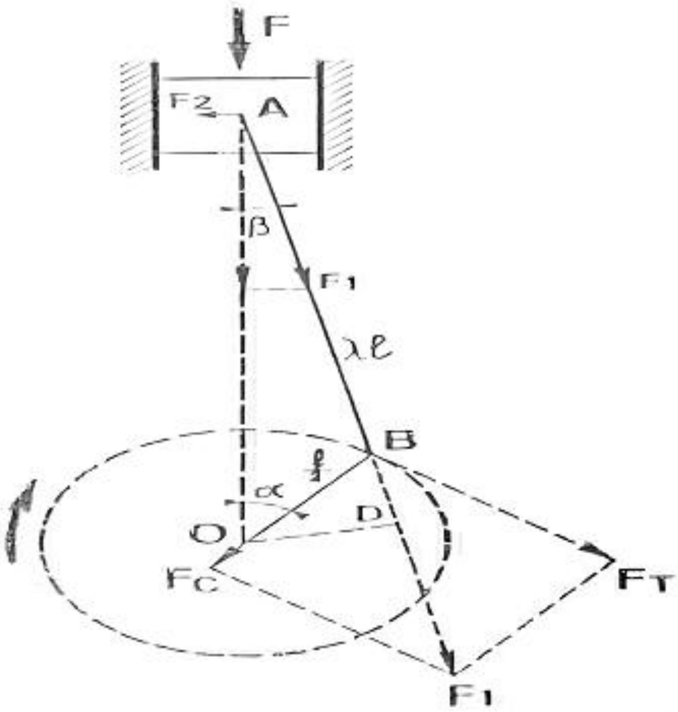


Figure 4.2. *Forces acting on the connecting rod-crank mechanism*

The pressure force: $\boldsymbol{F}(\propto) = \left[\boldsymbol{P}(\propto)\right) - \boldsymbol{P}_{\textbf{crankcase}}].\frac{\boldsymbol{\pi}.\boldsymbol{D}^2}{\boldsymbol{4}}$ (4.21)

According to the figure above, we can calculate the F_1 component as follows:

$$\boldsymbol{F_1} = \frac{\boldsymbol{F}}{\cos\boldsymbol{\beta}} \tag{4.22}$$

The F_T component that acts on the crank to produce the motor torque can be calculated from the following relationship:

$$\boldsymbol{F_T} = \boldsymbol{F}(\boldsymbol{\alpha}).\frac{\boldsymbol{cos}\left[\frac{\boldsymbol{\pi}}{2} - (\boldsymbol{\alpha}+\boldsymbol{\beta})\right]}{\cos\boldsymbol{\beta}} = \boldsymbol{F}(\boldsymbol{\alpha}).\frac{\boldsymbol{sin}(\boldsymbol{\alpha}+\boldsymbol{\beta})}{\cos\boldsymbol{\beta}} \tag{4.23}$$

Where :

$$\cos\beta = \sqrt{1-\lambda^2.sin^2\alpha}$$

The motor torque C (N.m) is calculated using the following equation:

$$C(\alpha) = R.F_T = \boldsymbol{R}.\boldsymbol{F}(\boldsymbol{\alpha}).\frac{\boldsymbol{sin}(\boldsymbol{\alpha}+\boldsymbol{\beta})}{\cos\boldsymbol{\beta}} = \boldsymbol{R}.\boldsymbol{F}(\boldsymbol{\alpha}).\frac{\boldsymbol{sin}\left[\boldsymbol{\alpha}+\boldsymbol{arccos}\left(\sqrt{\boldsymbol{1}-\boldsymbol{\lambda}^2.\boldsymbol{sin}^2\boldsymbol{\alpha}}\right)\right]}{\sqrt{\boldsymbol{1}-\boldsymbol{\lambda}^2.\boldsymbol{sin}^2\boldsymbol{\alpha}}} \tag{2.24}$$

Where R (meter) = radius of the crank ; F_T (Newton)

Intake phase

The resulting force F (pressure force) opposes the piston's movement (crankcase pressure > cylinder pressure). The torque required to move the piston downward is resistive.

If the intake pressure is lower (e.g., with the throttle closed or operating at high altitude), the resistive torque will be greater.

Compression Phase

The piston has changed direction of movement, but the force generated by the pressure in the cylinder has also reversed direction. The resulting torque is therefore still resistive, and its instantaneous value depends on:

- The position of the connecting rod at time t;

- The value of the instantaneous pressure in the cylinder.

If the mass of gas admitted during the intake phase is small, the resistive torque is less significant.

Expansion phase

This time, the force and the displacement are in the same direction; we have a driving torque. If there is no combustion (injection cut-off during deceleration, for example), the driving torque is the symmetrical equivalent of the torque generated by compression (minus heat losses and friction).

Exhaust Phase

The force generated by the pressure of the exhaust gases opposes the direction of piston movement. Therefore, we have a resisting torque. If the pressure outside the engine increases (atmospheric pressure), the resisting work of the exhaust will be greater.

The work done during the intake and exhaust phases is called "pumping work."

4.7. Exercises

Exercise 4.1 :

For the connecting rod-crank mechanism of an internal combustion engine, plot the following representative curves:

• The connecting rod's obliquity as a function of the crank's rotation angle:

- $\alpha = \alpha(\beta)$ For $0 \leq \beta \leq 2.\pi$

• The connecting rod's angular velocity as a function of the crank's rotation angle:

- $\dot{\alpha} = \dot{\alpha}(\beta)$ For $0 \leq \beta \leq 2.\pi$

• The connecting rod's angular acceleration as a function of the crank's rotation angle:

- $\ddot{\alpha} = \ddot{\alpha}(\beta)$ For $0 \leq \beta \leq 2.\pi$

Given: $\lambda = 0.25$ and N = 2000 rpm

Exercise 4.2

A spark-ignition automobile engine with the following characteristics:

- Bore D = 91.7 mm
- Piston stroke s = 81.6 mm
- Connecting rod center distance L = 137 mm
- Crankshaft radius R = s/2 = 40.8 mm

1- Perform the kinematic calculations of this mechanism for two different engine speeds:

• For N = 1000 rpm

• For N = 6000 rpm

a- Crankshaft kinematics

b- Piston kinematics

Exercise 4.3

An internal combustion engine is characterized by the following parameters:

D = 92 mm, CP = 90 mm, L = 140 mm, N = 2000 rpm, RC = 10 .

1- Calculate the average piston speed.

2- When the piston is at mid-stroke, determine:

a- The volume occupied by the gases?

b- The heat exchange surface area between the gases and the walls?

c- The piston speed and acceleration?

d- The engine torque, if at this position the gas pressure is 10 bar?

Exercise 4.4

For the connecting rod-crank mechanism of an internal combustion engine, we are asked to find the expression for the instantaneous volume as a function of the crankshaft rotation angle during the piston's downward movement from top dead center (TDC) to bottom dead center (BDC).

Given: Bore D, Piston stroke, Ratio of crankshaft radius to connecting rod length, Compression ratio

4.8. Solution of exercises

Solution of exercise 4.1

1- Connecting rod obliquity as a function of the crankshaft rotation angle:

For the connecting rod-crank mechanism of an internal combustion engine (see Figure 4.1), the relationship between the connecting rod obliquity α and the crankshaft rotation angle β is:

$L.sin(\theta)(\pi - \alpha) = R.\sin(\beta)$, therefore: $\sin(\pi - \alpha) = \lambda.sin\beta$ sin(θ). Or: $\pi - \alpha = \arcsin[\lambda.\sin\beta]$ Finally:

$$\alpha = \pi - \arcsin[\lambda.\sin\beta] = \alpha(\beta)$$

- If $\beta = 0$ then $\alpha = \pi = 180°$
- If $\beta = \frac{\pi}{2}$ then $\alpha = \pi - \arcsin\left[0.25 \times sin\frac{\pi}{2}\right] = \pi - \arcsin[0.25] = 165.52°$

- If $\beta = \pi$ then $\alpha = \pi = 180°$
- If $\beta = \frac{3.\pi}{2}$ then $\alpha = \pi - \arcsin\left[0.25 \times sin\frac{3.\pi}{2}\right] = \pi + \arcsin[0.25] = 194.47°$
- If $\beta = 2.\pi$ then $\alpha = \pi = 180°$

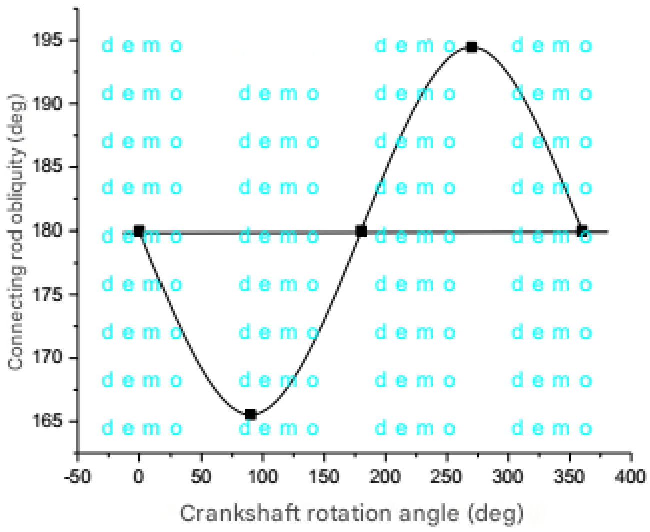


Figure 4.3. *Variation of the connecting rod inclination as a function of the crankshaft angle*

2- The angular velocity of the connecting rod as a function of the crankshaft rotation angle:

We differentiate the fundamental kinematic relationship with respect to time:

$$\frac{d(R.\sin\beta)}{dt} = \frac{d(L.\sin\alpha)}{dt}$$

or :

$$R.\dot{\beta}.\cos\beta = L.\dot{\alpha}.\cos\alpha \quad \text{with :}$$

$$\dot{\alpha} = \frac{R.\dot{\beta}.\cos\beta}{L.\cos\alpha} = \frac{\lambda.\dot{\beta}.\cos\beta}{\cos\alpha} = -\dot{\beta}.\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.\sin^2\beta}} = \frac{\pi.N}{30}.\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.\sin^2\beta}} = \dot{\alpha}(\beta)$$

- If $\beta = 0$ then $\dot{\alpha} = -52.33\ rad/sec$
- If $\beta = \frac{\pi}{2}$ then $\dot{\alpha} = 0\ rad/sec$
- If $\beta = \pi$ then $\dot{\alpha} = 52.33\ rad/sec$
- If $\beta = \frac{3.\pi}{2}$ then $\dot{\alpha} = 0\ rad/sec$
- If $\beta = 2.\pi$ then $\dot{\alpha} = -52.33\ rad/sec$

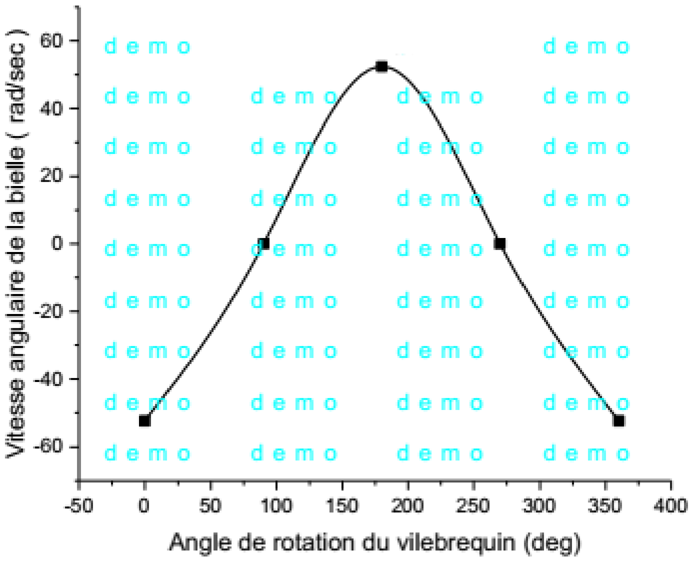


Figure 4.4. *Variation of the connecting rod's angular velocity as a function of the crankshaft angle*

3. Angular acceleration of the connecting rod as a function of the crankshaft rotation angle:

We differentiate the fundamental kinematic relationship with respect to time a second time:

$$\frac{d(R.\dot{\beta}.cos\beta)}{dt} = \frac{d(L.\dot{\alpha}.\cos\alpha)}{dt}$$

$$R.\ddot{\beta}.\cos\alpha - R.\dot{\beta}\,.\cos\alpha = L.\ddot{\alpha}.cos\alpha\ (with\ \ddot{\beta} = 0\ because\ it\ is\ constant)$$

where :

$$R.\sin\beta = L.\sin\alpha$$

So $\ddot{\alpha} = \frac{L.\dot{\alpha}^2.\sin\alpha - L.\dot{\beta}^2.\sin\alpha}{L.\cos\alpha}$

Finelly :

$$\ddot{\alpha} = (\dot{\alpha}^2 - \dot{\beta}^2).tg\,\alpha = -\left[\frac{\lambda^2.\dot{\beta}^2.cos^2\beta}{(1-\lambda^2.sin^2\beta)} - \dot{\beta}^2\right].\frac{\lambda.\sin\beta}{\sqrt{1-\lambda^2.sin^2\beta}} = \dot{\beta}^2.\frac{(1-\lambda^2).\lambda.\sin\beta}{(1-\lambda^2.sin^2\beta)^{\frac{3}{2}}}$$

- If $\beta = 0$ then $\ddot{\alpha} = 0\ rad/sec^2$
- If $\beta = \frac{\pi}{2}$ then $\ddot{\alpha} = 45257.56\ rad/sec^2$
- If $\beta = \pi$ then $\ddot{\alpha} = 0\ rad/sec^2$
- If $\beta = \frac{3.\pi}{2}$ then $\ddot{\alpha} = -45257.56\ rad/sec^2$
- If $\beta = 2.\pi$ then $\ddot{\alpha} = 0\ rad/sec^2$

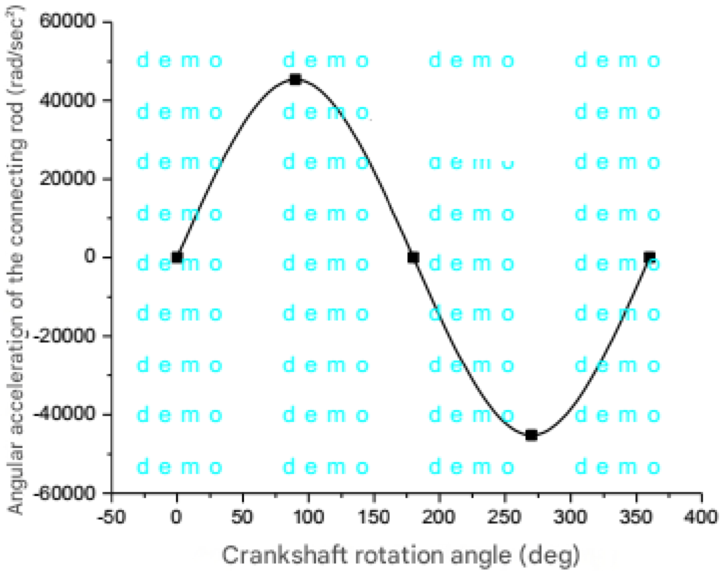


Figure 4.5. *Variation of the connecting rod's angular acceleration as a function of the crankshaft angle*

Solution of exercise 4.2

1- Kinematics of the crankpin:

• The center A of the crankpin moves on a circle with center O and radius R = s/2 =40.8mm

• The velocity of point A is constant in magnitude and equal to $R.\dot{\beta}$

- For N = 1000 rpm, we have $\vec{V}\left(A,\frac{1}{0}\right) = -4.27\,\frac{m}{sec}.\overrightarrow{X_1}$

- For N = 6000 rpm, we have $\vec{V}\left(A,\frac{1}{0}\right) = -25.63\,\frac{m}{sec}.\overrightarrow{X_1}$

The acceleration of point A is centripetal and constant in magnitude and equal to $R.\dot{\beta}^2$

- For N = 1000 rpm, we have $\vec{\gamma}\left(A,\frac{1}{0}\right) = -447\,\frac{m}{sec^2}.\overrightarrow{Y_1}$

- For N = 6000 rpm, we have $\vec{\gamma}\left(A,\frac{1}{0}\right) = -16107\,\frac{m}{sec^2}.\overrightarrow{Y_1}$

2- Piston kinematics:

Piston position:

$$\overrightarrow{OB} = d.\overrightarrow{Y_0} = (R.\cos\beta - L.\cos\alpha).\overrightarrow{Y_0} = (R.\cos\beta - L.\sqrt{1-\lambda^2.sin^2\beta}).\overrightarrow{Y_0}$$

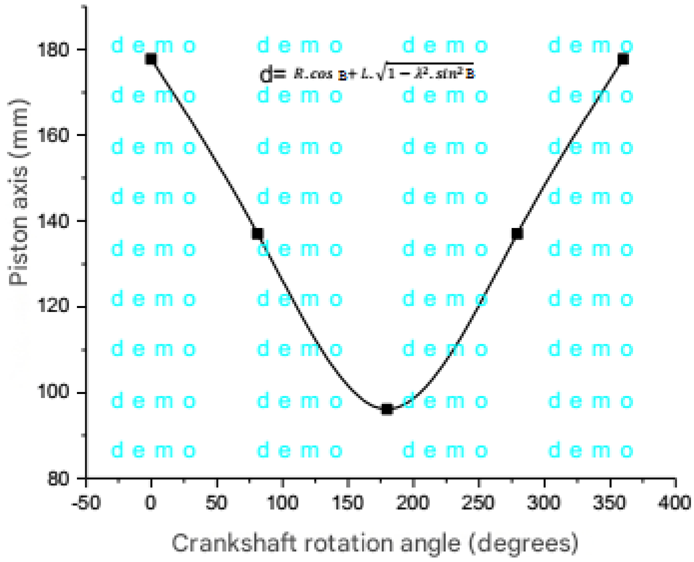


Figure 4.6. *Piston axis as a function of crankshaft rotation angle*

If $\beta = 0$, point B is at TDC (Top Dead Center), then $\cos\beta = 1$ and $cos \propto = -1$.

"d" takes its maximum value.

$$d_{max} = R + L = 177.8\, mm$$

• If $\beta = \pi$, point B is at BDC (Bottom Dead Center), then $cos\ \beta = -1$ and $cos \propto = -1$.

"d" takes its minimum value.

$$d_{max} = -R + L = 96.2\, mm$$

Mid-stroke angle:

The piston is at mid-stroke when $d = \frac{d_{max} + d_{min}}{2} = 137\ mm = L$

This corresponds to $\beta = 81°\ and\ \beta = 297°$

Piston speed:

$$\vec{V}\left(B, \frac{2}{0}\right) = R.(\dot{\propto} - \dot{\beta}).\sin\beta.\overrightarrow{Y_0} = R.\left(-\frac{\pi.N}{30}.\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.sin^2\beta)}} + \frac{\pi.N}{30}\right).\sin\beta.\overrightarrow{Y_0}$$

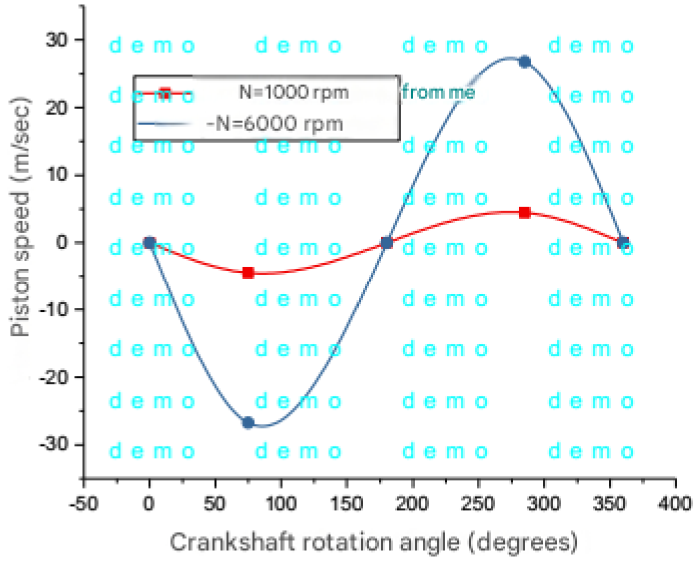


Figure 4.7. *Piston speed as a function of crankshaft rotation angle for 2 different engine speeds*

If $\beta = 0$ and $= \pi$, the piston is at TDC and BDC respectively, the velocity is zero

If $\beta = \frac{\pi}{2}$ and $\beta = \frac{3.\pi}{2}$, the piston velocity is $\vec{V}\left(B, \frac{2}{0}\right) = \pm R.\dot{\beta}.\overrightarrow{Y_0}$

- For N = 1000 rpm, we have $\vec{V}\left(B, \frac{2}{0}\right) = \pm 4.27\,\frac{m}{sec}$

- For N = 6000 rpm, we have $\vec{V}\left(B, \frac{2}{0}\right) = \pm 25.63\,\frac{m}{sec}$

The piston reaches its maximum speed at $\beta = 75°$ and $\beta = 285°$

- For N = 1000 rpm, we have $\overrightarrow{V_{max}}\left(B, \frac{2}{0}\right) = \pm 4.46\,\frac{m}{sec}$

- For N = 6000 rpm, we have $\overrightarrow{V_{max}}\left(B, \frac{2}{0}\right) = \pm 26.75\,\frac{m}{sec}$

We can note that the approximation $tg\ \beta = \frac{1}{\lambda}$ is good since it gives $\beta = 73.416°$, whereas the exact formula:

$$sin^6\,\beta - \frac{1}{\lambda^2}.sin^4\,\beta - \frac{1}{\lambda^4}.sin^2\,\beta + \frac{1}{\lambda^4} = 0$$

Gives $\beta = 74.621°$

Average piston speed:

The average piston speed is equal to $\frac{Stroke.N}{30}$

- For N = 1000 rpm, we have $V_{moy} = 2.72\,\frac{m}{sec}$

- For N = 6000 rpm, we have $V_{moy} = 16.32\,\frac{m}{sec}$

Piston acceleration:

$$\vec{\gamma}\left(B,\frac{2}{0}\right) = \left[R.\dot{\beta}.\left(\dot{\alpha} - \dot{\beta}\right).\cos\beta + R.\ddot{\alpha}.\sin\beta\right].\overrightarrow{Y_0}$$

$$= \left[R.\dot{\beta}.\left(-\dot{\beta}.\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.sin^2\beta)}} - \dot{\beta}\right).\cos\beta + R.\dot{\beta}^2.\frac{(1-\lambda^2).\lambda.\sin\beta}{(1-\lambda^2.sin^2\beta)^{\frac{3}{2}}}.\sin\beta\right].\overrightarrow{Y_0}$$

$$= \left[R.\dot{\beta}^2.\left(\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.sin^2\beta)}} + 1\right).\cos\beta + \frac{(1-\lambda^2).\lambda.\sin\beta}{(1-\lambda^2.sin^2\beta)^{\frac{3}{2}}}.\sin\beta\right].\overrightarrow{Y_0}$$

Solution of exercise 3:

Initial data:

D = 92 mm, CP = 90 mm, L = 140 mm, N = 2000 rpm, CR = 10

1- Average piston speed:

$$V_{moy-piston} = \frac{S.N}{30} = \frac{90.10^{-3}.2000}{30} = 6\ m/sec$$

2- When the piston is at mid-stroke, then the ordinate of the piston d=L, i.e.: $\cos\beta = \frac{\lambda}{2}$

a- The volume of the cylinder occupied by the gases at mid-stroke:

$$V_{mi-stroke} = V_{clearance} + \frac{V_u}{2}$$

When: $CR = \frac{V_u+V_c}{V_c} = \frac{V_u}{V_c} + 1\ so\ V_c = \frac{V_u}{(CR-1)}$

$$V_{mi-stroke} = V_{clearance} + \frac{V_u}{2} = \frac{V_u}{(CR-1)} + \frac{V_u}{2} = V_u.\left[\frac{1}{(CR-1)} + \frac{1}{2}\right]$$

Where: $V_u = S.\frac{\pi.D^2}{4}.10^{-9} = 5.97\times10^{-4}\ m^3\ \ and\ V_{mi-stroke} = 6.63\times10^{-4}\ m^3$

b- Piston speed: since $\cos\beta = \frac{1}{\lambda} = 0.125$ then $\beta \backsimeq 83°$

Substituting this angle, converted to radians, into the expression for the instantaneous speed of the piston:

$$\vec{V}\left(B,\frac{2}{0}\right) = R.\left(-\frac{\pi.N}{30}.\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.sin^2\beta)}} + \frac{\pi.N}{30}\right).\sin\beta.\vec{Y_0} = \frac{\pi.N}{30}.R.\left(\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.sin^2\beta)}} + 1\right)$$

We find : $\vec{V}\left(B,\frac{2}{0}\right) = -9.716\ m/s$

c- The acceleration of the piston at mid-stroke is calculated in the same way by substituting the $\beta \backsimeq 83°$ in the expression below:

$$\vec{\gamma}\left(B,\frac{2}{0}\right) = \left[R.\dot{\beta}^2.\left(\frac{\lambda.\cos\beta}{\sqrt{1-\lambda^2.sin^2\beta)}}+1\right).\cos\beta + \frac{(1-\lambda^2).\lambda.\sin\beta}{(1-\lambda^2.sin^2\beta)^{\frac{3}{2}}}.\sin\beta\right].\overrightarrow{Y_0}$$

We find : $\vec{\gamma}\left(B,\frac{2}{0}\right) = 256.728\, m/s^2$

d- The motor torque developed by the motor, if at this position the gas pressure is *P(83°)= 10 bar.*

$$C(\beta) = R.F_T = R.F.\frac{\sin(\alpha+\beta)}{\sin\beta} = R.F.\frac{sin(\beta+\arccos(1-\lambda^2.sin^2\beta))}{\sqrt{1-\lambda^2.sin^2\beta}}$$

$$= \frac{S}{2}.10^{-3}.P(83°).10^5.\frac{\pi.D^2}{4}.10^{-6}.\frac{sin(83°+\arccos(1-\lambda^2.sin^2 83°))}{\sqrt{1-\lambda^2.sin^2 83°}}$$

N.A. $C(\beta) = \quad N.m$

Solution of exercise 4:

The diameter D of the cylinder is also called the bore. The piston's displacement is limited by two points: Top Dead Center (TDC) and Bottom Dead Center (BDC). When the piston is at TDC (respectively, BDC), the combustion chamber volume is at its minimum (respectively, maximum). The stroke represents the distance L traveled by the piston between these two reference points. Note that the ratio between the stroke and the connecting rod radius, r, is an invariant parameter of the engine; the following relationship is almost always respected: L =2.r

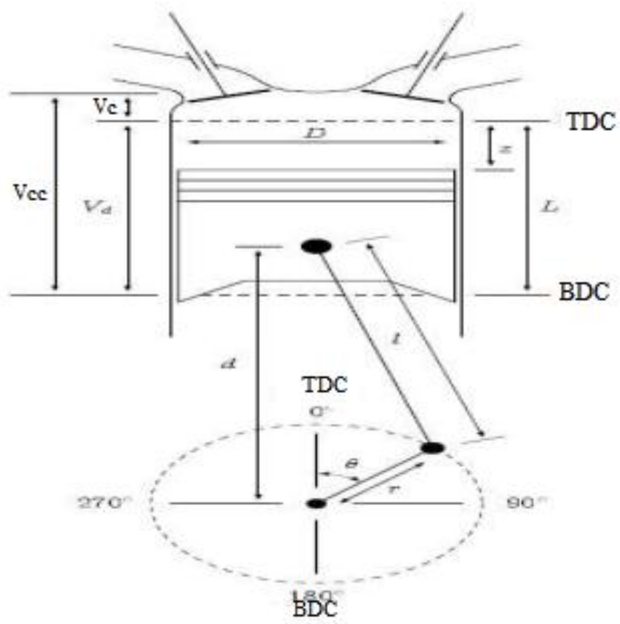


Figure 4.8. *Diagram for calculating the instantaneous volume occupied by gases*

The displaced volume (or unit displacement) V_d corresponds to the volume swept by the piston between TDC and BDC:

$$V_d = L.\frac{\pi.D^2}{4}$$

The total volume of the combustion chamber V_{cc} is equal to the sum of the displaced volume V_d and the clearance volume V_c

$$V_{cc} = V_d + V_c$$

The compression ratio: this corresponds to the ratio of the total volume to the dead volume:

$$CR = \frac{V_{cc}}{V_c} = \frac{V_d + V_c}{V_c} = 1 + \frac{V_d}{V_c}$$

The connecting rod-crank ratio: This is a ratio between the length of the connecting rod and the radius of the crank.

$$R_{cr-cc} = \frac{l}{r}$$

Instantaneous volume of the combustion chamber:

$$V_{cyl} = V_c + \frac{\pi.D^2}{4}(l + r - d)$$

Distance A: vertical position of the piston in the cylinder that separates the crankshaft axis from the piston axis.

$$d = r.\cos\beta + \sqrt{l^2 - r^2.sin^2\beta}$$

$$V_{cyl} = V_c + \frac{\pi.D^2}{4}(l + r - \left(r.\cos\beta + \sqrt{l^2 - r^2.sin^2\beta}\right))$$

$$V_{cyl} = V_c + \frac{\pi.D^2}{4}.r.(\frac{l}{r} + 1 - \left(\cos\beta + \sqrt{\frac{l^2}{r^2} - sin^2\beta}\right))$$

$$V_{cyl} = V_c + \frac{\pi.D^2}{4}.\frac{L}{2}(R_{cr-cc} + 1 - \cos\beta - \sqrt{{R_{cr-cc}}^2 - sin^2\beta})$$

$$V_{cyl} = V_c + \frac{V_d}{2}(R_{cr-cc} + 1 - \cos\beta - \sqrt{{R_{cr-cc}}^2 - sin^2\beta})$$

$$\frac{V_d}{V_c} = \frac{V_d + V_m - V_m}{V_m} = CR - 1$$

$$\frac{V_{cyl}}{V_c} = 1 + \frac{CR.1}{2}(R_{cr-cc} + 1 - \cos\beta - \sqrt{{R_{cr-cc}}^2 - sin^2\beta})$$

The relationship between the crankshaft angle β, the rotational speed ω, and time is written: $\beta = \omega.t$. The positions of top dead center and bottom dead center are expressed relative to the crankshaft angle; thus, $\beta_{TDC} = 0\ °CA$ and $\beta_{TDC} = 180\ °CA$, CA denotes the crankshaft angle. We represent the rotational speed of the crankshaft.

$$\beta = \omega.t = \frac{\pi.N}{30}.t$$

The heat exchange surface A: the combustion chamber with surface area A at any crank position β is given by:

$$A(\beta) = A_{ch} + A_p + \pi.D(l + r - d)$$

Where A_{ch} the cylinder head surface and /e the piston head surface.

$$A_{ch} \approx A_p = \frac{\pi.D^2}{4}$$

Substituting equation (of $_{\text{Ach}}$) into ($A(\beta)$), we find:

$$A(\beta) = 2.\frac{\pi.D^2}{4} + \frac{\pi.D.l}{2}\left(R_{cr-cc} + 1 - \cos\beta - \sqrt{R_{cr-cc}{}^2 - sin^2\beta}\right)$$

Average piston speed: Since the piston travels twice the distance between TDC and BDC for one crankshaft revolution, the average speed is expressed as:

$$\nabla_p = \frac{s.N}{30}$$

The instantaneous speed of the piston V_p is obtained by deriving expression (6):

$$V_p = \frac{dd}{dt} = \frac{dd}{d\beta}\frac{d\beta}{dt} = \frac{dd}{d\beta}.\omega = r.\sin\beta.\left[1 + \frac{\cos\beta}{\sqrt{R_{cr-cc}{}^2 - sin^2\beta}}\right].\omega$$

Chapter 5 Performance and characteristics of reciprocating engines

5.1. Mean Indicated Pressure (MIP)

The Mean Indicated Pressure of the cycle is the equivalent pressure, assumed constant, delimited by the compression and expansion strokes. Where S1 is the positive work done during the cycle and S2 is the sum of negative work done during the cycle. This value is calculated for a single cylinder as follows:

$$S_1 = W_{indicated} = \int PdV = MIP.V_{cyl} \tag{5.1}$$

$$W_i = MIP.\frac{\pi.D^2}{4} \tag{5.2}$$

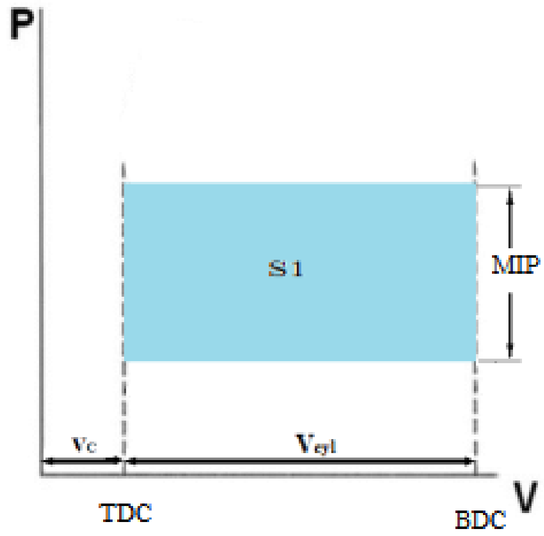


Figure 5.1. *Representation of the motor cycle with the MIP.*

The mean indicated pressure for a Z-cylinder engine is given by:

$$MIP = \sum_{i=1}^{Z} \frac{PMI}{Z} = \frac{\sum_{i=1}^{Z} W_i}{Z.V_{cyl}} \tag{5.3}$$

For a spark-ignition engine, the successive cycles are not identical, and the maximum pressures of the cycles are significantly different from one another.

However, for diesel engines, the successive cycles are identical, and we can thus write:

$$MIP = \frac{W_i}{V_{cyl}} \tag{5.4}$$

5.2. Mean indicated power (Pi)

The mean indicated power developed by a single cylinder is given by bay :

$$P_i = \frac{W_i}{\Delta t} \quad with\ \Delta t = \frac{60\times n_r}{N} \ ,\ P_i = \frac{W_i.N}{60\times n_R} = \frac{MIP.V_{cyl}.N}{60\times n_R} \qquad (5.5)$$

$$MIP = \frac{60\times P_i.n_R}{V_{cyl}.N} \qquad (5.6)$$

Where $n_R = \begin{cases} 2 \text{ if the engine is a four} - \text{stroke} \\ 1 \text{ if the engine is a two} - \text{stroke} \end{cases}$

The mean output indicated power by the engine is therefore:

$$P_i = \frac{MIP.V_{cyl}.N}{60\times n_R}.Z \qquad (5.7)$$

5.3. Effective Power (Pe)

The power developed during the cycle is called indicated power. However, this power is not entirely transmitted to the crankshaft due to irreversible factors. Indeed, due to heat losses through thermal transfer across the walls and through friction in the various mechanical components, the effective work developed at the crankshaft is less than the indicated work produced during the engine cycle. Similarly, the average effective power is related to the average effective pressure. This pressure produces the effective (useful) work at the crankshaft.

$$W_e = MEP.V_{cyl} \quad and\ \ P_e = \frac{W_e}{\Delta t} \quad thus\ P_e = \frac{W_e.N}{60\times n_R} = \frac{MEP.V_{cyl}.N}{60\times n_R} \qquad (5.8)$$

The effective power of the engine is:

$$P_e = \frac{MEP.V_{cyl}.N}{60\times n_R}.Z \qquad (5.9)$$

The effective power of the motor can be measured on a motor test bench from the algebraic product of the torque times the angular velocity measured at the motor shaft.

$$P_e = C.\omega = C.\frac{2.\pi.N}{60} \qquad (5.10)$$

The actual power is always less than the stated power, and we can note:

$$P_i = P_e + P_{\text{mechanical losses}} \tag{5.11}$$

$P_{\text{mechanical losses}}$ This is the power dissipated by mechanical losses due to friction between the engine components, the power expended to drive auxiliary parts.

The ratio between the actual power and the indicated power is the mechanical efficiency.

$$\eta_{mec} = \frac{P_e}{P_i}$$; We can write

$$\eta_{mec} = \frac{P_i - P_{mec}}{P_i} = 1 - \frac{P_{mec}}{P_i} \tag{5.12}$$

Mechanical losses are generally between 5 and 20%. Mechanical efficiency η_{mec} is in the range of 80 to 95%.

5.4. Specific Fuel Consumption (b_i)

To operate the engine, a certain quantity of fuel, mf, is required. Dividing mf by time gives the fuel flow rate, B, in kg/h or g/s.

The specific fuel consumption, bi, is the quantity of fuel required by the engine to develop a power output of 1 kWh (i.e., 1 kW for 1 hour).

$$b_i = \frac{B}{P_i}.3600 \tag{5.13}$$

If B is the fuel flow rate in g/s and P_i is the power output in kW, then bi is estimated in g/kWh. For a diesel engine, bi varies between 170 and 200 g/kWh.

5.5. Indicated efficiency (η_i)

This is the ratio of the indicated work done to the amount of heat released by combustion to perform that work. For a working time of one hour, the indicated efficiency can be calculated using the following relationship:

$$\eta_i = \frac{W_i}{LHV.m_f} \tag{5.14}$$

m_f is the quantity of fuel in kg, W_i is the work indicated in kJ, and LHV is the low heating value of the fuel (kJ/kg). However, more practically, it is easier to calculate the indicated efficiency from the fuel flow rate and the indicated power using the following relationship:

$$\eta_i = \frac{P_i}{LHV.B} \tag{5.15}$$

B is the fuel flow rate in kg/h, P_i is the power output in kW, and LHV is the calorific value of the fuel (kJ/kg). The stated efficiency represents the sum of heat losses through the cylinder liners, piston head, and cylinder head, plus the heat dissipated by the exhaust system and losses due to incomplete combustion and cooling.

The stated efficiency of the diesel engine is in the range of 40 to 50%.

5.6. Effective Specific Fuel Consumption (be)

Obtained by analogy with the stated specific fuel consumption. The actual fuel consumption must be higher than the stated consumption due to losses. Thus, to obtain 1 kW at the engine shaft, more fuel must be burned. The excess amount corresponds to the work and power lost through friction and heat transfer.

$$b_e = \frac{B}{P_e}.3600 \tag{5.16}$$

If B is the fuel flow rate in g/s and P_e is the effective power in kW, then be is estimated in g/kWh. For the diesel engine, be varies between 190 and 230 g/kWh.

5.7. Effective efficiency (η_e)

Effective efficiency is also called overall efficiency. It is mainly used to compare different engines. By analogy with the stated efficiency, the effective efficiency can be calculated using the following relationship:

$$\eta_e = \frac{W_e}{LHV.m_f} \tag{5.17}$$

m_f is the quantity of fuel in kg, *Wi* is the effective work in kJ, and LHV is the low heating value of the fuel (kJ/kg). However, more practically, it is easier to calculate the effective efficiency from the fuel flow rate and the power output using the following relationship:

$$\eta_e = \frac{P_e}{LHV.B}.3600 \tag{5.18}$$

$$\eta_e = \frac{\eta_{th}.P_i}{LHV.B}.3600 \tag{5.19}$$

$$\eta_e = \eta_{th} . \eta_i \tag{5.20}$$

B is the fuel flow rate in kg/s, P_e is the effective power in kW, and LHV is the low heating value of the fuel (kJ/kg). The effective efficiency of the diesel engine is in the range of 37 to 47%.

Given the last point, improving the overall efficiency of the engine implies increasing both thermal efficiency and indicated efficiency. This tends to reduce friction, shocks, and vibrations, and decrease the work required to drive auxiliary components.

1. To increase thermal efficiency, the following steps are necessary:

• Ensure proper lubrication.

• Use high-quality parts (correct fit, precision, and good finish).

• Use metals with a low coefficient of friction.

• Minimize the mass of moving parts.

• Eliminate imbalances on the crankshaft to ensure proper balance.

• Reduce horizontal forces on the parts.

2- To increase the stated efficiency, one must:

• Increase the volumetric compression ratio (within reasonable limits).

• Decrease pressure losses in the manifolds and intake and exhaust systems.

• Increase the stated power output by increasing the stated mean pressure, the number of cylinders, and the engine dimensions (bore and stroke).

5.8. Calculating the average piston speed (m/s)

The piston speed must not exceed a certain limit set by the manufacturer. This limit can be as high as 15 m/s. Excessive piston speed values increase mechanical losses and inertial forces. The average piston speed can be calculated using the following formula:

$$\overline{U_p} = \frac{S.N}{30} \tag{5.21}$$

5.9. Characteristic curves of an internal combustion engine

The static engine map illustrates all the operating points obtained on test benches specially designed for this purpose (Figure 5.2). The characteristic curves

1 – Engine and couplings.

2 – Electric brake.

3 – Fuel scale.

4 – HF acquisition card.

5 – Gas analysis bay.

6 – Engine-brake control.

7 – Processing unit (Trawley).

8 – Control and acquisition module.

9 – Control and command software.

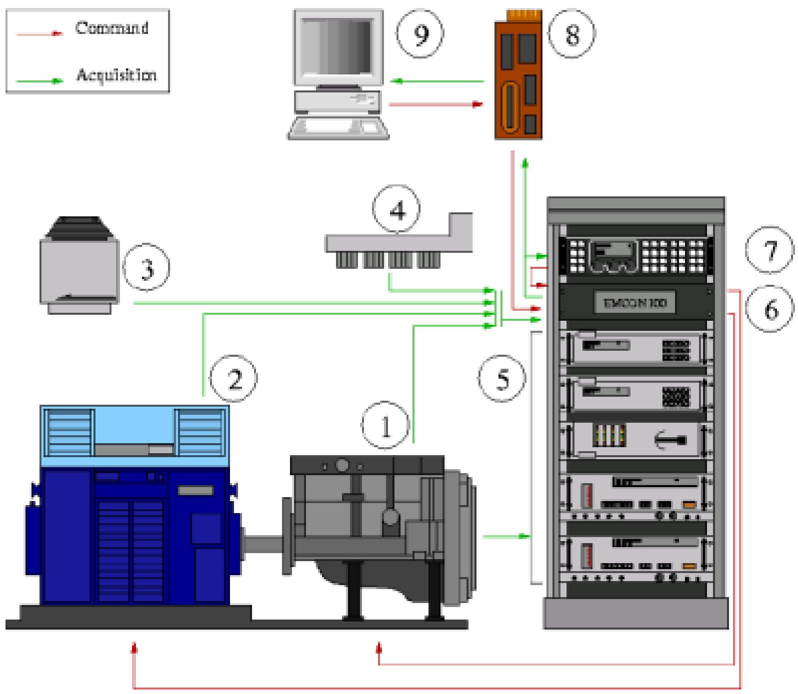


Figure 5.2. *Schematic representation of an engine test bench (AVL).*

Represent the evolution of the effective power, torque and specific consumption at full load as a function of engine rotation speed (figure 5.3).

5.10. Engine Torque (Nm)

Torque varies solely based on cylinder filling and is supposed to increase with increasing engine speed. However, at high engine speeds, cylinder filling is inefficient and the pressure at the end of the compression stroke decreases, causing engine torque to drop. The typical characteristic curve of engine torque increases slightly with engine speed, reaching a maximum and then dropping rapidly.

5.11. Power (kW)

Power is related to torque since P = C . ω. Power increases when torque increases. When torque begins to gradually decrease, the increase in speed compensates for this decrease, and power continues to increase. When torque begins to decrease drastically, power will have reached its maximum value because the increase in speed cannot compensate for this decrease, and power begins to drop rapidly.

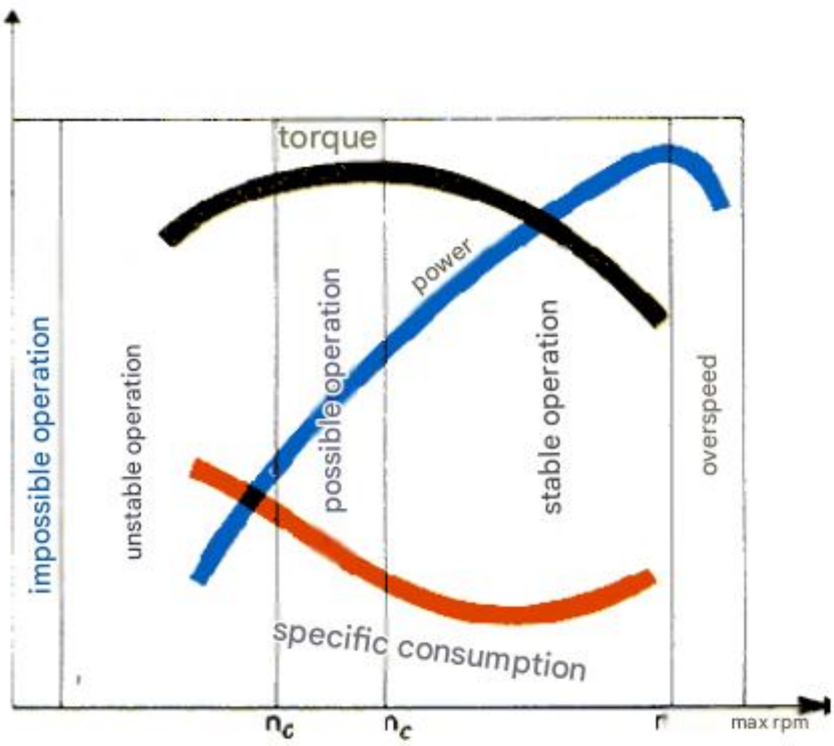


Figure 5.3. *Characteristic curves of a heat engine.*

5.12. Calculation of forces on the Piston-Connecting Rod Crank system

The forces exerted on the Piston-Connecting Rod Crank system come from the gas force applied to the piston head F_{Gas} as well as the inertial force of the Piston-Connecting Rod system (figure5.4)

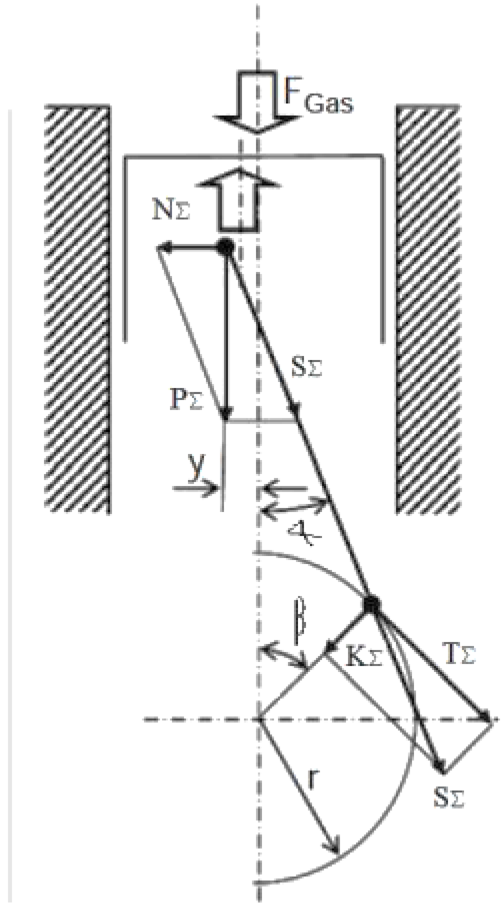


Figure 5.4. *Main forces on the Piston-Connecting Rod-Crank system.*

Knowing the diameter of the piston and the average pressure in the cylinder, this can be easily calculated using the following relationship:

$$F_{Gas} = \left(P_{cyl} - P_{atm}\right).\frac{\pi.D^2}{4} \tag{5.22}$$

The inertial forces can be calculated as follows:

$$F_i = -m.\ddot{x}(\beta) \tag{5.23}$$

$$F_i = m.r.\omega^2[\cos(\beta) + \lambda.\cos(2\beta)] \tag{5.24}$$

Where m is the moving mass and is given by:

$$m = m_{piston} + m_{1-C.R.} \tag{5.25}$$

$$m_{1-C.R.} = (0.2 \div 0.3)m_{C.R.} \tag{5.26}$$

$m_{1-C.R.}$ is the mass of the connecting rod relative to the mass of the connecting rod on the axis. The inertial pressure is then given by the relation:

$$P_i = \frac{F_i}{\left(\frac{\pi.D^2}{4}\right)} \tag{5.27}$$

The sum of the two forces mentioned above is called the overall force:

$$F_{\Sigma} = F_{Gas} \mp F_i \tag{5.28}$$

The summary pressure is obtained once the summary force is divided by the piston area:

$$P_{\Sigma} = \frac{F_{\Sigma}}{\left(\frac{\pi.D^2}{4}\right)} = \frac{F_{Gas} \mp F_i}{\left(\frac{\pi.D^2}{4}\right)} = P_{Gas} \mp P_i \tag{5.29}$$

The lateral force on the piston axis N_{Σ} tends to press the piston against the inner surface of the cylinder. It is given by:

$$N_{\Sigma} = P_{\Sigma} \tan \propto \tag{5.30}$$

The axial force on the connecting rod S_{Σ} is calculated using the following relationship:

$$S_{\Sigma} = \frac{F_{\Sigma}}{\cos\propto} \tag{5.31}$$

The axial force S_{Σ} is decomposed on the crank into two components: a tangential component T_{Σ} and a radial component K_{Σ} .

$$T_{\Sigma} = S_{\Sigma} \sin(\propto + \beta) = \frac{P_{\Sigma} \sin(\propto + \beta)}{\cos \alpha} \tag{5.32}$$

$$K_{\Sigma} = S_{\Sigma} \cos(\propto + \beta) = \frac{P_{\Sigma} \cos(\propto + \beta)}{\cos \alpha} \tag{5.33}$$

Bibliographical references

[1] John B. Heywood., Internal combustion engines fundamentals, Mc Graw Hill, 1988

[2] P. Arquès, « Conception et construction des moteurs alternatifs », Ellipse, 2000.

[3] J C. Guibet, « Carburants et moteurs », 1997.

[4] P. Arquès, Moteurs alternatifs à combustion interne, Masson édition, 1987.

[5] U.Y. Famin Gorban, A.I., Dobrovolsky V.V, Lukin A.I. et al., « Moteurs marins à combustion interne », Leningrad: Sudostrojenij, 1989, 344p.

[6] W. Diamant, « Moteurs à combustion interne », ECAM, 1984.

[7] Colin R. Ferguson , Allan T. Kirkpatrick. Internal Combustion Engines, Applied Thermo-sciences, Second Edition, John Wiley & Sons. Inc , 1986.

[8] Colin R. Ferguson, Allin T. Kirkpatrick . Internal combustion engines, applied thermo sciences, third edition. New York: Wiley, 2016.

[9] Kreiger R. B. and. Borman G. L. The computation of apparent heat release for internal combustion engines. ASME(66-WA, DGP-4), 1966.

[10] Borman G., Nishiwaki K. Internal-combustion engine heat transfer. Progress in. Energy and Combustion. Sciences 1987; 13: 1–46.

[11] G. Eichelberg, Some New Investigations on Old Combustion Engine Problems, Engineering 1939; 148: 446-463.

[12] Woschni G, Doll M, Spindler W. Simulation of the stationary and transient performance of small high speed car diesel engines. MTZ 1991; 52:468–77.

[13] Willard W. Pulkrabek. Engineering Fundamentals of the Internal Combustion Engine. Prentice Hall – Upper Saddle River, New Jersey 07458

[14] Jolivet D. « Le moteur diésel », Paris Ellipses ,1986.

[15] Benabbassi A. « Les moteurs à combustion interne, Introduction à la théorie », Alger, OPU. 2002.

[16] Constantine D. Rakopoulos , Evangelos G. Giakoumis. Diesel Engine Transient Operation, Principles of Operation and Simulation Analysis, Springer 2009

[17] R. A. Olsen, Analysis and Simulation of the Rate of Heat Release (ROHR) in Diesel Engines, Springer 2013.H.N. Gupta, Fundamentals of internal combustion engines, 2009.

[18] Brain E. Milton, Thermodynamics, combustion and engines, 3dr edition, School of mechanical and manufacturing engineering UNSW 2005.