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Thermal machines Cycles

Courses and application examples

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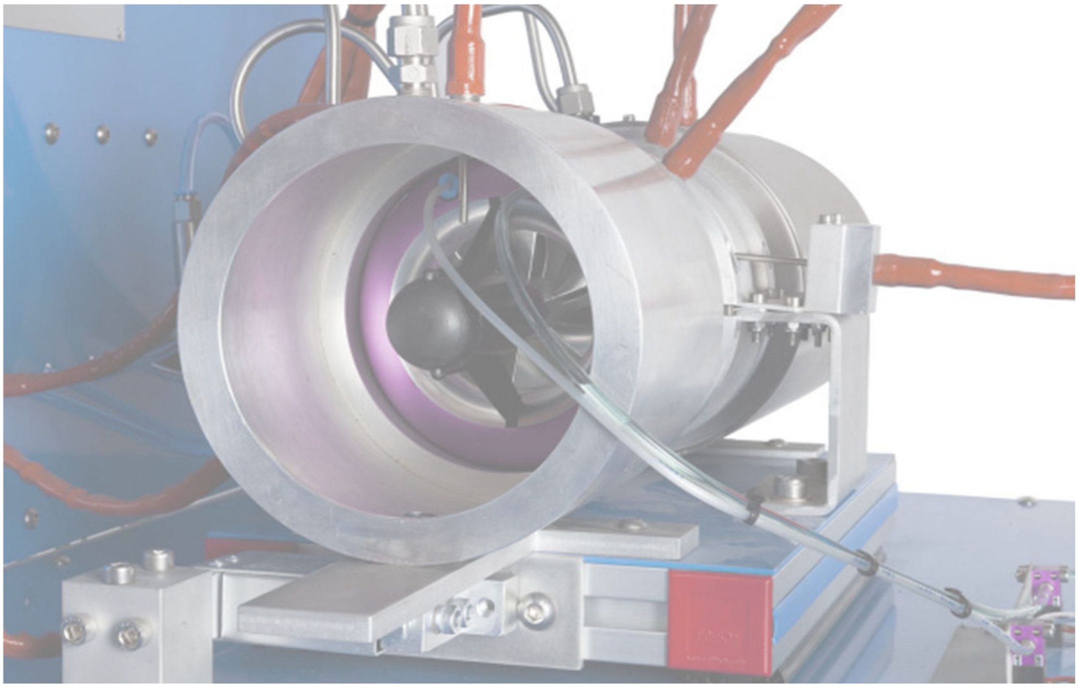

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Foreword

This handout is a course material with corrected applications dedicated to all mechanical engineering students, in particular to students following the Master's course specializing in energy, and can be useful for other disciplines such as hydraulics and maritime engineering.

The support developed respects the university program adopted for the Master's training in energy in its content as well as in its educational architecture (Master1 energy framework).

The main objective of this handout is the dissemination of an educational resource containing useful and simplified information on thermodynamics applied to different thermal installations in particular: receiving machines such as compressors and refrigeration installations as well as internal combustion engines.

This manuscript is organized as follows:

- ✓ A thermodynamic reminder explaining the thermodynamic quantities and functions through the fundamental principles and theories of ideal and real gases.
- ✓ A collection of machines with receiver cycles relating to applications of thermodynamics. The study takes care of the different energy formulations and hypotheses linked to the properties of compressors and refrigeration installations.
- ✓ A simplified view of spark ignition engines and their different types as well as the operation of existing energy cycles.

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Abstract (Arabic)

Abstract (English)

Nomenclatures

T	Temperature	[°C]
θ	Temperature	[K]
Qϱɹɹéɥ	Irreversible heat	[J]
Qɹéɥ	Reversible heat	[J]
d	Volume Density	[-]
δW	Elementary work	[J]
P◌ූ൭/l	External space pressure	[Pascal]
δV	Elementary volume	[m^3]
Pଞ൬/ଞ	System Pressure	[Pascal]
δWϱɹɹ◌ූɥ	Elementary Irreversible work	[J]
Wϱɹɹ◌ූɥ	Irreversible work	[J]
V	Final volume	[m^3]
Vϱ	Initial volume	[m^3]
δQ	Elementary Heat	[J]
C	Molar heat capacity at constant volume	[J/mole/K]
Cг	Molar heat capacity at constant pressure	[J/mole/K]
h, μ	Calorimetric coefficients	[J/mole/pascal]
α	Calorimetric coefficient	[J/mole/m^3]
n	Moles number	[Moles]
m	Mass	[Kg]
L ,L◌ٗ],L ,	Latent heat	[J/Kg]
dϱ f, d◌ූ f	State functions of interior and exterior environments	/
R	Universal ideal gas constant	[J/mole/K]

r	Ideal gas constant	[J/mole/Kg]
b	Correction coefficient (real gas)	[m^3/mole]
a	Correction coefficient (real gas)	[Pascal/$mole^2$/m^6]
Nଊ	Avogadro Number	[Molécules/mole]
γ	Adiabatic constant	[-]
ʒ	Energy efficiency	[-]
U	Internal energy	[J/kg]
K	Kinetic energy	[J/kg]
E	Potential energy	[J/kg]
H	Enthalpy	[J/kg]
S	Entropy	[J/kg]
F	Free energy	[J/kg]
G	Free enthalpy	[J/kg]
ρ	Volume density	[Kg/m^3]
g	Gravitational acceleration	[m/s^2]
z	Altitude	[m]
ηi	Isentropic efficiency	[-]
COP	Performance coefficient (efficiency)	[-]
COP_C	Calorific performance coefficient	[-]
COP_F	Refrigeration performance coefficient	[-]
W6ା, W	High and low pressure compressor work	[J]
Ø6ା, Ø	Mass flow at high and low pressure	[kg/s]
P୍ଵ	Effective power	[KW]
Pୡ	Turbine mechanical power	[KW]
Pୡ6◌	Compressor mechanical power	[KW]
P ୡ◌ାଋଏ୵	Power lost by mechanical friction and auxiliaries	[KW]
η6◌	Rendement du compresseur	[-]

CHAPTER I

Technical thermodynamic reminder

1.1 Concepts of state variables, state equations of ideal gases

I.1.1. Macroscopic description of a thermodynamic system :

A thermodynamic system is characterized by physical quantities accessible on the macroscopic scale, for example:

V: Volume

N: Number of molecules contained in the system

P: Pressure

T: Temperature

The volume density of particles is defined empirically by the relation d = N/V for an homogeneous thermodynamic system.

The mass **m** can also be a characteristic of the system according to the volume density.

I.1.2. Thermodynamic systems

The main purpose of thermodynamics is the study of mechanical phenomena (work, pressure, etc.) coupled with thermal phenomena (heat, temperature, etc.), the study of the coupled phenomena is carried out in a macroscopic description. This science has been hilighted in the 19th century in order to fully understand how the produced thermal machines work during the industrial revolution. The conventional principles of thermodynamics were useful and applicable, and which subsequently went beyond the study of machines, to cross all areas of physics in which heat plays an important role (electromagnetism, optics, etc.), as well as than other scientific disciplines (chemistry, biology, etc.).

The content of a certain volume of space can be called « a system ». The boundaries of this volume can be real (e.g. gas contained inside a closed container) or imaginary.

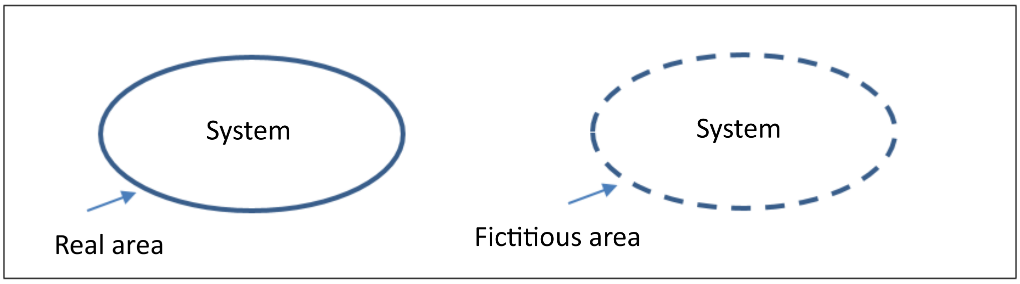


Fig.I.1. Thermodynamic systems

I.1.2.1. Closed systems

A closed system is such that it does not exchange matter with the external environment (it means : its entire boundary is impermeable).

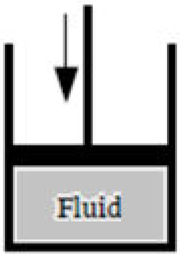


Fig. I.2. Closed systems

I.1.2.2. Open systems

A system is called « opened » if matter exchanges with the external environment is possible (it is sufficient that at least part of the boundary is permeable).

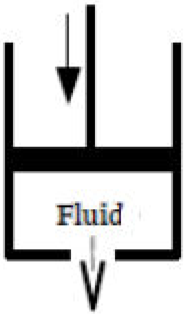


Fig. I.3. Open systems

I.1.2.3. Isolated systems

A system is called « isolated » when there is neither heat exchange or work exchange with the external environment or area. (Example : Calorimeter – thermos).

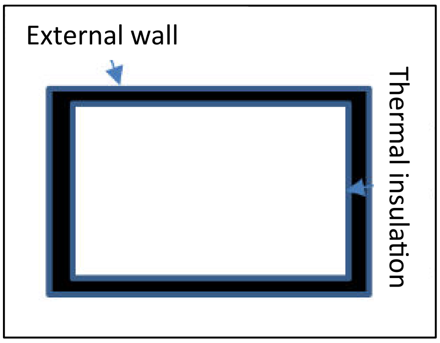


Fig. I.4. Isolated systems

I.1.3. State variables

State variables are generally statistical quantities characterizing the thermodynamic system, and making it possible to describe this system on the macroscopic scale. If these quantities are systematically determinable at any given moment, to describe the history of evolution of this system of the system, these quantities can be considered as state variables. for example: volume, moles number, pressure, temperature.

I.1.3.1. Extensive variables

Quantities proportional to the quantity of material, it means that’s dependent on geometry and defined over the entire system. The moles number, work, and volume are extensive variables.

I.1.3.2. Intensive variables

Quantities independent of the quantity of matter, or, independent of geometry and defined at each point of the system. Pressure and temperature are intensive variables. Consequently, we define a phase (liquid, solid, gas) when, in all or part of a system, the intensive quantities are constant.

I.1.4. Thermodynamic terminology

I.1.4.1. Temperature

Temperature is an intensive macroscopic physical quantity which allows us to account for the thermal state of a body. The measured value of this quantity is used to indicate the level of thermal particles agitation of the system or body. if the system temperature is high, the particles receive more agitation.

Temperature scales: We mainly use two:

The Kelvin scale, called the absolute scale, The temperature T = 0 K is the lowest temperature that exists (it corresponds to zero agitation of the molecules of the body).

The Celsius scale is defined by the relation : $\theta = T - 273.15$, is the absolute scale by convention. The temperature is expressed in °C or in Kelvin degrees K.

I.1.4.2. Work

The calculation of the work **w** exchanged between the system and the external environment is carried out by the following relationship:

$$\delta W = -P_{ext}\delta V \qquad (I.1)$$

With: δW is the elementary work and P_{ext} is the pressure of the external environment:

$$P_{ext} = -P_{sys} = P \ \rightarrow \delta W = PdV \qquad (I.2)$$

If the transformation is reversible at each moment we have :

$$\delta W_{irrev} = -P_{ext}dV \ \rightarrow \ W_{irrev} = -\int_i^f P_{ext}dV = -P_{ext}(V_f - V_i) = -P_{ext}\Delta V \qquad (I.3)$$

1.1.4.3. Quantity of heat Q

In the case of an irreversible transformation, the quantity of heat Q_{irrev} is calculated directly from the first law of thermodynamics. Otherwise, the quantity of heat exchanged with the external environment is defined by :

$$\delta Q = C_V dT + ldV \qquad (I.4)$$

The expression of the heat exchanged with the external environment as a function of the variables T and P with molar heat capacity at constant pressure

$$\delta Q = C_P dT + hdP \qquad (I.5)$$

depending on P and V :

$$\delta Q = \mu dP + \alpha dV \qquad (I.6)$$

I.1.4.4. Latent heats

Latent heat is the necessary heat for a quantity of matter to change its physical state at a constant temperature. It is proportional to the quantity of matter (mass or number of moles) and the value of the latent heat is linked to this physical state changing.

$$Q = nL = mL \tag{I.7}$$

Naturally, for each type of matter, three types of existing latent heats depend on changes in physical state (Ls, Lv and Lf).

Ls, Lv or Lf: are specific or molar heats associated respectively with the change of phases: sublimation, vaporization or fusion.

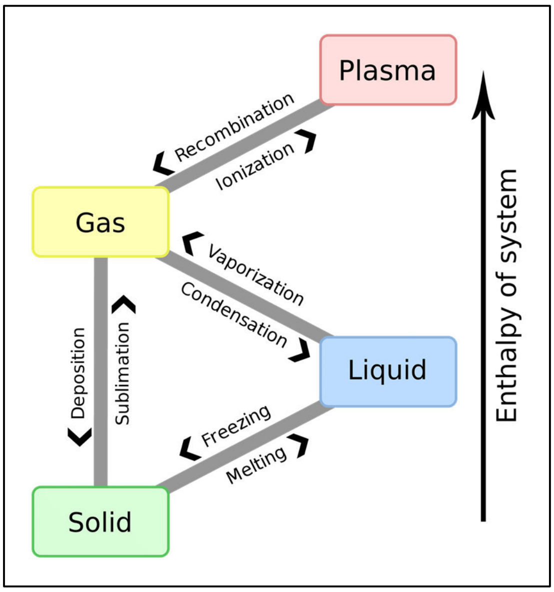


Fig.I.5. Phase change

I.1.5. State functions

State functions are extensive quantities which only depend on state variables. Their values therefore do not depend on previous transformations. Likewise, the variation of these state functions during a transformation is independent of the followed path.

Examples: internal energy U, enthalpy H, entropy S, free energy F, free enthalpy G are all state functions.

$$df = d_i f + d_e f \tag{I.8}$$

I.1.6. Sign convention

By convention, when the system receives, the quantity is counted positively, on other hand, the supplied quantity is negative:

Q > 0: the system receives heat (endothermic process).

Q < 0: the system transfers heat to the external environment (exothermic process).

W > 0: the system receives a woek (receiver system).

W < 0: the system provides work (motor system).

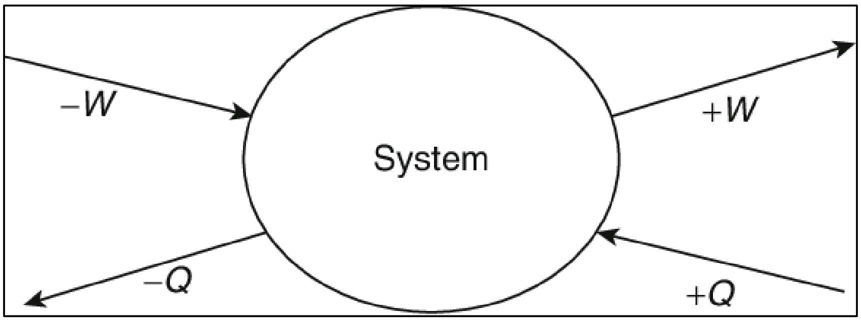


Fig. I.6. Sign convention

I.1.7. Equations of states of ideal gases

The experimental study of gases has led to the definition of the ideal gas model, widely used in thermodynamics. Provided that we operate at sufficiently low pressure, we observe experimentally the three laws cited below :

I.1.7.1. Boyle and Mariotte law

At constant temperature, the pressure **p** of a given mass **m** of gas is inversely proportional to its volume V. ($T = cte : P_1 V_1 = P_2 V_2$)

Which amounts to saying that the multiplication of the pressure **p** of a gas and its volume **V** depends only on the temperature. It was issued in 1662 by Robert Boyle and Edme Mariotte.

I.1.7.2. Gay-Lussac law

At constant pressure **p**, the volume **V** occupied by a given mass **m** of gas is proportional to its temperature. This law was issued in 1800 by Louis Joseph Gay-Lussac.

($P = cte : V_2/V_1 = T_2/T_1$).

I.1.7.3. Avogadro and Ampère's law

Equal volumes of gases of different nature, taken under the same conditions of temperature and pressure, contain the same number of moles. This law was hypothesized in 1811 by Amedeo Avogadro, and stated in 1814 by André-Marie Ampère. ($V = cte : P_2/P_1 = T_2/T_1$).

- ✓ **Definition of an ideal gas**

By definition, an ideal gas is a gas that exactly follows the laws of Boyle and Mariotte, Gay-Lussac, and Avogadro and Ampère.

- ✓ **Ideal gas law**

Gases that obey the state equation are called ideal gases:

$$PV = \mathrm{nRT} \tag{I.9}$$

Where R = 8.314 J.mol-1.K-1 is the universal constant of ideal gases.

For an ideal gas, Mayer's law becomes:

$$C_P - C_V = R \tag{I.10}$$

I.1.8. Van Der Waals Gas Equations of State

I.1.8.1. The Van Der Waals equation

The ideal gas equation of state describes a gas whose molecules are point-like and non-interacting. This equation correctly describes monatomic gases and relatively well polyatomic gases at low density, the deviation from experiment becomes significant. A better approximation is obtained with respect to real gases with the Van Der Waals equation which takes into account the volume of the molecules and introduces a simple interaction term by introducing empirical coefficients a and b.

$$\left(P + \frac{n^2 a}{V^2}\right)(V - nb) = nRT \qquad \text{(I.11)}$$

We see in the Van Der Waals equation of state that a correction is made on the pressure and on the volume using two coefficients a and b.

I.1.8.2. Molar volume

Molecules are considered to be impenetrable spheres of radius r. the Van Der Waals equation of state is established such that the available volume is that of the enclosure minus that of molecules where V-nb, the coefficient b is called covolume it is linked to the radius of the molecules

$$b = \frac{4\pi}{3V^2} N_a r^3 \qquad \text{(I.12)}$$

Where N_a is Avogadro's number equal to 6.022. 10^{23} molecules per mole.

r gets closer of 1nm which is consistent with the real size of the molecules.

I.1.9. Thermodynamic transformations

Transformation is the Chronological succession of states that a system travels between an initial and a final equilibrium states.

The transformation is said to be infinite if it joins two infinitely close equilibrium states and can be finite if it joins two equilibrium states separated by a finite distance.

During a transformation, the state of the system considered evolves because exchanges take place, in this system if it is composite and/or between this system and the external environment. The quantities exchanged are extensive.

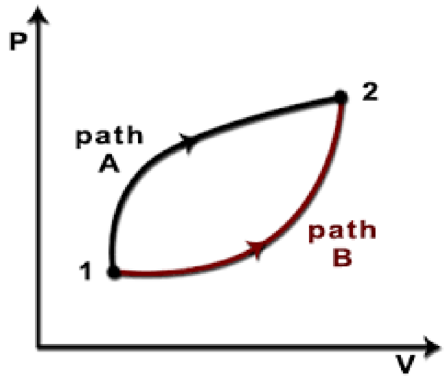


Fig. I.7. Equilibrium and path of thermodynamic systems

I.1.9.1. Reversible and irreversible transformation

A system undergoes a reversible transformation if the following two conditions are met : the transformation that the system undergoes is quasi-static the system is permanently in equilibrium with the external environment. If one of the two conditions is not met, the transformation that the system in question undergoes is described as irreversible. As with quasi-static transformations, it is crucial to clearly specify the system undergoing a transformation before declaring the latter reversible.

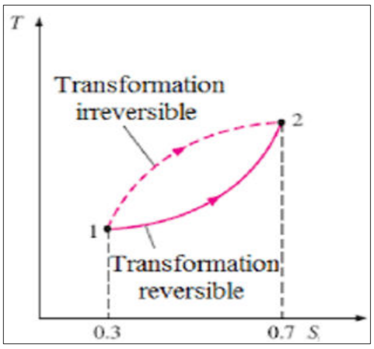


Fig. I.8. Reversible and irreversible transformation

I.1.9.2. Isochoric transformation

An assumed ideal gas and enclosed in a rigid, strained enclosure (dV = 0).

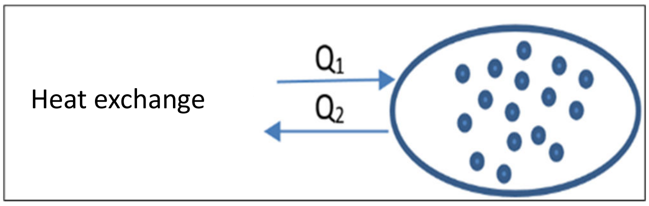


Fig. I.9. Isochoric transformation

The state equation of an ideal gas PV=nRT

Initial state $P_1 V_1 = \mathrm{nRT}_1$

Final state $P_2 V_2 = \mathrm{nRT}_2$

$$V = cte \rightarrow \frac{P_1}{P_2} = \frac{T_1}{T_2} \tag{I.13}$$

I.1.9.3. Isobaric transformation

Given a gas assumed to be perfect and enclosed with an indestructible volume, it undergoes a transformation at constant pressure dP=0.

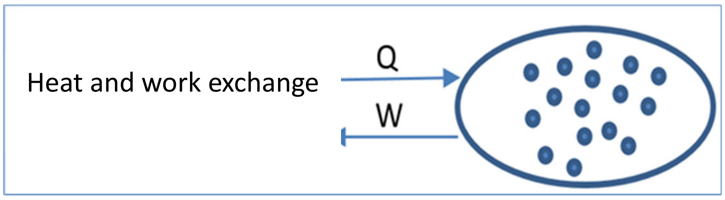


Fig. I.10. Isobaric transformation

Initial state $PV_1 = \mathrm{nRT}_1$

Final state $PV_2 = \mathrm{nRT}_2$

$$P = cte \rightarrow \frac{V_1}{V_2} = \frac{T_1}{T_2} \qquad \text{(I.14)}$$

I.1.9.4. Isothermal transformation dT= 0

Initial state $P_1V_1 = \mathrm{nR}T$

Final state $P_2V_2 = \mathrm{nR}T$

$$T = cte \rightarrow \frac{P_1}{P_2} = \frac{V_2}{V_1}$$

$$\mathrm{PV} = \mathrm{cte} \qquad \text{(I.15)}$$

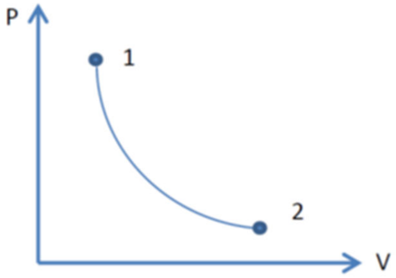


Fig. I.11. Isothermal transformation

I.1.9.5. Adiabatic transformation

The transformation is called adiabatic if the system does not undergo any heat exchange (enclosed) dQ=0

For an ideal gas the equation becomes:

$$\mathrm{V}^{\gamma} = \mathrm{cte} \rightarrow P_1{V_1}^{\gamma} = P_2{V_2}^{\gamma} \text{ (Laplace Relationship)}$$

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^{\gamma} \qquad \text{(I.16)}$$

Note: The adiabatic and reversible transformation is called isentropic.

I.1.9.6. Polytropic transformation

The polytropic transformation encompasses all existing transformations and is defined by the equation:

$$\mathrm{PV}^{n} = \mathrm{cte} \qquad \text{(I.17)}$$

$$\rightarrow \begin{cases} n = 0 \rightarrow \mathrm{P} = \mathrm{cte} \text{ isobaric transformation} \\ n = 1 \rightarrow \mathrm{T} = \mathrm{cte} \text{ isothermal transformation} \\ n = \gamma \rightarrow \mathrm{PV}^{\gamma} = \mathrm{cte} \text{ adiabatic transformation} \end{cases}$$

I.1.10. Thermodynamic cycle efficiency

- ✓ **Ideal thermodynamic cycles :**

A cycle is a thermodynamic states succession through which a system goes back indefinitely, either in time (system without transfer) or in space (system with transfer) » “Closed transformation undergone by a thermodynamic system" Representation ⇒ closed curve in a T-S or P-V thermodynamic diagram.

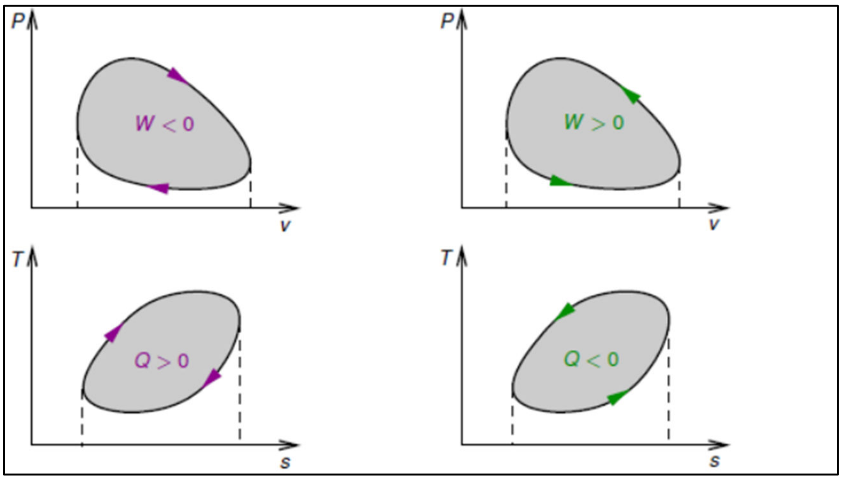


Fig. I.12. Closed thermodynamic cycles

- ✓ **Energy efficiency:**

The energy efficiency of the thermodynamic cycle is the ratio of work supplied and the quantity of heat received.

- ✓ **Exergy efficiency**

Also called second law efficiency or rational efficiency) is a quantity used to calculate the efficiency of a process by taking into account the second law of thermodynamics.

- **Example : Carnot cycle**

Q1: Received heat

Q2: Supplied heat

$$\text{ȝ} = \frac{|W|}{Q} = \frac{Q_1 + Q_2}{Q_1} = 1 + \frac{Q_2}{Q_1}$$

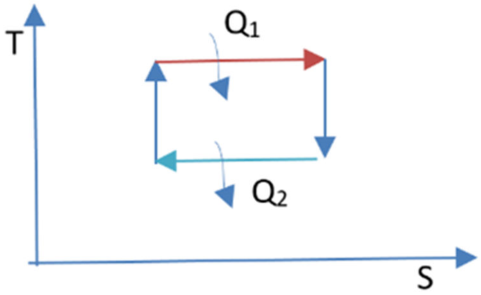


Fig. I.13. Carnot cycle

I.2. First law of thermodynamics

I.2.1. Statement

A closed system that undergoes a thermodynamic transformation exchanges energy with the external environment in two ways:

- ✓ by producing or consuming work W: work of the macroscopic actions exerted by the external environment on the system.
- ✓ by producing or consuming thermal transfer (or heat) Q. (This results at the microscopic level in an incoherent, chaotic movement of the particles of the system).

During any transformation of a closed system, the variation in its total energy is equal to the energy exchanged with the external environment.

$$E_T = (Ec_{macro} + Ec_{micro}) + (Ep_{macro} + Ep_{micro}) \quad (I.18)$$

I.2.2. Concept of internal energy

The internal energy is determined in the form of variation between two states 1 and 2 of a thermodynamic transformation of a system and it is equal to the sum of the quantities of heat and work exchanged during this transformation, i.e. for the total mass.

$$\Delta U + \Delta K + \Delta E = W + Q \quad (I.19)$$

ΔK,ΔE Neglected for a purely thermodynamic system

$$\Delta \mathrm{U}_{1-2} = \mathrm{U}_2 - \mathrm{U}_2 = \mathrm{Q}_{1-2} + \mathrm{W}_{1-2} = \mathrm{Q}_{1-2} - \int_1^2 PdV \quad (I.20)$$

for a cycle : $W + Q = 0$

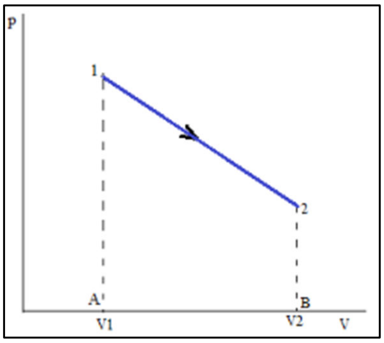


Fig.I.14. Internal energy

I.2.3. Open system and concept of enthalpy

Enthalpy is equal to the sum of the internal energy ‘U’ and the product of the pressure

‘P’ and volume ‘V’ is:

$$H = U + PV \quad (I.21)$$

Enthalpy change

$$dH = dU + d(PV)$$

$$dH = dU + PdV + VdP \quad (dW_{échangé} = -PdV)$$

$$dH = dQ + VdP \quad (I.22)$$

During a transformation of a closed and isolated system from an initial state to a final state, its internal energy is conserved and its variation equal to the quantity of energy exchanged with the external environment by thermal transfer (heat) and mechanical (work for an open system).

I.3. Second law of thermodynamics

I.3.1. Statement

Heat does not itself pass from a cold source to a hot source.

- **Kelvin's statement**

‘’You cannot build an engine that draws heat from a single source and transforms it entirely into work”.

along the thermodynamic cycle the second principle allows to write:

$$\frac{Q_1}{T}+\frac{Q_2}{T}\leq 0 \quad \text{Relation de Clausus} \qquad (I.23)$$

$$\frac{Q_1}{T}+\frac{Q_2}{T}< 0 \quad \textit{Cycle irréversible} \qquad (I.24)$$

$$\frac{Q_1}{T}+\frac{Q_2}{T}= 0 \quad \text{Cycle réversible} \qquad (I.25)$$

I.3.2. Concept of entropy

During an elementary transformation, the second principle is written:

$$dS \geq \frac{\delta Q}{T}\begin{cases} \mathrm{dS} > \dfrac{\delta \mathrm{Q}}{\mathrm{T}} & \text{irreversibletranformation} \\ \mathrm{dS} = \dfrac{\delta \mathrm{Q}}{\mathrm{T}} & \text{reversibletranformation} \end{cases}$$

$\mathrm{ds} \geq \frac{\delta \mathrm{Q}}{\mathrm{T}} \rightarrow \delta \mathrm{Q} - \mathrm{T\,dS} \leq 0$ Criteria of evolution

- ✓ **For an isolated system :**
- ✓ $dS \geq \frac{\delta \mathrm{Q}}{\mathrm{T}}$ et $\delta \mathrm{Q} = 0 \rightarrow \mathrm{dS} \geq 0 \begin{cases} \mathrm{dS} > 0 & \text{irreversible tranformation} \\ \mathrm{dS} = 0 & \text{reversible tranformation} \end{cases}$

I.3.3. Free energy and free enthalpy

- ✓ **Fundamental Gibbs equation**

Entropy, a state variable or state function, makes it possible to express the internal energy variation of a system in another way than that seen previously. Indeed, for a reversible transformation, the joint use of the two principles gives:

$$dU = \delta Q_{rév} + \delta W_{rév} \text{ et } dS = \frac{\delta Q_{rév}}{T} \rightarrow dU = TdS - PdV \text{ (Gibbs equation)} \qquad (I.26)$$

The same process can be done with enthalpy. For a reversible transformation

$$H = U + PV \rightarrow dH = TdS - PdV + d(PV)$$
$$dH = TdS + VdP$$

We obtain a new set of relations by introducing the free energy F

$$F = U - TS$$
$$dH = TdS - PdV + d(TS)$$
$$dF = -SdT - PdV \qquad (I.27)$$

By introducing the free enthalpy G, it becomes :

$$G = H - TS$$
$$\rightarrow dG = TdS + VdP \pm d(TS)$$
$$dG = -SdT + VdP \qquad (I.28)$$

CHAPTER 2

Receiver cycles machines

II.1. Compressors

II.1.1. General concepts :

II.1.1.1. Definition

The compressor is called “heart” of the refrigeration circuit ; it draws in the gaseous refrigerant (at very low temperature and low pressure) from the evaporator, compresses it (at high temperature and high pressure) and then discharges it to the condenser.

Fig. II.1. Piston compressor

II.1.1.2. Operating principle

Compressors or pumps are used to increase the pressure of a fluid and also allow it to be transported by pipeline. Since, gases are compressible, gas compressors also reduce the volume of the compressed gas and can be used only for this purpose (for example, filling a balloon). While liquids are relatively incompressible, the main pump function is the transport of the liquid.

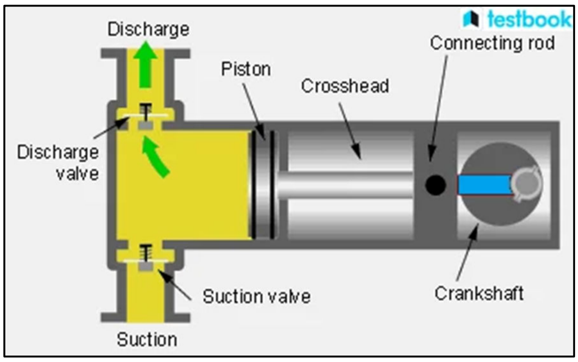


Fig. II.2. Operating principle

II.1.2. Alternative compressors

II.1.2.1. Types of reciprocating compressors

a) Hermetic compressors

The electric motor and the compressor are part of the same hull. These are low-power compressors intended for domestic or commercial applications (refrigeration display cases, individual air conditioners, household refrigerators).

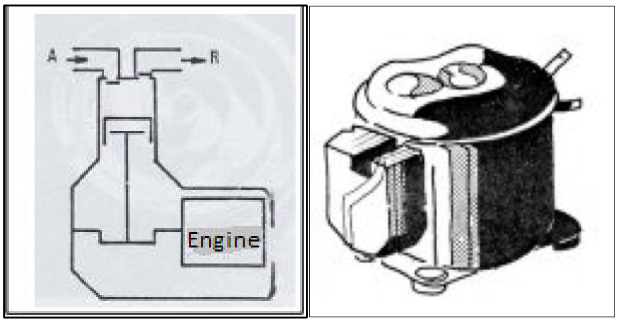


Fig. II.3. Hermetic compressor

Advantage :

- No coupling problems between the driving engine and the compressor.
- Large series production, low cost.
- Reduced congestion.
- Very robust.

Disadvantages:

- Difficult interventions due to the waterproof hull
- Fairly poor performance.
- Little power regulation possible, unless the motor has a double winding, That's immersed in the refrigerant (chemical compatibility, insulation problems, etc.).

b) Opened compressors

The driving engine and compressor casings are separated (open).

One end of the crank shaft passes through the compressor. It can be completely dismantled for repair purposes.

The motor and the compressor are housed in a common body. The cylinder head, the end flanges, as well as the bottom of the crankcase can be dismantled for access to the internal organs during maintenance visits.

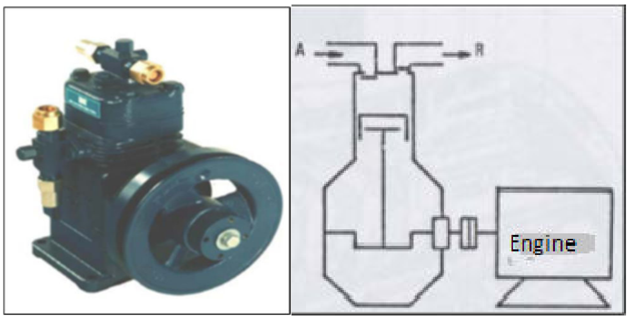


Fig. II.4. Opened compressor

Advantage :

- Choice of training mode
- Robust device

Disadvantages:

- Sealing problems at the shaft output seal. The “open” compressor

c) Semi-hermetic compressors

The motor and compressor casings are joined together (semi-hermetic). The mechanical connection is provided by a transmission shaft.

Advantage :

- Better performance than hermetic compressors
- Easy intervention
- Easier power regulation
- Limitation of risks of leaks
- Rotation speed of piston compressors: typically 1400 or 2800 rpm.

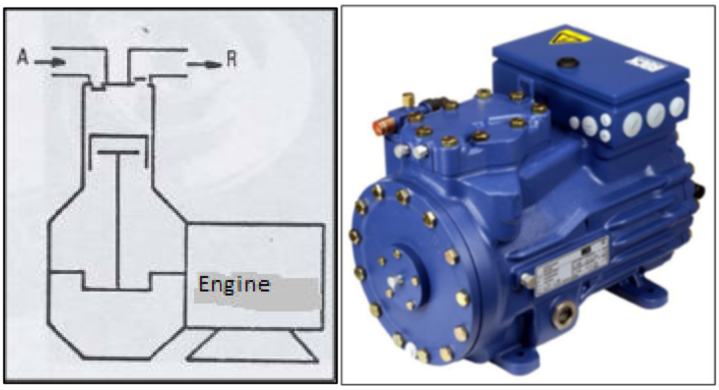


Fig. II.5. Semi-hermetic compressor

II.1.2.2. Single-stage compressors

A single-stage compressor has one or more cylinders. Each cylinder compresses the air or steam to bring it up to operating pressure.

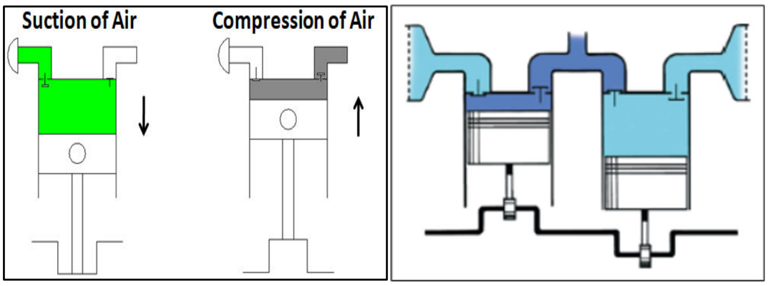


Fig. II.6. Single-stage compressor

II.1.2.3. Multi-stage compressors

A multi-stage compressor has two or more cylinders connected in series, in which the air or vapor is progressively compressed to the final operating pressure. Between stages, the compressed air or steam is cooled by air or water. This improves efficiency, while obtaining a much higher pressure than that provided by a single-stage compressor.

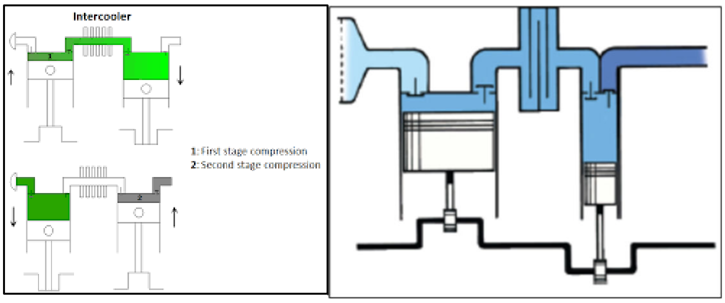


Fig. II.7. Multi-stage compressor

II.1.2.4. Compressor efficiency

Polytropic compression is real compression, the real enthalpy increase is higher than the isentropic enthalpy increase.

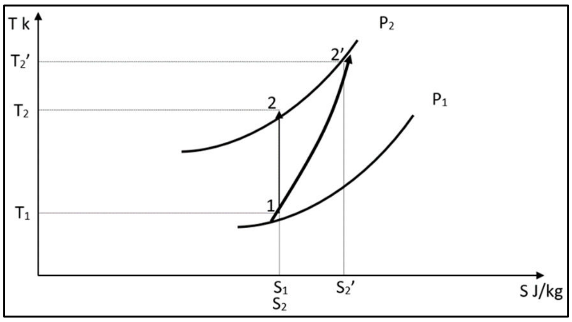


Fig. II.8. TS Diagram : Real compression (polytropic)

Point 2' : End point of real compression

$$H_{2'} - H_1 \rangle H_2 - H_1 \tag{II.1}$$

$$T_{2'} - T_1 \rangle T_2 - T_1 \tag{II.2}$$

With :

$$P.V^n = Cte \tag{II.3}$$

Polytropic exponent $\left(1 \langle n \langle 1.4\right)$

Work can be expressed by :

$$W = PdV = VdP \tag{II.4}$$

The discharge temperature can be written

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \tag{II.5}$$

By analogy we obtain the relation :

$$W_{1-2} = P_1 V_1 \frac{n}{n-1}\left[\left(\frac{P_2}{P_1}\right)-1\right]^{\frac{n-1}{n}} \tag{II.5}$$

Therefore

$$W_{1-2} = rT_1 \frac{n}{n-1}\left[\left(\frac{P_2}{P_1}\right)-1\right]^{\frac{n-1}{n}} \tag{II.6}$$

Likewise, the work depends on the compression ratio, the initial temperature and the compression efficiency (often called isentropic efficiency), can be also written in the following form.

$$\eta i = \frac{H_2 - H_1}{H_{2'} - H_{1'}} \tag{II.7}$$

Or

$$\eta i = \frac{Cp(T_2 - T_1)}{Cp(T_{2'} - T_1)} = \frac{T_2 - T_1}{T_{2'} - T_1} \tag{II.8}$$

The work required for compression is significantly lower than the isentropic compression, therefore, the energy contribution is the lowest in industrial applications, we must always try to approach the perfect compression for a Perceptible cooling.

II.2. Refrigerating machines

II.2.1. Refrigerating setup

A refrigerating machine is a thermodynamic machine intended to ensure the cold of a room or system from an external heat source whose temperature is higher than that the internal room or system to be cooled.

The natural flow of heat always takes place from a hot body to a cold body. The refrigerating machine can also be defined as equipment allowing the flow of heat in the opposite direction to the natural direction, « from a cold environment to a hot environment ». An expenditure of energy will be inevitable to achieve this reverse transfer. The energy required to ensure the transfer must be lower than the useful heat energy to be useful.

Refrigerating or cold production setup for domestic or industrial use (also called refrigerating pumps and which are receiving setups) aims to cool an enclosure (or volume) to keep products colder than the outside environment.

(1-2) Compression

(2-3) Formed Condensation

(3-4) Pressure drop

(4-1) Evaporation

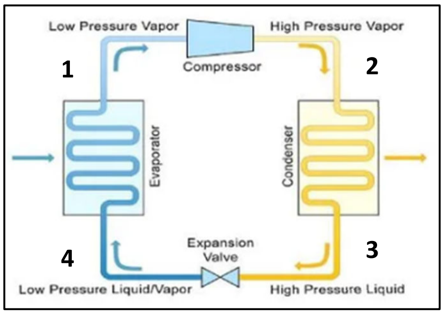


Fig. II.9. Refrigerating setup principle

II.2.2.1. Types of refrigerants

a) CFC (Chlorineofluorocarbon)

The best known CFC was R12. ODP of R12 = 1 GWP100 of R12 = 10,900 according to IPCC 4 and 10,200 according to IPCC 5 Boiling temperature at atmospheric pressure : – 29.8 ˚C. The R12 molecule is made up of one carbon atom, 2 chlorineine atoms and 2 fluorines atoms. The production of CFCs and therefore R12 was banned from January 1, 1995.

b) HCFC (Hydrochlorineofluorocarbon)

The best known of the HCFCs is R22. R22 is a chlorineinated fluid: ODP = 0.055. GWP100 = 1,810 according to IPCC 4 and 1,760 according to IPCC 5. Boiling temperature at atmospheric pressure : – 40.8 ˚C. The R22 molecule is made up of a carbon atom, a chlorineine atom, a hydrogen atom and 2 fluorines atoms. The use of HCFCs, and therefore R22, was banned from January 1, 2015.

c) HFC (Hydrofluorocarbon)

The best known HFC is R134a. ODP of this fluid: 0. GWP100 = 1,430 according to IPCC 4 and 1,300 according to IPCC 5. Boiling temperature at atmospheric pressure : – 26.08 ˚C. The subscript “a” indicates that the molecule is isomeric. Constitution of the molecule: 2 carbon atoms, 2 hydrogen atoms and 4 fluorine atoms. R134a is currently widely used, however, its warming potential being relatively high (1,300), its replacement by low GWP fluids is now being considered.

d) Natural fluides.

Or low GWP fluids The only natural fluids are actually air and water. CO_2, propane, isobutane and ammonia are all chemically transformed. « Saying that are natural is not accurate », it is better to call them low GWP (Low global warming potential) fluids. They are classified in the R700 series. The standardized name is R700 to which the molecular mass value must be added. Exemples:

- Water (H_2O) = R (700 + 2 + 16) = R718

- Ammonia (NH_3) = R (700 + 14 + 3) = R717, ODP = 0, GWP100 = 0

- Carbon dioxide (CO_2) = R (700 + 12 + 32) = R744, ODP = 0, GWP100 = 1e.

e) Hydrocarbons

These are chemical compounds whose molecule contains only carbon and hydrogen.

Examples: - Propane (R290) Boiling temperature at normal atmospheric pressure -42 ˚C, ODP = 0, GWP = 3

- Isobutane (R600a) or methylpropane Boiling temperature at normal atmospheric pressure : – 11.7 ˚C ODP = 0 GWP = 3. These fluids being very flammable, their use is limited to low powers, for example cold domestic and in sealed hermetic equipment. Furthermore, the handling of these fluids requires specific training.

II.2.2.2. Environmental impact

The environmental impacts linked to refrigerants are based on two phenomena:

- The ozone layer depletion
- Global warming

➢ The ozone layer depletion

Ozone is a form of oxygen made up of three atoms instead of two. It is an unstable gas and is particularly vulnerable to attack from natural compounds containing hydrogen, nitrogen and chlorineine.

Ozone located in the stratosphere (region located between 11 and 48 km above the earth's surface) is essential to life as oxygen. It indeed forms a shield that is certainly extremely thin but effective because it manages to filter almost all of the harmful ultraviolet rays of the sun (absorption of most UV B rays). The formation and destruction of the ozone layer is a cyclical and natural process following the following reactions:

$UV + O_3 \longrightarrow O_2 + O$ (Destruction of the ozone layer) – UV: Ultraviolet light – O_3: Ozone

$O_2 + O \longrightarrow O_3$ (Ozone shaping)

Indeed, UV acts on the molecules of certain fluids (mainly CFCs and to a lesser extent HCFCs) to release the chlorine atoms and it is these atoms which will react with the ozone to destroy it following a chain reaction.

- **Case of CFC R12 (CF_2Cl_2)**

$UV + CF_2Cl_2 \longrightarrow Cl + CF_2Cl + O$ (Realise of chlorine atom CFC R12)

$Cl + O_3 \longrightarrow ClO + O_2$ (Reaction of released chlorine atom with Ozone: Ozone Destruction)

$ClO + O \longrightarrow Cl + O_2$ (Reaction of ClO with O and release of chlorine atom again...)

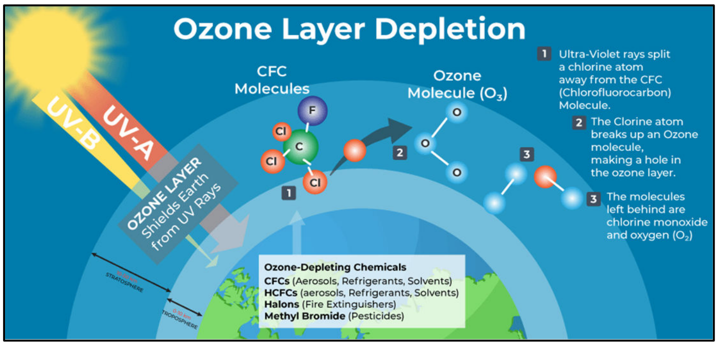


Fig. II.10. The Ozone layer depletion

➢ Global warming (greenhouse effect)

The earth's temperature is maintained by a balance between the warming effect emanating from solar radiation coming from space and the cooling effect of infrared rays emitted by the hot surface of the earth's crust and the atmosphere which rise towards the 'space.

Solar radiation in the form of visible light that reaches the earth is divided into several parts: - part is absorbed by the atmosphere

- A part is reflected by the clouds and the ground (especially the desert and the snow) the rest is absorbed by the surface which is heated and which in turn heats the atmosphere, the heated surface and the atmosphere of the earth emit long-wavelength infrared (IR) rays.

- Part of the long-wavelength IR rays returned is absorbed in the atmosphere by certain gases, thus slowing down the release of radiation, thus cooling and warming the surface of the earth, this is the greenhouse effect which natural phenomenon without which life on earth would be unlivable with an average temperature of -18°C compared to +15°C currently on the surface of the earth.

The gases present in the atmosphere and which absorb part of this IR radiation are called greenhouse gases. These are mainly CO2, water vapor, methane (CH4) and nitrous oxide (N2O) and refrigerants released into the atmosphere (mainly CFCs).

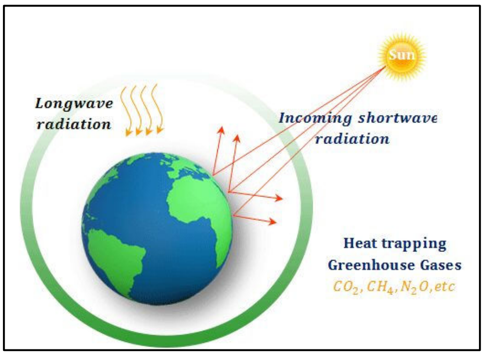


Fig. II.11. Global warming (greenhouse effect)

II.2.3. Performance of a refrigerating setup

The coefficient of performance (COP) is defined by the ratio of the quantity of heat absorbed by the evaporator to the quantity of heat supplied to the generator plus the work of the solution pump.

For a dithermic refrigeration system (1), consuming mechanical energy, or equivalent, W, by absorbing a quantity of heat Qf in the cold source at the temperature Tf and by releasing a quantity of heat Qc to the hot sink at the temperature Tc, the system performance coefficient is, in this case:

- ✓ **Cold production (refrigerating machine) :**

$$\mathrm{COP_{MF}} = \frac{|Q_f|}{|Q_c - Q_f|} \text{ et } (\mathrm{COP_{MF}})_{\text{théo}} = \frac{T_f}{T_c - T_f} \quad \text{(II.9)}$$

Car : $Q_f = T_f \Delta S_f;\ Q_c = T_c \Delta S_c\ et\ \Delta S_f = \Delta S_c$ (isentropic)

- ✓ **Heat production (heat pump) :**

$$\mathrm{COP_{PC}} = \frac{|Q_c|}{|Q_c - Q_f|} \text{ et } (\mathrm{COP_{PC}})_{\text{théo}} = \frac{T_c}{T_c - T_f} \quad \text{(II.10)}$$

The coefficient of performance of the most efficient real systems is between 50 and 70% of that of ideal systems operating between the same temperatures, this in the range of usual temperatures.

II.2.4. Refrigerating cycles

The purpose of a refrigeration setup is to extract a quantity of heat from a volume maintained at a temperature lower than the ambient temperature. The extracted heat must be also transferred outside the system. That’s why, we use a physical property of fluids, ‘the boiling temperature varies with pressure’, especiallly, for the refrigerant fluids, Whereas, the boiling temperature is lower than the ambient temperature (see Fig. II.9).

The reference refrigeration cycle (practical cycle) is a compromise which allows the study and sizing of refrigeration machines with acceptable precision.

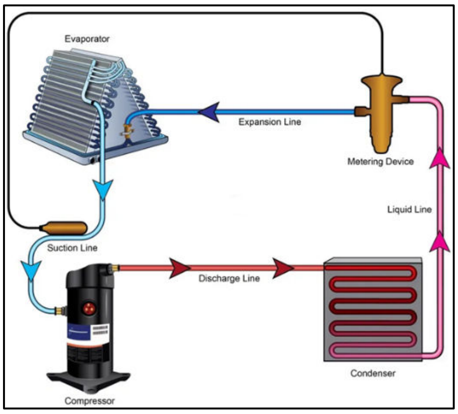


Fig. II.12. Refrigeration cycle

II.2.4.1. Refrigeration cycles with steam compression

II.2.4.1.1. Basic cycle (Carnot cycle)

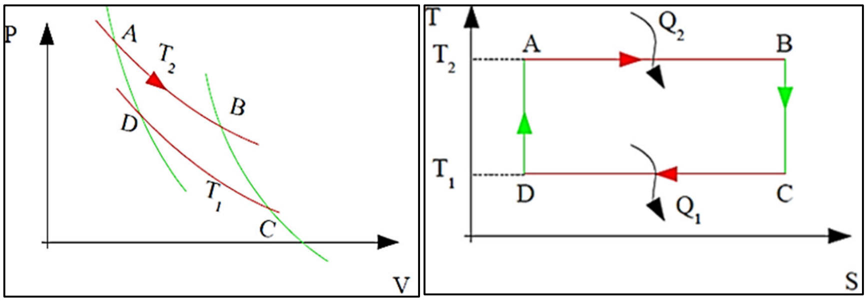


Fig. II.13. TS, PV diagrams

Reversible cycle :

$$\Delta S = \frac{Q_1}{T_1} + \frac{Q_2}{T_2} = 0 \qquad \text{(Second Principle)}$$

$$\Delta U = 0 \qquad \text{(Cycle fermé)}$$

$$U_A - U_A = (U_A - U_D) + (U_D - U_C) + (U_C - U_B) + (U_B - U_A) = 0 \qquad \text{(II.11)}$$

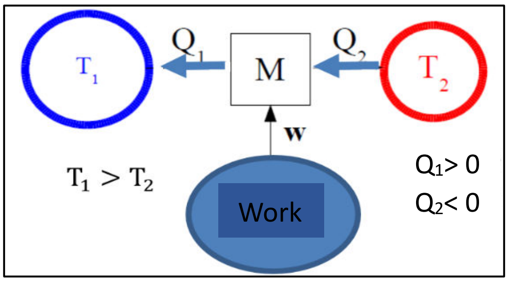


Fig. II.14. Refregeration heat transfer

$$U_i - U_j = Q_{ji} + W_{ji} = 0 \tag{II.12}$$

$$Q_{AB} + Q_{CD} + W_{ij} = 0 \tag{II.13}$$

$Q_1 + Q_2 + W = 0$

Refregerator cycle efficiency

$$\delta = \frac{Q_2}{W} = \frac{Q_2}{-Q_1 - Q_2} = \frac{Q_2}{-Q_1 + Q_1\frac{T_2}{T_1}} = \frac{T_2}{T_2 - T_1} \tag{II.14}$$

$\delta = \frac{T_2}{T_1 - T_2}$ (reversible cycle)

II.2.4.1.2. Single-stage compression refrigeration cycles

The cold production is not distinguished from the production of heat only in terms of temperature scale or in terms of the quantity of heat exchanged sign. Indeed, we cannot say that the cold production is only a production of negative heat at low temperature. The main distinction comes from the existence of the second principle of thermodynamics according to the Clausius postulate "A heat transfer can’t be carrie out from a cold body to a hot one without compensation” whereas it is entirely possible, without compensation, for heat to pass from a hot body to a less hot body. We can therefore define the production of cold as the implementation of a series of thermodynamic transformations. These transformations are undergone by an active substance (the refrigerant), which takes heat from the cold source, and rejects it into the hot source but energy must be supplied to compensate it.

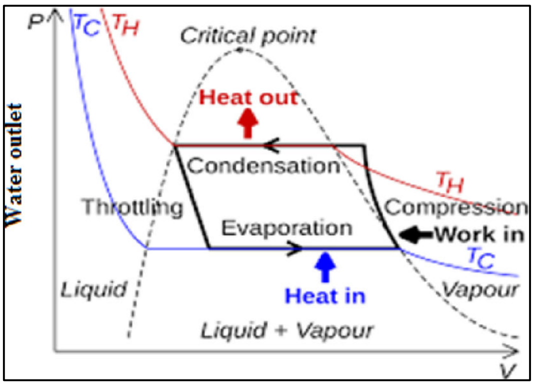


Fig. II.15. Single-stage compression cycle

a) Ideal cycle (reversible)

(1-2) Isentropic compression : $W_C = H_2 - H_1$

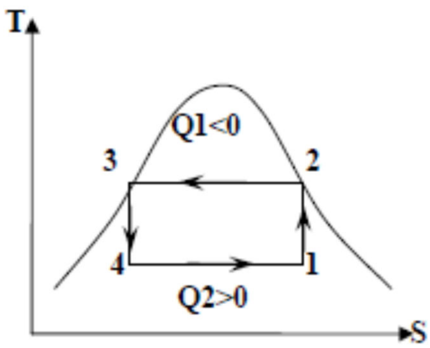
(2-3) Isobaric condensation :

$$Q_{1rev} = H_3 - H_2 = T_1(S_3 - S_2)$$

(3-4) Isentropic expansion (rolling): $W_D = 0$

(4-1) Isobaric évaporation:

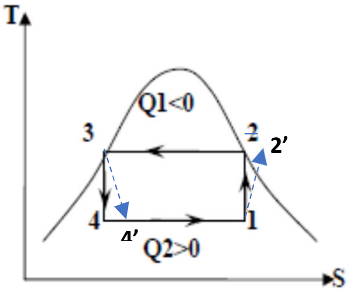
$$Q_{2rev} = H_1 - H_4 = T_2(S_1 - S_4)$$



$$\text{COP}_{rév} = \frac{Q_{2rév}}{|Wrev|} = \frac{Q_{2rév}}{|Q_{2rév}-Q_{1rév}|} = \frac{H_1-H_4}{|(H_1-H_4)-(H_3-H_2)|} = 1 + \frac{H_1-H_4}{H_3-H_2} \qquad \text{(II.15)}$$

$$\text{COP}_{rév} = \frac{\text{T}_2}{\text{T}_1-\text{T}_2} \quad \text{Carnot cycle } (\Delta S_{14} = \Delta S_{32}) \qquad \text{(II.16)}$$

b) Real cycle (irreversible)

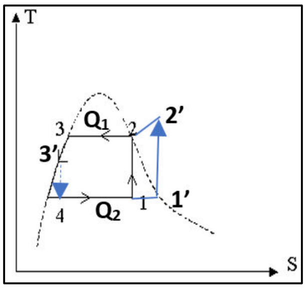


(1-2') Real compression : $W_{C'} = H_{2'} - H_1$
(2'-3) Isobaric condensation : $Q_{1\text{irrév}} = H_3 - H_{2'} = T_1(S_3 - S_{2'})$
(3-4') Real expansion (rolling) : $W_D = W_{34'} = 0$
(4'-1) Isobaric evaporation : $Q_{2\text{irrév}} = H_1 - H_{4'} = T_2(S_1 - S_{4'})$

$$COP_{irrév} = \frac{Q_{2irrév}}{|W_{irrév}|} = \frac{Q_{2irrév}}{|Q_{2irrév}-Q_{1irrév}|} = \frac{H_1-H_{4'}}{|(H_1-H_{4'})-(H_3-H_{2'})|} = 1 + \frac{H_1-H_{4'}}{H_3-H_{2'}} \qquad \text{(II.17)}$$

$$\boxed{\begin{cases} W_{irrev} > W_{rev} \\ Q_{2irrev} < Q_{2rev} \\ Q_{1irrev} > Q_{1rev} \end{cases}}$$

c) Ideal cycle with cooling after condensation (under cooling)



(1'-2') Isentropic Compression : $W_{C'} = H_{2'} - H_{1'}$
(2'-3) Isobaric condensation : $Q_1 = H_3 - H_{2'} = T_{2'}(S_3 - S_{2'})$
(3-3') Under isobaric cooling: m.cp. m.cp.$(T_{3'} - T_3)$
(3'-4) Isentropic expansion (rolling): $W_D = W_{34'} = 0$
(4-1') Isobaric evaporation: $Q_2 = H_{1'} - H_4 = T_{1'}(S_{1'}\_S_4)$

$$COP = \frac{Q_2}{|W_C|} = \frac{H_{1'}-H_4}{H_{2'}-H_{1'}} = 1 + \frac{T_{1'}(S_{1'}\_S_4)}{H_{2'}-H_{1'}} \qquad \text{(II.18)}$$

$$\begin{cases} W_{irrév} > W_{rév} \\ Q_{2irrév} < Q_{2rév} \\ Q_{1irrév} > Q_{1rév} \end{cases}$$

II.2.4.1.3. Double-stage compression refrigeration cycles

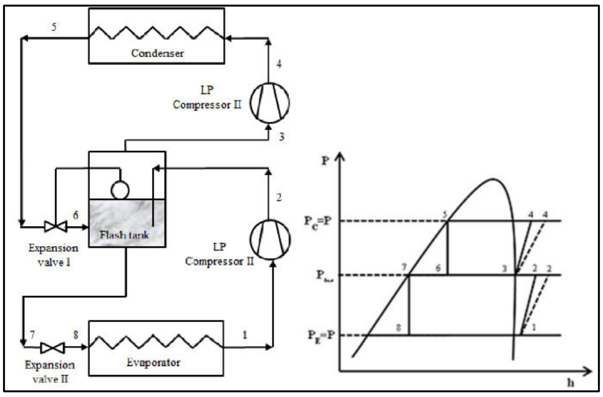


Fig. II.16. Refrigerator diagram.

- For industrial applications such as the cold production at low temperatures, it may be necessary to adopt different evaporation and condensation temperatures. The compression ratio of the refrigeration machine very high.
- The mass refrigeration production decrease is due to the steam volume increase.
- The evaporation temperature drop and the compression ratio increase in results a significant increase in the temperature of the discharged steam.
- The simple fact of splitting the compression does not make it possible to limit the temperature of the discharged steam: it is also necessary to implement a steam cooling system during compression. This cooling takes place between the two compressors, at the intermediate pressure.
- Simplest double-stage cycle.
- The cooling of the steam in this cycle can be achieved in two ways:
  1. Full injection cycle.
  2. Partial injection cycle

a. Full injection cycle

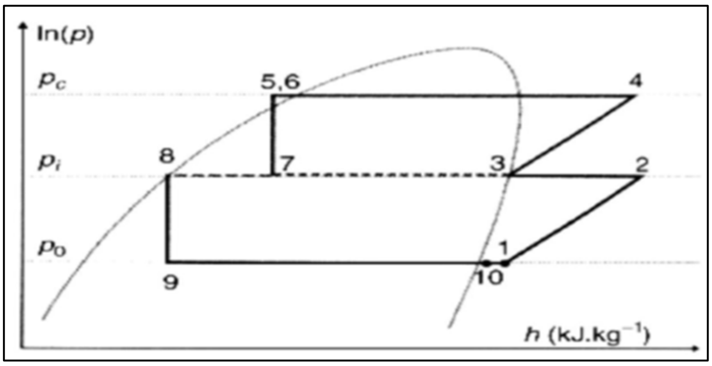


Fig. II.17. Refrigerator diagram.

1-2 : Steam compression from P_0 to P_i.

2-3 : Desuperheating of the compressed steam into the intermediate bottle. This desuperheating is ensured by the vaporization of a small fraction of the liquid contained in the bottle.

3-4 : Steam compression of from P_i to P_c. Point 3 can also correspond to overheated steam (depending on the quality of the thermal insulation).

4-5 : Desuperheating, condensation at P_c, and sub-cooling of the liquid form.

5-6 : sub-cooling and possible pressure losses in the liquid line.

6-7 : Isenthalpic expansion of the liquid from P_c to P_i.

7-8 : Separation of the liquid in the intermediate bottle.

8-9 : Isenthalpic expansion of the saturating liquid from P_i to P_0

9-10 : Vaporization, pressure P_0.

10-1 : superheating and possible pressure drops in the suction piping.

7-3 : Steam separation in the intermediate bottle; suction by the HP compressor (3-4).

- We assume that the intermediate pressure is already constant.

- The choice of other internal quantities is made in the same way as for single-stage cycles.

- **Mass flow rate of steam sucked in by the LP compressor :**

$$\emptyset_B = \frac{Q_2}{H_{10} - H_9} \tag{II.19}$$

- **Mass flow rate of steam sucked in by the HP compressor :**

The flow rate is deduced from an enthalpy balance on the separator bottle:

$$\emptyset_B . H_2 - \emptyset_H . H_3 + \emptyset_H . H_6 - \emptyset_B . H_8 + Q_{bt} = 0 \tag{II.20}$$

Q_{bt} Thermal power received by the intermediate bottle (loss through insulation).

$$\emptyset_H = \emptyset_B . \frac{H_2 - H_8}{H_3 - H_6} + \frac{Q_{bt}}{H_3 - H_6} \tag{II.21}$$

If the losses in the intermediate bottle are neglected :

$$\emptyset_H = \emptyset_B . \frac{H_2 - H_8}{H_3 - H_6} \tag{II.22}$$

The COP performance coefficient of the installation is expressed by :

$$COP = \frac{Q_2}{W_H + W_B} \tag{II.23}$$

b. Partial injection cycle

The LP regulator is not supplied from the intermediate bottle under pressure $\mathcal{P}0$, but by liquid at pressure $\mathcal{P}$k. However, with liquid taken directly from the condenser outlet, the mass refrigeration production would be too low. To increase this quantity, the subcooling of the LP mass flow is increased using a sub-cooler exchanger placed in the liquid contained in the intermediate bottle.

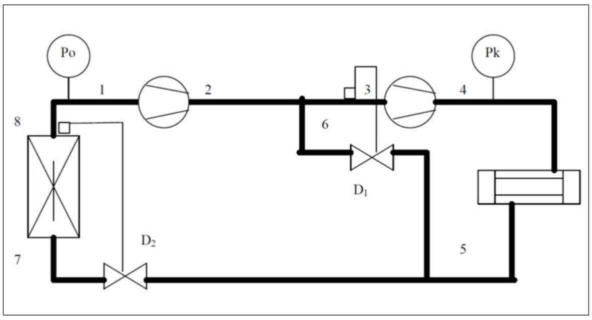


Fig. II.18. Double-stage compression refrigeration

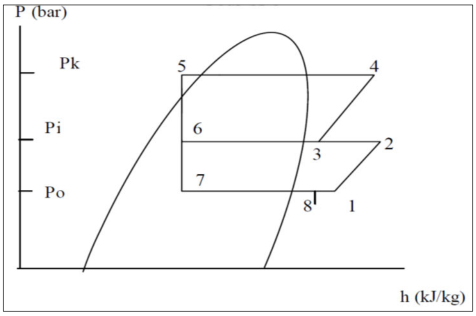


Fig. II.19. Partial injection cycle

Unlike the total injection case, the expansion of the mass flow supplying the evaporator is not fractionated, it is a single-stage expansion between P_k and P_0.

The HP mass flow also increases; in fact, if we neglect the thermal losses through the insulation, the ratio of the two flow rates is expressed by the relation:

$$\emptyset_H = \emptyset_B . \frac{H_2 - H_8}{H_3 - H_6} \tag{II.24}$$

II.2.5. Thermal power

II.2.5.1. Thermal power of the evaporator

If the general assessment is established for 24 hours, it is obvious that for reasons of good performance and longevity of the equipment, the slightest drop in efficiency or an accidental shutdown of the machine would jeopardize the daily refrigeration production.

The cooling capacity of the machine is therefore calculated on an average operating time of between 14 and 20 hours.

The refrigerating power necessary for the evaporator will therefore be under these conditions, called **t** the operating time of the refrigerating machine expressed in seconds :

$$\mathrm{P_E} = \frac{\mathrm{Q_T}}{\mathrm{t}} \ (\mathrm{KW}) \tag{II.25}$$

Q_T : Bilan total des quantités de chaleur ; t : Temps de fonctionnement

If we neglect the pressure losses in the installation :

$$\mathrm{P_E} = \frac{\mathrm{Q_2}}{\mathrm{t}} (\mathrm{KW}) \tag{II.26}$$

Q_2: Quantity of heat extracted from the cold source in the evaporator.

II.2.5.2. Thermal power of the compressor

If the compressor only supplies the refrigeration installation for which the heat balance calculation has just been carried out, the refrigerating power of the evaporator is equal to the useful refrigerating power of the compressor under conditions 00 and 0k corresponding to the evaporation temperatures and condensation specific to the installation.

$$\mathrm{P_C} = \mathrm{P_E}\frac{\Delta \mathrm{H}_C}{\Delta \mathrm{H}_E} \ (\mathrm{KW}) \tag{II.27}$$

With ΔH_E the enthalpy variation between the evaporator outlet and inlet, in kJ/kg, and ΔH_C the enthalpy variation between the evaporator inlet and the compressor suction, in kJ /kg.

II.2.6. Technical choice and sizing of a refrigeration setup

II.2.6.1. Choice of evaporator

- Type of product to cool or store
- Fin spacing
- Materials used
- Evaporator type and implementation
- Available pressure
- Acoustic noise

II.2.6.2. Choice of compressor

The current sizing of the compressor for a food cold installation is naturally conditioned by:

- Refregeration power to be provided
- Type of refrigerant;
- Temperature necessary for the application at the level of the evaporator (positive or negative cold, type of food to be stored, etc.) and this, in optimal conditions

- Extreme temperature that can prevail at the condenser (air or water temperature depending on the type of condenser).

II.2.6.3. Choice of regulator

- Low investment cost
- Very good reliability
- Relatively simple adjustment
- Also lower maintenance costs
- Only works correctly at low Δp by reducing the filling rate of the evaporator
- Its regulation is not very fine compared to that of electronic regulators
- Limited accuracy in measuring overheating

II.2.6.4. Choice of condenser

- Type of heat transfer fluid available (air, fresh water, sea water)
- Heating power to reject
- Nominal condensation and heat transfer fluid temperatures, estimated or requested subcooling
- Type of technology that we want or can use, space, noise and environmental constraints

II.3. Heat pumps

II.3.1. Operating principle

The principle is based on absorbing thermal energy (heat) from outside the house to release it inside the house (hence its name). It is therefore simply a refrigerator open to the outside and whose rear grille is placed in our apartment.

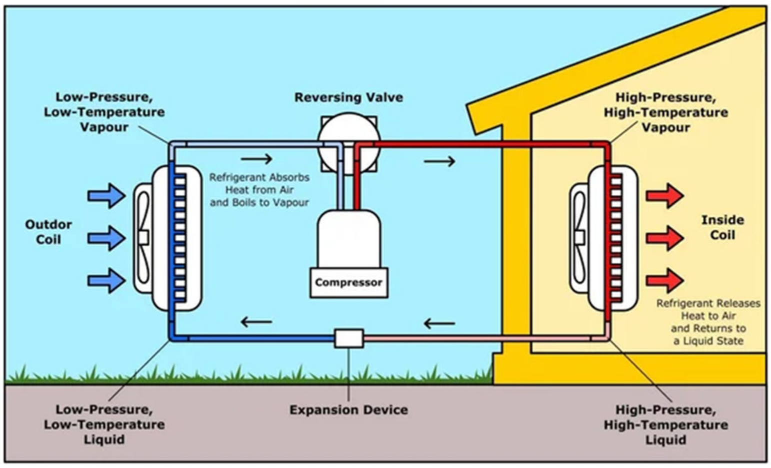


Fig.2.20. Air source heat pump

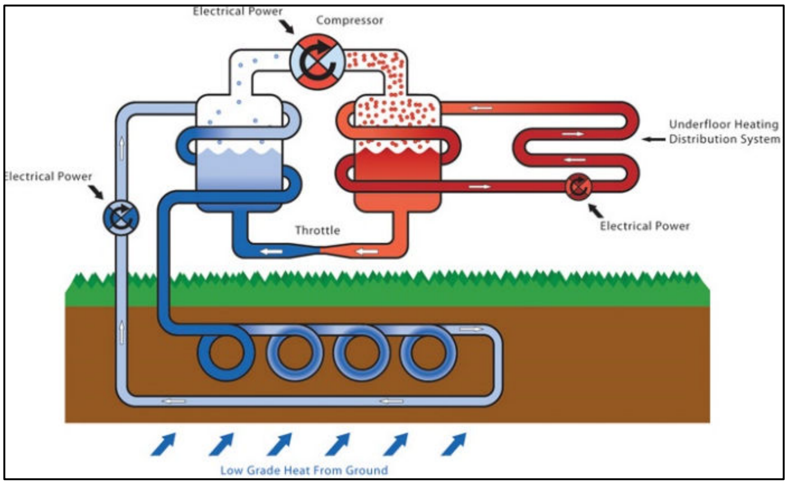


Fig.2.21. Geothermal source heat pump

The heat pump absorbs heat Q_2 from the outside and the hot grid of the refrigerator which heats the room. In this case the heat Q_1 transferred to the internal area. Therefore, the efficiency is defined by :

$$\mathrm{COP} = \left|\frac{Q_1}{W}\right| \tag{II.28}$$

For the Carnot cycle (reversible dithermal cycle) we have :

$$COP_{max} = \frac{T_1}{T_1 - T_2} \tag{II.29}$$

CHAPTER 3

Ideal cycles of inernal combustion engines

III.1. General information on internal combustion engines

III.1.1. Description

A device which provides mechanical energy by transforming the heat energy released by the calcination of a fuel. There are four main types of internal combustion engines : the spark-ignition engine, or internal combustion engine, the diesel engine, the rotary piston engine and the gas turbine. Many types of engines use the principle of jet propulsion (Rocket). The spark-ignition engine, invented by the German technician Nikolais August Otto, is the classic gasoline engine, used to propel automobiles and airplanes; The Diesel engine, designed by Rudolf Christian Karl Diesel, operates on a different principle and uses diesel or heavy oil as fuel. This engine is used in electric generators, propulsion of ships, trucks and buses, and in some automobiles. The spark ignition engine and the Diesel engine are available in two-stroke or four-stroke versions.

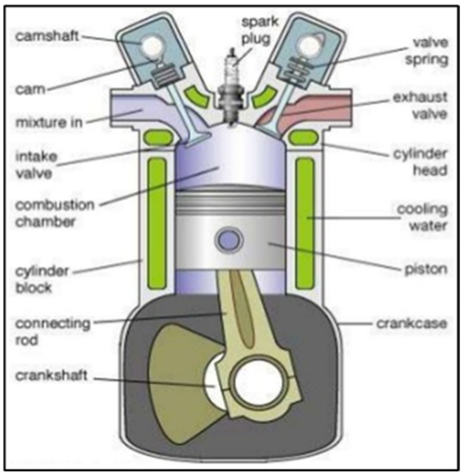


Fig. III.1. Internal combustion engine

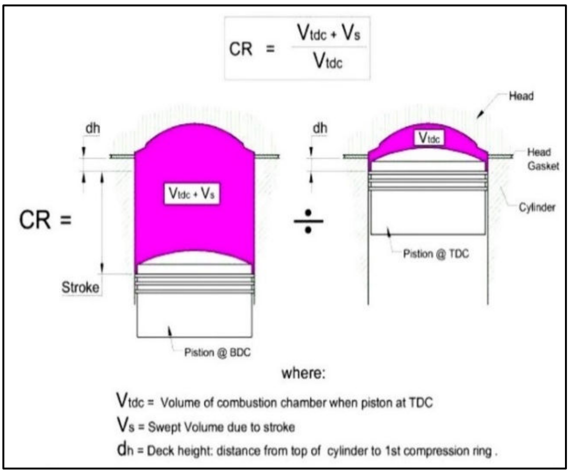


Fig. III.2. Volumetric ratio

The AC engine is first of all a reciprocating internal combustion engine, the work is produced by the combustion of a fuel mixture inside a cylinder, in which a piston moves in reciprocating movement. The two extreme limits of movement are called top dead center (TDC) and bottom dead center (BDC), respectively. The volume swept between these two points constitutes the unit displacement, and if d is the diameter of the cylinder (or bore) and C the stroke of the piston, the displacement V is written:

$$V = C\pi\frac{d^2}{4} \qquad \text{(III.1)}$$

At top dead center the residual volume is called dead volume ; it determines what we call the combustion chamber, which is therefore the portion of volume limited by the cylinder head, the top of the liner and the upper part of the piston. From the dead volume v and the cylinder capacity V, the volumetric compression ratio is deduced.

$$CR = \frac{V+v}{v} \qquad \text{(III.2)}$$

All internal combustion engines operate following the same general process described schematically below. A variable volume is delimited by a cylinder, one of its bases which is fixed, called a cylinder head, and the other which is a movable piston in the bore of the cylinder, driven by a connecting rod-crank system. In a four-stroke engine, the organs which control the delivery or admission are valves actuated by pushers coupled to the engine shaft by a camsh

The production of spark-ignition (AC) engines is largely the majority in the world, since it represents 85% of the total of alternative thermal engines (102 millions units in 2000, out of a total production of 121 million). France stands out with higher than average diesel production, which brings the proportion of AC engines to 52% (around 1.75 million units). Since diesel engines are rather reserved for high-power units, it is obvious that the proportions indicated would become significantly different if they were evaluated based on the kilowatts produced.

The reserved domain of the AC motor is automobile traction, which represents 46% of world production of this type of motor (36% for private vehicles, 10% for trucks), the proportion reaching 65% if we include motorcycles. The rest is shared between agricultural engines (21%), industrial engines (12%), marine and aeronautical engines (2000 statistics). The vast majority of these engines are 4-stroke cycle (> 90%), with 2-stroke engines being confined to small displacements (less than 500 cm3), the preferred domain of motorcycles and outboard motors.

III.2.1. Operation principle

III.2.1.1. Four-stroke cycles

- **During intake :** the combustion chamber is open, the piston descends from top dead center (TDC) to bottom dead center (BDC), which increases the volume of the chamber. The air + fuel mixture is admitted into the combustion chamber at a constant pressure; we speak of isobaric transformation (P= Cte).

- **During compression** : the chamber is closed, the piston rises from BDC to TDC, the chamber volume decreases, the pressure and temperature increase; we speak of adiabatic transformation (Poisson's law: P.V = Cte).

- **During ignition and combustion** : the chamber remains closed, and the piston does not move. The transformation of the energy contained in the fuel causes a sharp increase in temperature and pressure, resulting in an explosion ; we speak of isochoric transformation (V = Cte).

- When the ignited gas is released: the piston moves again towards BDC, the pressure drops according to a new adiabatic transformation.

- During the fourth stage, the exhaust, the burnt gases are evacuated from the cylinder by the rise of the piston according to a new isobaric transformation.

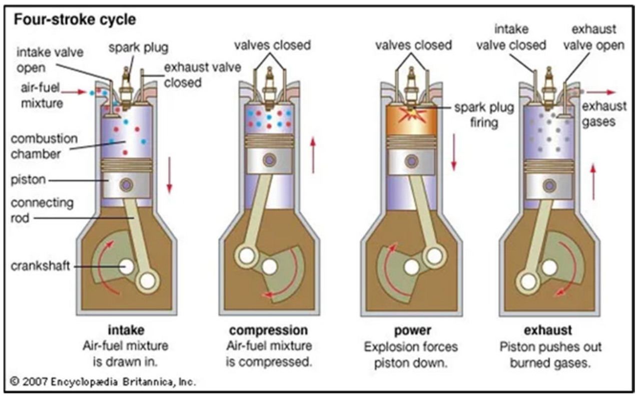


Fig. III.3. Four-stroke engine

III.2.1.2. Two-stroke cycle

The major differences in the structure of the cylinder of a “2-stroke” engine compared to a “4-stroke” engine :

- The absence of a valve

- A transfer light

- The piston can be extended by a skirt

- The cylinder is pierced with several lights

Step 1: Piston rising to the highest point

Above the piston, the air/fuel mixture is compressed in the cylinder head. Thus, we clearly find the compression stage of the 4-stroke engine. Below the piston, as it rises, a depression is created in the engine crankcase. The piston will then clear the intake port and thanks to the depression which then prevails, the air/fuel mixture will be able to enter inside the engine. This correspond to the intake stage of the 4-stroke engine.

Step 2: Piston descending to the lowest point

Above the piston, the candle emits the spark, the gas ignites. The pressure increases and the piston is pushed down. It's relaxation. Arriving at approximately the lowest point, the piston clears the exhaust port and the exhaust gases will be pushed by the fresh mixture which arrives through the transfer port which is discovered shortly after the exhaust port. This fresh mixture arrives all the more quickly as it is located below the piston and as it is compressed by the descent of the piston. This step is called scanning. The gas then finds itself above the piston as it rises. Thus, expansion and exhaust in a two-stroke engine occur at the same time.

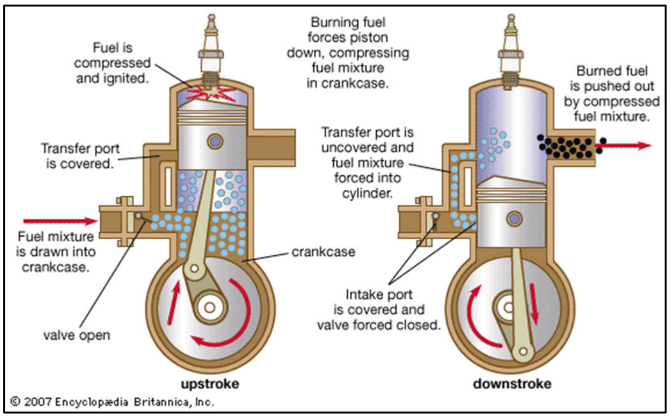


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Fig. III.3. Two-stroke engine

III.2.1.3. Compression rate

Corresponds to the ratio of pressures at the end PB and at the start PA of the compression phase :

$$\varepsilon = \frac{P_B}{P_A} \qquad \text{(III.3)}$$

In the context of perfect adiabatic compression, we can write :

$$\varepsilon_v = \left(\frac{P_B}{P_A}\right)^{\gamma} = \frac{V_A}{V_B} \tag{III.4}$$

III.2.1.4. Thermodynamic efficiency

The theoretical thermodynamic efficiency δ of a spark ignition engine is the ratio of the mechanical work provided by the engine during the expansion phase to the thermal input received during the fuel combustion phase. This yield is determined by the formula:

$$\varepsilon_v = \left(\frac{P_B}{P_A}\right)^{\gamma} = \frac{V_A}{V_B} \tag{III.4}$$

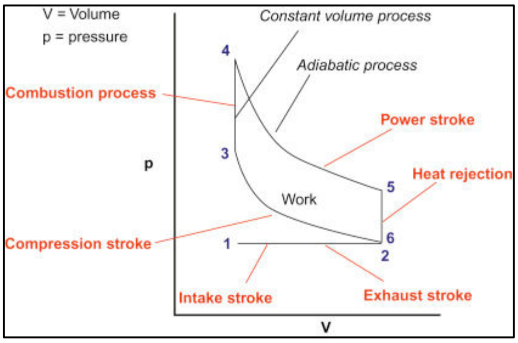


Fig. III.4. Spark-ignition engine cycle

III.3. Diesel cycle

Theoretically, the thermodynamic cycle of the Diesel engine differs from the spark-ignition engine: in the Diesel engine, combustion takes place at constant pressure, and not at constant volume. Most Diesel engines are also four-stroke engines. Unlike spark-ignition engines, Diesel engines do not have a carburetor or an ignition system per se. The first stage is the intake: the cylinder sucks in pure air - whereas it is a mixture of air and gasoline in the spark ignition engine - through the intake valve. During the second stage, or compression, the air is compressed, which brings it to approximately 440°C. At the end of the compression time, the vaporized fuel is injected under high pressure into the combustion chamber and burns instantly, the air temperature being very high in the combustion chamber. Thus, unlike the spark-ignition engine, the gas mixture ignites here by itself.

III.3.1. Thermodynamic efficiency

The efficiency of the cycle being the ratio of useful energy to energy spent:

$$\delta = \frac{\text{travail du cycle}}{\text{chaleur de combustion}} = \frac{W_{cycle}}{Q_C} = \frac{\oint TdS}{Q_C} \tag{III.6}$$

(1-2) isentropic transformation $S_1 = S_2$

(3-4) isentropic transformation $S_3 = S_4$

(1-4) isochoric transformation $V_1 = V_4$

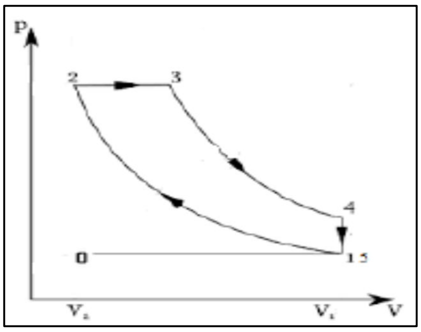


Fig. III.5. Diesel Cycle

$\frac{V_3}{V_2} = \tau, \frac{V_2}{V_1} = \varepsilon$

$$\delta_{Diesel} = \frac{\int_1^2 TdS + \int_2^3 TdS + \int_3^4 TdS + \int_4^1 TdS}{\int_2^3 TdS}$$

$$= \frac{\int_2^3 TdS + \int_4^1 TdS}{\int_2^3 TdS} = 1 - \frac{\int_4^1 TdS}{\int_2^3 TdS} \qquad \text{(III.7)}$$

$$\rightarrow \qquad \delta_{Diesel} = 1 - \frac{\int_1^4 C_v dT + \int_1^4 pdV}{\int_2^3 C_p dT - \int_2^3 Vdp} = \frac{\int_1^4 C_v dT}{\int_2^3 C_p dT}$$

$$\delta_{Diesel} = 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{(\frac{T_4}{T_1} - 1)}{\frac{\gamma T_2}{T_1}(\frac{T_3}{T_2} - 1)} = 1 - \frac{1}{\varepsilon^{\gamma - 1}} \cdot \frac{(\tau)^{\gamma} - 1}{\gamma\,(\tau - 1)}$$

(III.8)

If the gas is considered perfect : $p_1V_1 = RT_1$; $p_2V_2 = RT_2$; $p_3V_3 = RT_3$; $p_4V_4 = RT_4$

$\rightarrow \frac{T_4}{T_1} = \frac{p_4}{p_1} ; \frac{T_3}{T_2} = \frac{V_3}{V_2} = \tau$ et $pV^{\gamma} = cte$

$\rightarrow \frac{T_4}{T_1} = \left(\frac{V_3}{V_2}\right)^{\gamma}$; $\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} . \frac{V_2}{V_1} = \varepsilon^{\gamma - 1}$ donc $\delta_{Diesel} = f(\tau, \varepsilon, \gamma)$

ε: Compression rate

τ : Introduction rate

III.4. OTTO cycle (Beau de Rauchas)

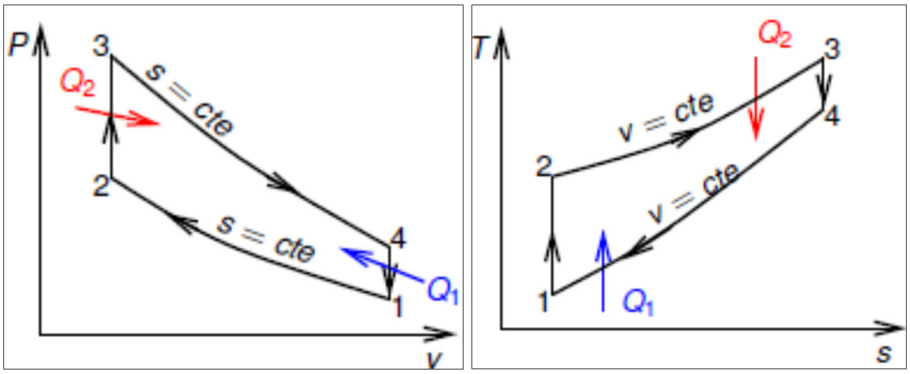


Fig. III.6. Otto cycle (Clapeyron and entropic diagram)

(1-2) Isentropic compression

(2-3) Heat supplied to V = cte

(3-4) Isentropic expansion

(4-1) Heat given up at V = cte

- The thermodynamic system is the fluid included in the cylinder during the compression and expansion phases. For a unit of mass and for each step we have :

$$U = C_v\,\Delta T = Q + W \qquad \text{(III.9)}$$

$U_2 - U_1 = Cv\,(T_2 - T_1) = W_{21}$
$U_3 - U_2 = Cv\,(T_3 - T_2) = Q_2$
$U_4 - U_3 = Cv\,(T_4 - T_3) = W_{43}$

$$U_1 - U_4 = Cv\,(T_1 - T_4) = Q_1$$

On the cycle :

$$0 = W + Q_1 + Q_2$$

$$U = C_v\,\Delta T = Q + W$$

- Transformations (1-2) et (3-4) are isentropic

$$S_2 - S_1 = 0$$

$$S_4 - S_3 = 0$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \frac{T_1}{T_2} \qquad \text{(III.10)}$$

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

$$\eta = \frac{-W}{Q_2} = 1 + \frac{Q_1}{Q_2} = 1 + \frac{(T_1 - T_4)}{(T_3 - T_2)} = 1 - \frac{T_1}{T_2}\frac{\frac{T_4}{T_1}-1}{\frac{T_3}{T_2}-1} \qquad \text{(III.11)}$$

$$\eta = 1 - \frac{1}{\left(\frac{V_1}{V_2}\right)^{\gamma-1}} \qquad \text{(III.12)}$$

Efficiency depends only on the compression ratio (V_1/V_2) and the heat capacity ratio : efficiency increases according to the compression ratio increase.

This conclusion also applies to real combustion engines.

III.5. Mixed cycle

This cycle is characterized by the evolution of a gaseous mass m which we assimilate in a first approximation to an ideal gas and of which we admit that the escape into the atmosphere and the subsequent admission are equivalent to cooling at constant volume. The cycle is composed of processes different from that of Diesel.

III.5.1. Thermodynamic efficiency

The efficiency of the cycle being the ratio of the useful energy to the energy spent (see equation (III.6) :

(1-2), (4-5) : isentropic transformation $S_1 = S_2$; $S_3 = S_4$

(1-5), (2-3) : isochoric transformation $V_1 = V_5$; $V_2 = V_3$

(3-4) : isobaric transformation $P_3 = P_4$

$$\frac{V_4}{V_3} = \tau' \;;\; \frac{p_3}{p_2} = \sigma \;;\; \frac{V_2}{V_1} = \varepsilon$$

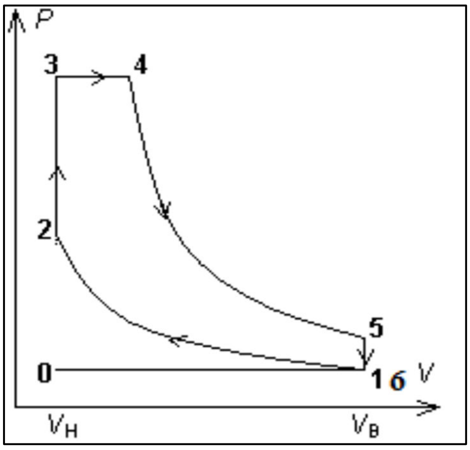


Fig. III.7. Mixed cycle

$$\delta_{mixte} = \frac{\int_1^2 TdS + \int_2^3 TdS + \int_3^4 TdS + \int_4^5 TdS + \int_5^1 TdS}{\int_2^3 TdS + \int_3^4 TdS}$$

$$= \frac{\int_2^3 TdS + \int_3^4 TdS + \int_5^1 TdS}{\int_2^3 TdS + \int_3^4 TdS} = 1 - \frac{\int_1^5 TdS}{\int_2^3 TdS + \int_3^4 TdS} \quad \text{(III.13)}$$

$$\delta_{mixte} = 1 - \frac{\int_1^5 C_v dT + \int_1^5 pdV}{\int_2^3 C_v dT + \int_2^3 pdV + \int_3^4 C_p dT - \int_3^4 Vdp}$$

$$\delta_{mixte} = 1 - \frac{\int_1^5 C_v dT}{\int_2^3 C_v dT + \int_3^4 C_p dT} = 1 - \frac{C_v(T_5 - T_1)}{C_v(T_3 - T_2) + C_p(T_4 - T_3)} \quad \text{(III.14)}$$

By continuing the same operations as the Diesel cycle, we find the efficiency of the mixed cycle as follows:

$$\delta_{mixte} = 1 - \frac{\sigma \tau'^{\gamma}}{\varepsilon^{\gamma-1}[(\sigma-1) + \gamma\sigma(\tau'-1)]} \quad \text{(III.15)}$$

The mixed cycle efficiency is depending on compression ratio ε and gas introduction rate τ'.

CHAPTER 4

Gas turbine and turbojet

IV.1. General concepts

IV.1.1. Turbomachines

Turbomachines, i.e. engines operating with a turbine (sometimes simply called “gas turbines”), have two major advantages over their piston-cylinder counterparts :

The power-to-weight ratio of turbomachines is approximately three times greater. When high powers are required with space or weight constraints, turbomachines are essential.

In the case of aeronautical propulsion, the working fluid can be used as the propulsion medium itself. It useful let the air go out of the turbine with residual pressure and let it relax in a nozzle. We then obtain a thrust by reaction (equal to the mass flow multiplied by its speed) : this is the principle of the turbojet.

Among the associated disadvantages, we will notice that the efficiency and reactivity of turbomachines drop very quickly at low power. Indeed, at partial load, the compression ratio and isentropic efficiency of turbines and compressors collapse.

Turbomachines are therefore used when high powers are required on a sustained basis. For example, the automotive sector, where power variations are multiple and must be recorded instantly.

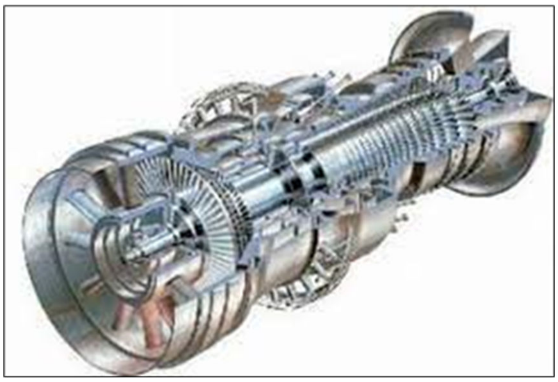


Fig. IV.1. Turbine à gaz

Fig. IV.2. Turbojet

IV.1.1.1. Turbojets

It has a diffuser section at inlet for realizing some compression of air passing through this section. Due to this air reaching compressor section has pressure more than ambient pressure. This action of partly compressing air by passing it through diffuser section is called “ramming action” or “ram effect”. Subsequently compressor section compresses air which is fed to combustion chamber and fuel is added to it for causing combustion. Combustion products available at high pressure and temperature are then passed through turbine and expanded there. Thus, turbine yields positive work which is used for driving compressor.

Expanding gases leaving turbine are passed through exit nozzle where it is further expanded and results in high velocity jet at exit. This high velocity jet leaving nozzle is responsible for getting desired thrust for propulsion.The mass flow of air entering the fan is:

$$\emptyset_m = \emptyset_T + \emptyset_e \qquad (IV.1)$$

$\emptyset_e$ Mass flow of air passing through the conventional engine unit.

$\emptyset_T$ Mass flow of air that is diverted into the fan nozzle.

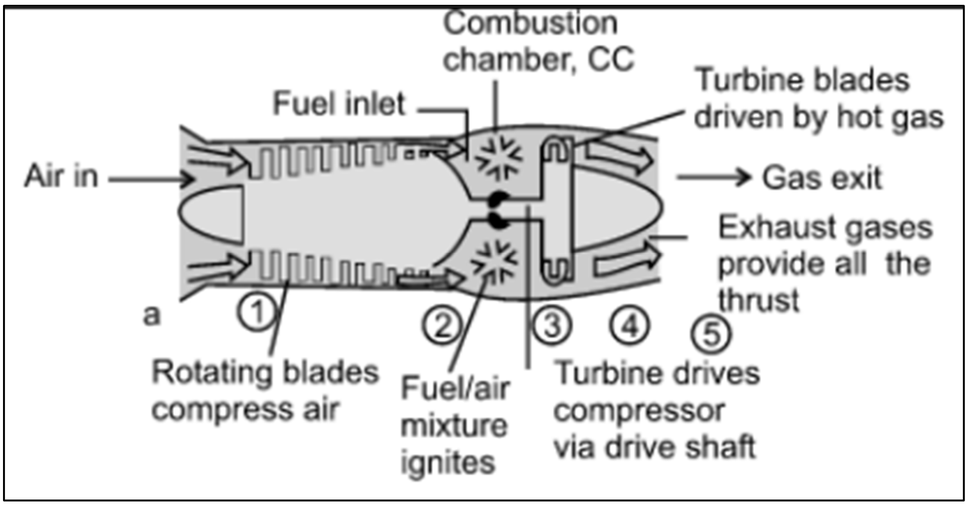


Fig. IV.3. Turbojet operating principle

IV.1.1.2. Gas turbine

IV.1.1.2.1. Description and operating principle

The gas turbine is a power plant in which a gas mixture is compressed by a turbocharger before being brought to high temperature in an almost isobaric manner, then expanded in a turbine. In its most widespread version, the gas turbine draws in atmospheric air which is compressed, then transformed into high temperature fumes by combustion of a fuel in a suitable chamber. These fumes are then expanded in the turbine, and finally evacuated into the atmosphere.

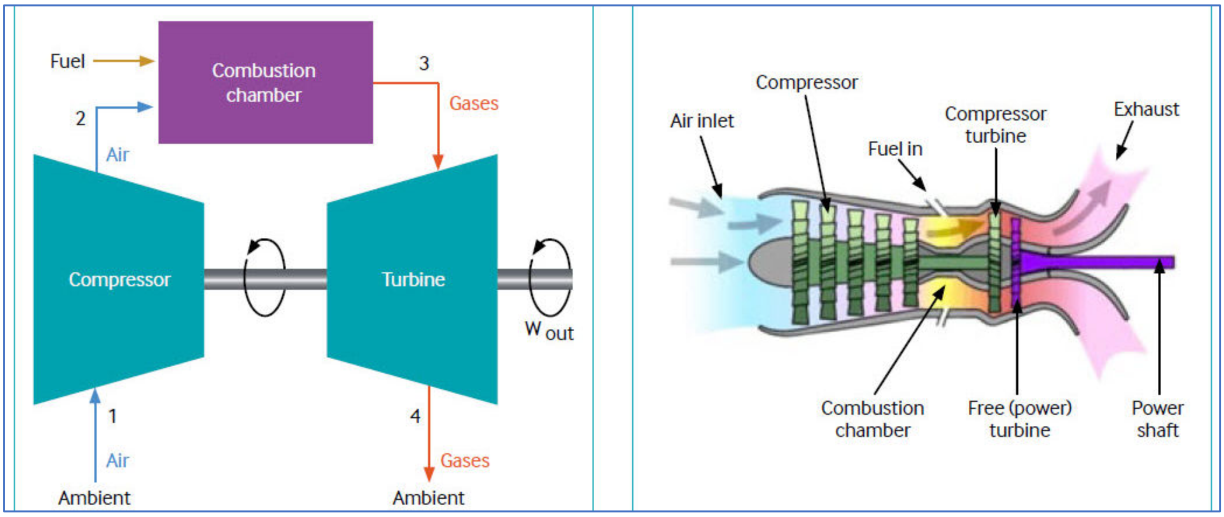


Fig. IV.4. Gaz turbine Operating principle

The thermal expansion of the gases due to the effect of the hot source gives rise to the production of an expansion motive power greater than that necessary for the compression of the fresh gas. The excess power of the turbine over all the powers drawn by the compressor, dissipated by mechanical friction (bearings and stops) and consumed by the auxiliaries (fuel pump, lubrication, cooling blower, etc.), constitutes the effective power P_e of this power plant :

$$P_e = P_{mT} - P_{mC} + P_{fm+aux} \tag{IV.2}$$

IV.1.1.2.2. Gas turbine components :

a. Compressors

Most compressors are axial, that is to say the air passes through them parallel to the rotation axis, but centrifugal compressors are sometimes used to project the air radially; whatever the process used, the thermodynamic evolutions of the air remain identical.

We quantify the efficiency of a compressor by comparing its power with that of an ideal compressor (a compressor that would be isentropic). The isentropic efficiency of the compressor can be defined based on the isentropic power for a constant air flow and the actual power consumed by the compressor:

$$\eta_C = \frac{P_{isentropique}}{P_{réelle}} \tag{IV.3}$$

As a turbine, the efficiency of a compressor is always less than 1. If this efficiency is known, we can compare the real properties of the air at the inlet and outlet of the compressor with those that the In the ideal case we would measure:

$$P_C = m.C_p.(T_{B\,r} - T_A) = \frac{1}{\eta_C}(T_B - T_A) \tag{IV.4}$$

Where :

$T_{B\,r}$, T_B : Real and Ideal Temperatures at the compressor outlet

T_A : Ambient temperature

b. Combustion chamber:

The heat input from turbomachines takes place in one or more combustion chambers. The air is heated there at constant pressure by combustion ; its temperature and specific volume increase sharply. No work is done in the combustion chamber, and the pressure remains approximately constant.

The power produced by the combustion chamber is defined as a function of the incoming air temperature T_A and the gas burned at the outlet T_B :

$$P_C = H_B - H_A = C_{p\,gaz}T_B - C_{p\,air}T_A \tag{IV.5}$$

c. Turbine :

The role of the turbine is to power the compressor: it must therefore extract sufficient power from the air to operate the latter and compensate for possible transmission losses. Depending on the configuration of the turbomachine, the turbine can then extract further energy to power other components.

Its performance is defined by :

$$\eta_T = \frac{P_{réelle}}{P_{isentropique}} \tag{IV.6}$$

The power extracted by the turbine is expressed as a function of the temperatures (real and ideal) at the turbine outlet:

$$P_T = C_{p\,gaz}.(T_{B\,r} - T_A) = \eta_T C_{p\,gaz}(T_B - T_A) \tag{IV.7}$$

IV.2. Basic cycle

IV.2.1. Ideal Joule-Brayton cycle

It is also called simplified theoretical cycle, the ideal form consists of two isobaric processes and two isentropic processes. The isobars include the TAG combustion chamber system and the exhaust side. The isentropics represent compression (compressor) and expansion (turbine expander).

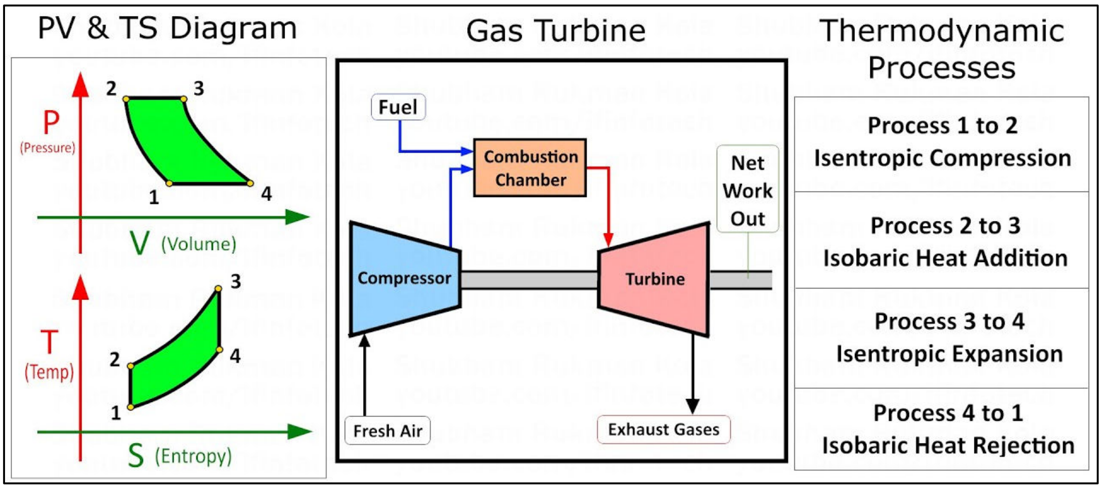


Fig. IV.5. Joule-Brayton cycle

The Brayton cycle is composed of four fundamental processes :

1 – 2 : Isentropic compression

2 – 3 : Heat supply at constant pressure (isobaric combustion)

3 – 4 : Isentropic expansion of gases in the turbine

4 – 1 : Exhaust of combustion gases to the atmosphere at constant pressure (exhaust and intake gases in the open cycle)

IV.2.1.1. Thermodynamic efficiency

By applying the first law of thermodynamics :

(1-2)

Compression work (isentropic) : $W_C = m.C_{p\,air}\,(T_2 - T_1)$ (IV.8)

Compression power : $P_C = \emptyset_{m\,air}\,(h_2 - h_1)$ (IV.9)

(2-3)

Heat provided by combustion (isobaric) : $Q_1 = Q_{23} = m.C_{p\,gaz}\,(T_3 - T_2)$ (IV.10)

(3-4)

Work recovered by turbine expansion (isentropic) : $W_T = m.C_{p\,gaz}\,(T_4 - T_3)$ (IV.11)

Turbine power : $P_T = (\emptyset_{m\,air} + \emptyset_{m\,gaz}).(h_4 - h_3)$ (IV.12)

(4-1)

Heat released during the exhaust of burnt gases : $Q_2 = Q_{41} = m.C_{p\ gaz}\ (T_4 - T_1)$ (IV.13)

- **Work of the cycle**

$$W_{cycle} = W_T - W_c \qquad \text{(IV.14)}$$

- **Thermal efficiency** :

$$\eta_{th} = \frac{W_{effectif}}{Q_{fournie}} = \frac{W_T - W_c}{Q_1} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \text{ (Isentropic transformation)} \qquad \text{(IV.15)}$$

IV.2.1.2. Disadvantages of the cycle

- Expenditure on compression work Wc compared to Wt
- Installed power higher than useful power (amplified effect by losses)

IV.2.1. Real Joule-Brayton cycle

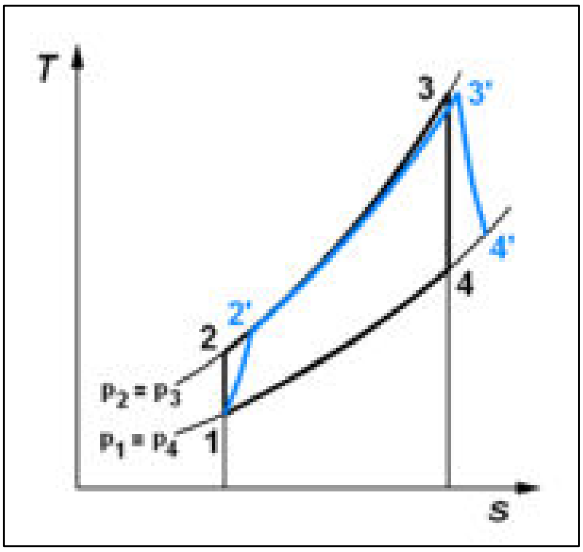


Fig. IV.6. Real Joule-Brayton cycle

- Irreversible cycle (1 2’ 3’ 4’ 1)

1 – 2’ : Irreversible adiabatic compression

2’ – 3’ : Heat supply at constant pressure (isobaric combustion + thermal loses in combustion chamber)

3’ – 4’ : Irreversible adiabatic expansion of the gases in the turbine

4’ - 1 : Exhaust of combustion gases to the atmosphere.

- **Compressor efficiency :**

$$\eta_C = \frac{h_{2^*} - h_1}{h_2 - h_1} = \frac{T_{2^*} - T_1}{T_2 - T_1} \qquad \text{(IV.16)}$$

- **Turbine efficiency :**

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4^*}} = \frac{T_3 - T_4}{T_3 - T_{4^*}} \qquad \text{(IV.17)}$$

- **Thermodynamic efficiency**

$$\eta_{th}=\frac{W_{effectif}}{Q_{fournie}}=\frac{C_{p\,gaz}\,(T_3-T_4)-C_{p\,air}\,(T_2-T_1)}{C_{p\,gaz}\,(T_3-T_2)} \qquad \text{(IV.18)}$$

After introducing the different effeciencies: ($C_{p\,gaz}=C_{p\,air}=cte$)

$$\eta_{th}=\frac{\eta_T(T_3-T_{4^*})-\frac{T_{2^*}-T_1}{\eta_C}}{(T_3-T_1)-\frac{T_{2^*}-T_1}{\eta_C}} \qquad \text{(IV.19)}$$

$$\eta_{th}=\frac{\eta_T\frac{T_3}{T_1}(1-\frac{T_{4^*}}{T_3})-\frac{\frac{T_{2^*}}{T_1}-1}{\eta_C}}{\left(\frac{T_3}{T_1}-1\right)-\frac{\frac{T_{2^*}}{T_1}-1}{\eta_C}} \qquad \text{(IV.20)}$$

IV.2.2. Recovery Joule-Brayton cycle

The cycle with recovery is characterized by the possibility of heating the air compressed by the gases escaped from the turbine before being introduced into the combustion chamber

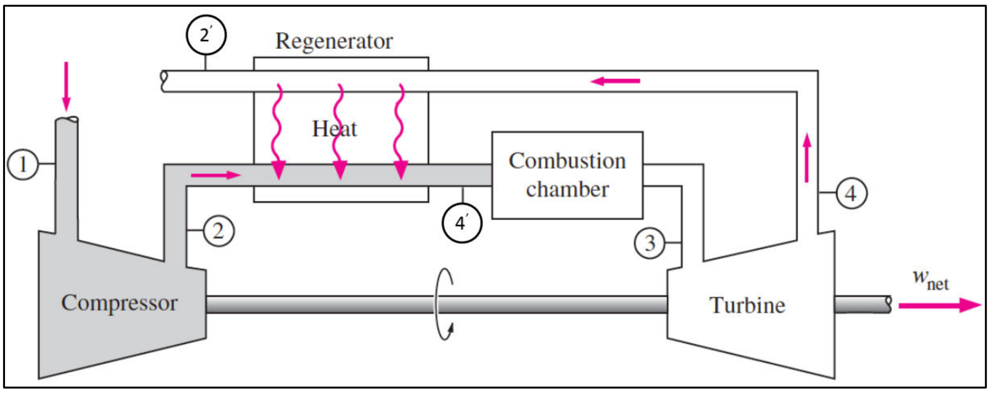


Fig. IV.7. Recovery gaz turbine setup

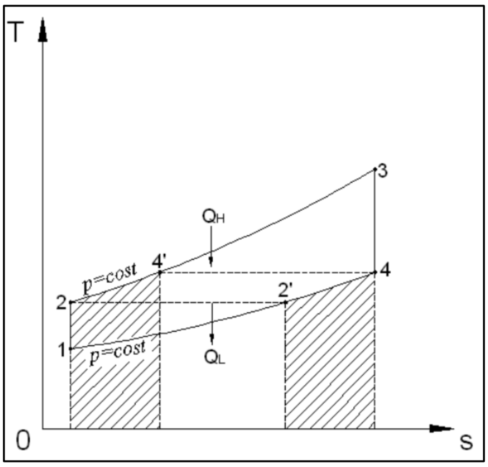


Fig. IV.8. Recovery gaz turbine cycle

$T_{2^*}=T_2$ and $T_{4^*}=T_4$ (perfect heat exchange and constant specific heat)

- **Thermodynamic efficiency**

$$\eta_{th}=\frac{C_p\,(T_3-T_4)-C_p\,(T_2-T_1)}{C_p\,(T_3-T_2)}=1-\frac{(T_4-T_1)}{(T_3-T_2)} \qquad \text{(IV.21)}$$

$$\eta_{th} = \frac{T_1}{T_3}\frac{\left(\frac{T_2}{T_1}-1\right)}{\left(1-\frac{T_4}{T_3}\right)} = \frac{T_1}{T_3}\frac{\left(\tau^{\frac{\gamma-1}{\gamma}}-1\right)}{\left(1-\frac{1}{\tau^{\frac{\gamma-1}{\gamma}}}\right)} \qquad \text{(IV.22)}$$

τ Compression rate $\tau = \frac{P_2}{P_1}$

IV.3.3. Multi-stage compression and expansion cycle

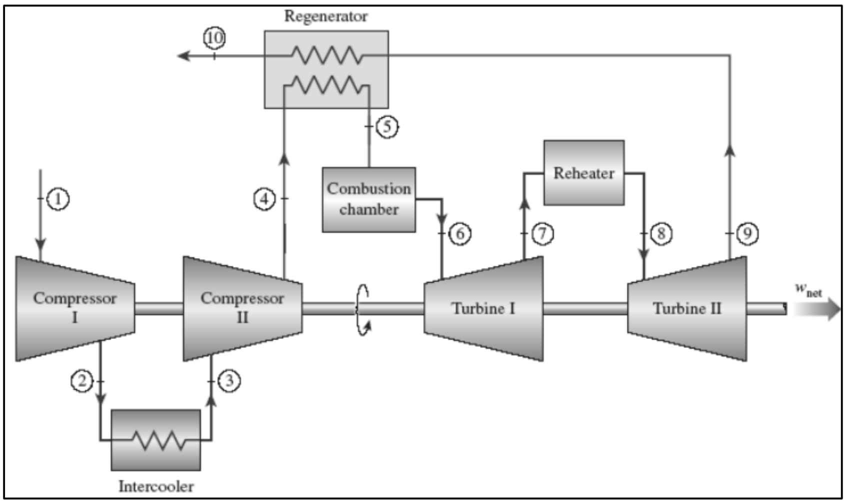


Fig. IV.10. Multi-stage compression and expansion GT with intermediate cooling and heating

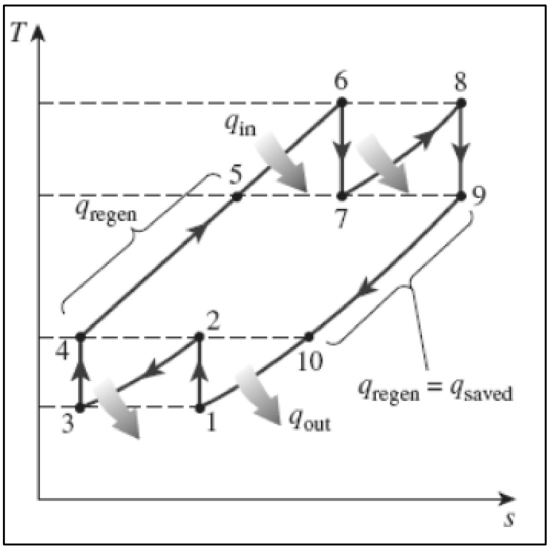


Fig. IV.11. Multi-stage compression and expansion GT cycle

- **Thermodynamic efficiency**
- $\eta_{Cycle} = \frac{W_{net}}{Q_{prov}} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_{10} - T_1}{T_6 - T_5}$ (IV.29)

IV.3.4. Combined cycle or cogeneration

In a combined cycle power plant, the hot gas stream leaving the gas turbine must be cooled by water from the steam recovery cycle. In a single pressure level cycle, this water enters the exchanger in the liquid state at approximately 30°C, after being compressed by the feed pumps located downstream of the condenser. It is then heated in the economizer to the boiling temperature corresponding to its pressure. It is then vaporized, at constant temperature, then superheated, before being expanded in the steam turbine.

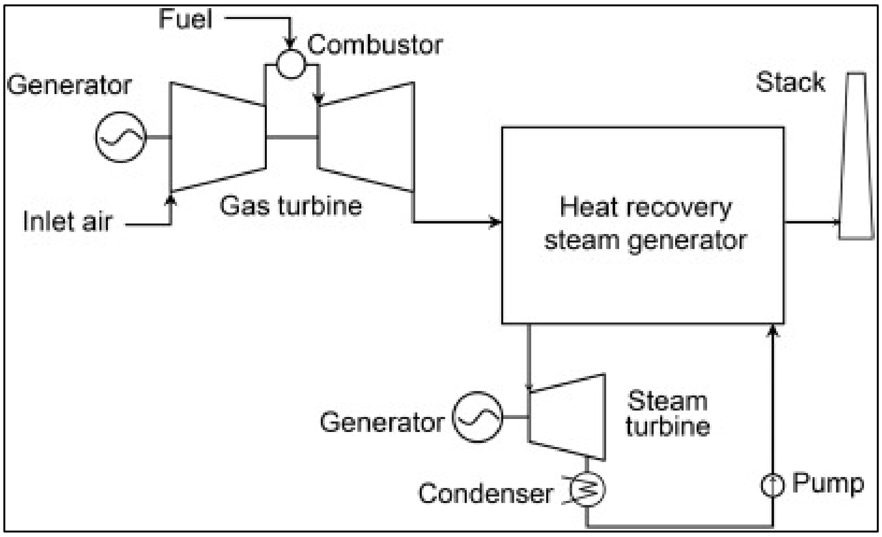


Fig. IV.9. Combined cycle (gas/steam)

IV.3.4.1. Combined electricity generation :

- Recovery of heat from hot gases at the turbine outlet to power a steam cycle.

- Combined generation of electricity and heat (efficiency > 60%)

- Recovery of heat from hot gases at the turbine outlet to supply an external water exchanger for the production of hot water.

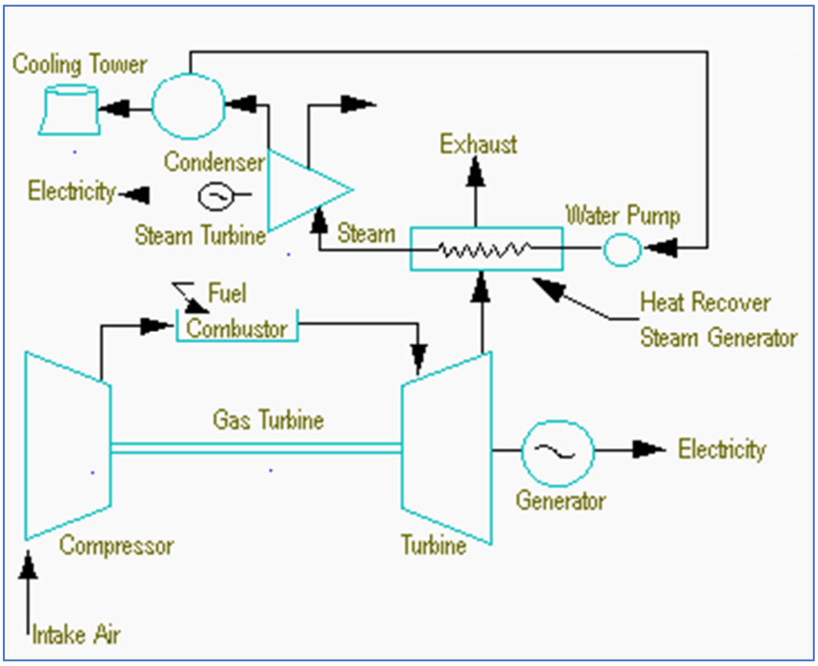


Fig. IV.10. Combined electricity generation plant

CHAPTER 5

Steam turbine

V.1. Phase change of a pure body (liquid-vapor)

When a pure body evolves from one state of equilibrium to another, we witness under certain conditions a significant and sudden modification of these optical, mechanical, electrical properties, etc.

We then say that it undergoes a change (transition) of state (phase). These phenomena are commonly observed, whether it is the change of liquid water into vapor or ice, but this definition also includes other phenomena such as the passage of iron! (centered cubic metallic network) with iron! (face-centered cubic metal lattice) at a temperature of 1176K under atmospheric pressure, it is then an allotropic variation.

V.1.1. Zone of existence of the different phases

We call a change of state or phase transition the crossing of an equilibrium curve of the P, T diagram.

During the phase change, the two states coexist: the state is two-phase.

A change in liquid-vapor state is explained qualitatively by an increasingly greater agitation or vibration of the molecules which end up overcoming the bonding forces ; if the surrounding pressure is high on the liquid, more agitation will therefore be required. a higher temperature. Entropy increases in a liquid-vapor transition.

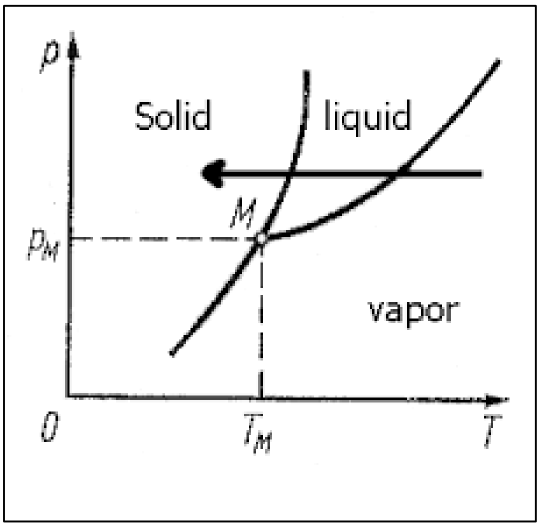


Fig. V.1. Liquid-steam transition

V.1.2. Equilibrium of a pure body under two phases

V.1.2.1. Condition and equilibrium curve

Consider a mass m of a pure body, undergoing a reversible evolution at constant T and p (isothermal and isobaric):

Let’s introduce the free enthalpy G = U+PV-TS

$$dG = dU+PdV+VdP-TdS-SdT=TdS-PdV+PdV+VdP-TdS-SdT \qquad (V.1)$$

$$dG=VdP-SdT, \text{ (G depends to T and P)} \qquad (V.2)$$

We will use this function for a system with two phases 1 and 2

The total free enthalpy of the system is an extensive function, therefore:

$$G = m\,x\,G_2 + m\,(1-x)\,G_1 \tag{V.3}$$

G_1 and G_2 the mass free enthalpies of phases 1 and 2 at the equilibrium temperature and pressure, m the total mass and x the mass fraction in phase 2.

For the coexistence of the two phases to be a state of equilibrium at T and P, G must reach a state of equilibrium, and no longer vary as a function of X : $G_1(T, P) = G_2(T, 2)$.

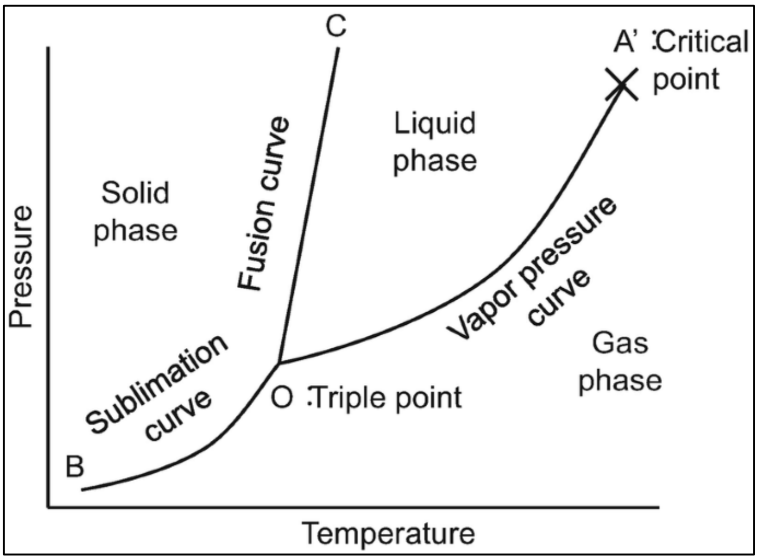


Fig. V.2. Phases equilibrium curve

V.1.2.2. Clapeyron formula

Consider a pure body under two phases in equilibrium. We have equality of mass enthalpies $G_1(T, P) = G_2(T, P)$. If we take another equilibrium state at T + dT and at P + dP

We have :

$$G_1(T + dT, P + dP) = G_2(T + dT, P + dP) \tag{V.4}$$

Now we have shown that $dG = VdP - S\,dT$

Either per mass unit

$$dG_1 = dG_2 \quad \text{Or} \quad -S_1\,dT + V_1\,dP = -S_2\,dT + V_2\,dP \tag{V.5}$$

$$\Delta S = S_2 - S_1 = (V_2 - V_1)\,dP/dT \tag{V.6}$$

V_2, V_1 and ΔS are mass quantities

$$H_{12} = L_{12} = T\,(V_2 - V_1)\,dP/dT \tag{V.7}$$

dP/dT is calculated along the T equilibrium curve.

Note : L_{12} and ΔS have the same sign, L_{12} is therefore positive when the disorder increases (for example for vaporization).

V.1.3. Steam title (steam content)

When we are in the presence of the liquid-vapor mixture, the pressure being a function of T, the state variables to use are T or p and x the mass proportion of vapor in the mixture, or even vapor content.

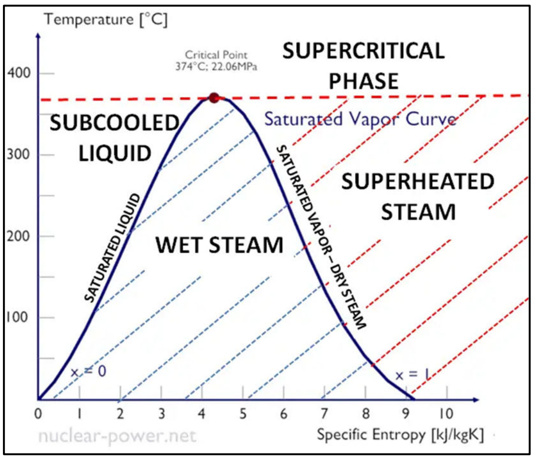


Fig. V.3. Steam title

Usual notation : *it is customary to use letters awarded once for the quantities relating to the saturating liquid and letters awarded twice for those of the saturated vapor. Lowercase letters are often reserved for mass quantities.*

m The total mass of the body in two phases:

$$m = m' + m'' \tag{V.8}$$

The vapor content is then X=m"/m

X will be expressed as a function of useful extensive variables, practically volume, enthalpy and entropy.

By analogy:

V=V'+V", for total volumes

In mass quantities V = mv = m'v'+ m"v"

By introducing x we obtain v = x v"+ (1-x) v'

We deduce that :

$$X = \frac{V-V'}{V''-V'} = \frac{h-h'}{h''-h'} = \frac{s-s'}{s''-s'} \tag{V.9}$$

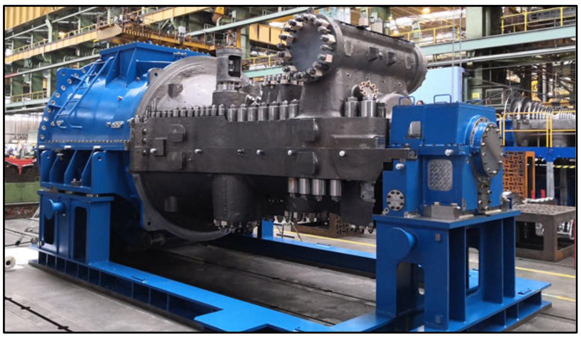
V.2.General concepts :

V.2.1. Operating principle of the steam turbine (ST)

The steam turbine is an external combustion heat engine, operating according to the so-called Clausius-Ranhime thermodynamic cycle. This cycle is distinguished by the change of state affecting the driving fluid which is generally water vapor. This cycle includes at least the following steps :

- Liquid water is compressed by a pump and sent to the boiler
- The steam expands in the turbine providing mechanical energy
- The expanded vapor is condensed on contact with the cold source under partial vacuum. The turbine constitutes an evolution exploiting the main advantages of turbomachines :

1. High specific power and density power
2. Improved efficiency by the multiplication of expansion stages



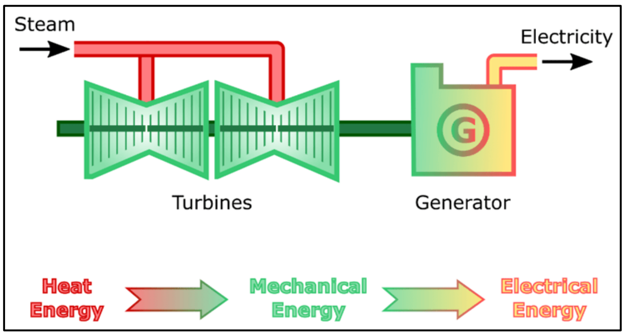


Fig. V.4. Industrial steam turbine **Fig. V.5.** Steam turbine setup

V.2.1. The main components of steam turbines

• Alternator

The alternator is an electrical machine of the alternating current generator type which transforms mechanical energy into electrical energy. It is driven by the turbine [3].

• Transformer

Main transformer (TP): The evacuation of the energy produced by the alternator is evacuated on the high voltage network through a main step-up transformer: 13800V/63000V, a 63 KV circuit breaker, three underground oil pressure cables and an overhead three-phase line. Withdrawal transformer (TS): The group's auxiliaries are supplied through a step-down withdrawal transformer (TS): 13800V/6300V in normal service and a step-down start-up transformer (TD): 63000V/6300V as emergency.

• Boiler

The role of the steam generator is to extract the heat energy from the fuel to transfer it to the water and produce steam at fixed parameters. It constitutes the hot source of the thermodynamic cycle. This steam will be used by the turbine to provide mechanical energy.

• Condenser

To maximize the efficiency of the steam turbine, the pressure and temperature of the steam outlet must be as low as possible. To do this, the steam leaving the turbine is directed to the condenser where it is cooled and condensed. The condenser is a heat exchanger with thousands of tubes through which the water in the cooling circuit circulates. The steam circulates over the tubes and condenses on contact with them. The water in the cooling circuit then extracts heat from the steam.

• Feed pump

The KSB very high pressure pump is a multistage centrifugal pump. It includes a suction body, a discharge body and a certain number of stages or cells assembled by tie rods. The water, coming from the food tank at the pump, has a pressure energy and a kinetic energy which will be increased in the moving turbines to supply the steam generator (boiler) with the necessary quantity of water to maintain the normal level.

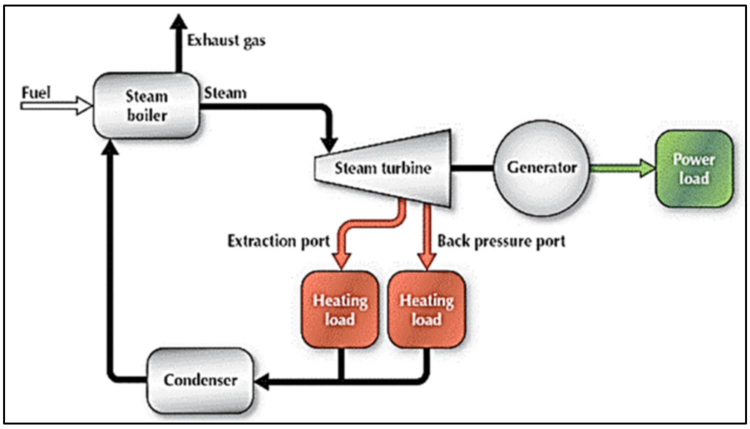


Fig. V.6. Steam turbine scheme

V.3. Rankine cycle without superheating

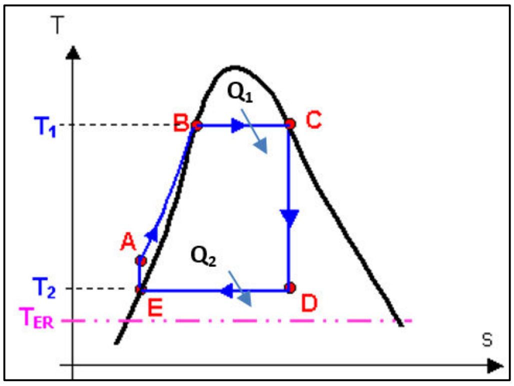


Fig. V.6. Rankine cycle without superheating

V.3.1. Study of the different transformations :

V.3.1.1. Compression work

The application of the first law of thermodynamics allows us to write :

$$dh = dQ + dW + \cancel{K} + \cancel{gz} \qquad dQ=0; \qquad dW=Vdp \tag{V.10}$$

On the other side $dh = TdS + dW$

$$\longrightarrow \quad dh = Vdp \tag{V.11}$$

By making the approximation « the mass volume of the water remains constant over the pressure interval considered »

$$\Delta h=h_A-h_E=Vdp= V\int_{p_E}^{p_A} dp = V\,(p_A - p_E) > 0 \tag{V.12}$$

V.3.1.2. Heat transfer in the steam generator (boiler)

We break down the AC transformation into two steps : isobaric of the liquid then vaporization at pressure $P_A = P_B = P_C$

Calculation of the heat received during the isobaric heating of water (AB) :

Applying the first principle allows us to write:

dh = dQ + dW; dW=0 (since there are no moving parts in the Steam generator SG)

$$dh = dQ = C_w .\ dT \tag{V.13}$$

Assuming that Cw (as a first approximation) is independent of the temperature and after integration: $\Delta h = h_B - h_A = C_w\,(T_B - T_A)$ (V.14)

Calculation of the enthalpy of vaporization of water (BC) :

$dh = dQ = dm\ L_{vap}(T_B) = Lvap\ (T_B)$ (Lvap (T): enthalpy of vaporization at the temperature of the phase change).

$$\Delta h = h_C - h_B = L_{vap}(T_B) \tag{V.15}$$

➡ The unit mass of water receives thermal energy during its heating and then its vaporization: $Q_{hot} = Q_1 = h_C - h_A$ (if Q_1 increases the efficiency of the cycle decreases)

V.3.1.3. Isentropic expansion in the turbine

dh = dQ + dW; dQ=0 (isentropic transformation) $\Delta h = h_D - h_C < 0$

h_D can be determined from the saturation curves or calculated on average by knowing the quantities from the thermodynamic tables:

Entropy is an extensive quantity which allows the following relation to be written at point D :

$$S_D = X\,S'' + (1-X)\,S' \quad X = \frac{s_D - s_{liq}}{s_{vap} - s_{liq}} \quad s_D = s_C \text{ (Isentropic transformation)} \tag{V.16}$$

With S'' and S' are respectively the mass entropies of the saturated vapor and the saturation liquid at the end of expansion pressure and and X the title of mixture (liquid-vapor) Which gives the enthalpy h_D:

$$h_D = X\,h'' + (1-X)\,h' \tag{V.17}$$

V.3.1.4. Heat transfer by condenser

Quantity of heat exchanged at the condenser (DE) :

Hypothesis : $P_D = P_E$ isobaric transformation

Applying the first principle allows us to write:

dh = dQ + dW; dW=0 (isobaric transformation) and $\Delta h = h_D - h_E < 0$

V.3.2. Thermal efficiency

Thermal efficiency is defined as the ratio between useful energy and the total energy received by the system (1 kg of water). For the studied setup, during a cycle :

The mass work recovered during the expansion of the steam in the turbine constitutes the useful energy.

The thermal energy received by 1 kg of water at the SG and the specific work received by the water during its compression constitute the total energy received, allows us to write:

$$\eta_{th} = \frac{\text{turbine work} - \text{pump work}}{\text{heat received}} = \frac{(h_c - h_D) - (h_A - h_E)}{h_C - h_A} \tag{V.18}$$

According to the 1st Principle $(W + Q_1 + Q_2)_{cycle} = 0$; $w_e = -W = Q_1 + Q_2$

$$\eta_{th} = \frac{W_e}{Q_1} = \frac{Q_1 + Q_2}{Q1} = 1 + \frac{Q_2}{Q_1} \quad ; \quad Q_2 < 0 \tag{V.19}$$

V.4. Rankine cycle with suprheating (Hirn)

In the case of the RANKINE cycle, we can see that at the end of the relaxation, the operating point is inside the saturation curve. This shows that at the end of the expansion, liquefaction begins and drops of liquid water form. This can seriously damage the turbine wheel. It will be beneficial to eliminate this formation of liquid water during expansion. To do this, we can increase the temperature of the steam by superheating it.

The isobaric superheat of the saturated steam is the transformation CC', The steam at point C is called dry steam.

The benefit of overheating is to increase the efficiency of the installation. The work recovered during the expansion of the superheated steam compared to the expenditure in thermal energy mainly is more important for the Hirn cycle than for the Rankine cycle.

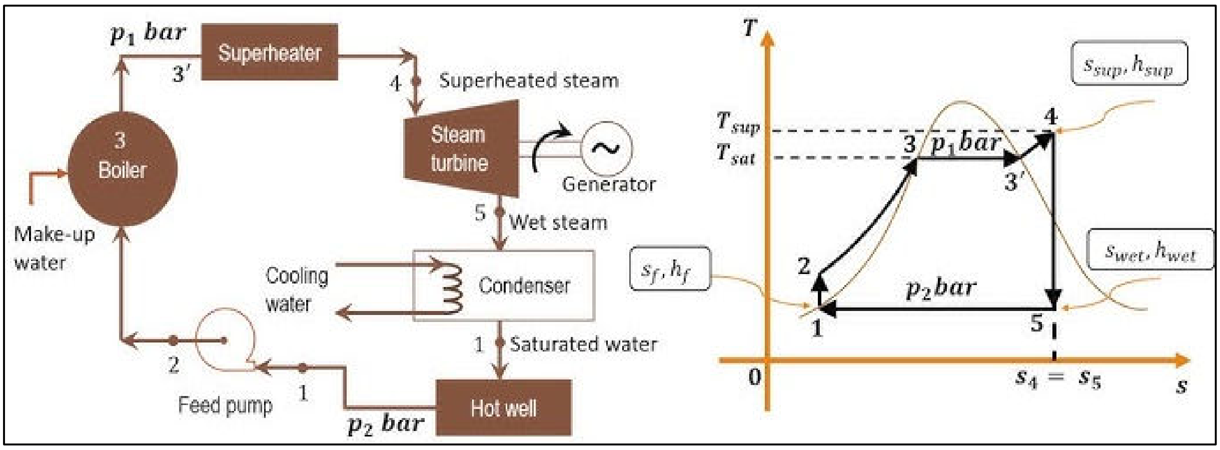


Fig. V.8. Steam turbine with superheated cycle (Cycle of Hirn)

V.4.1. Thermal efficiency

$$\eta_{th} = \frac{W_T}{Q} = \frac{|W_{45}|}{Q_{24} + W_{12}} = \frac{|h_5 - h_4|}{(h_4 - h_2) + (h_2 - h_1)} \tag{V.20}$$

$W_p = W_{12} = h_2 - h_1$ (Neglected)

$$\eta_{th} = \frac{h_5 - h_4}{(h_4 - h_2)} \tag{V.21}$$

V.5. Withdrawal Hirn cycle

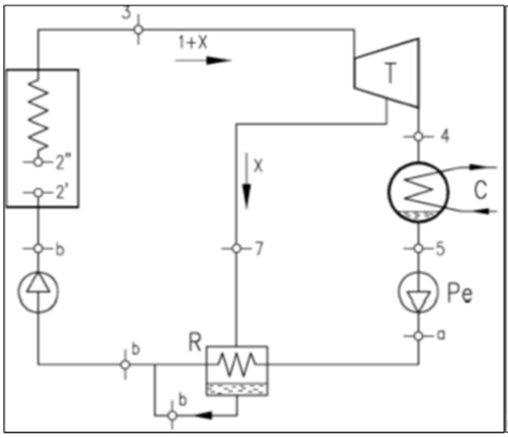


Fig. V.9. Withdrawal steam turbine

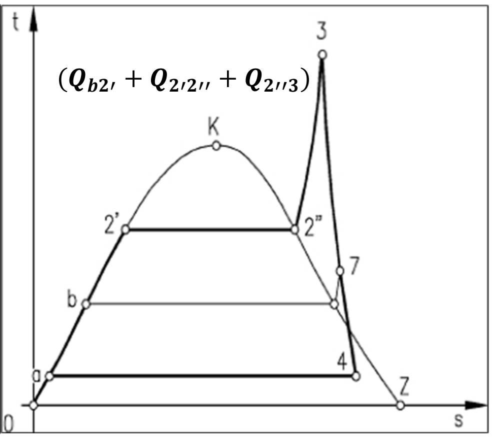


Fig. V.10. Withdrawal Hirn cycle

Withdrawal serves to increase the flow of live steam by a fraction X, the residual heat is used to heat the flow leaving the condenser.

(3-4) Expansion work: $W_{34} = (h_3 - h_4)$ (V.22)

(4-5) Isothermal and isobaric condensation (steam/liquid)

(5-a) Water pumping $h_5 = h_a$

(a-b) Heating of the water in the mixer in contact with the withdrawn fraction X hypothesis: point b is on the saturation curve

(3-7) Expansion of the fraction x of steam, from HP to MP, in the turbine.

$$W_{37} = (h_3 - h_7) \tag{V.23}$$

(7-b) Condensation of the fraction

(b-b) pumping the quantity (1 + x) of liquid water, transition from MP to HP (the points are combined and located on the saturation curve).

(b-2') Isobaric heating of the quantity (1 + X) of water

$$Q_{b2'} = (h_{2'} - h_b) \tag{V.24}$$

(2'-2") Isobaric and isothermal evaporation of the quantity (1 + X) in the GV.

$$Q_{2'2''} = (h_{2'} - h_{2''}) \tag{V.25}$$

(2"-3) isobaric heating of the quantity (1 + X) of steam.

$$Q_{2''3} = (h_3 - h_{2''}) \tag{V.26}$$

V.5.1. Thermal efficiency

$$\eta_{th} = \frac{W_{37} + W_{34}}{Q_{b2'} + Q_{2'2''} + Q_{2''3} + W_{p1} + W_{p2}} \approx \frac{(h_3 - h_4) + X(h_3 - h_7)}{(1 + X)(h_3 - b)} \tag{V.27}$$

$$Q_1 = Q_{b2'} + Q_{2'2''} + Q_{2''3} + W_{p1}$$

V-6 Resuperheated Rankine cycle (double superheating)

The thermal efficiency of the superheated Rankine cycle can be further increased by reheating (7-8 in Figure V-11) the working fluid in a reheater (i.e. double superheating). And therefore, compared to the overheating Rankine cycle, this cycle includes an additional isobaric transformation 7-8 of supply of a fourth quantity of heat $Q7\text{-}8$ in the reheater and the expansion is done in two stages in two bodies of different turbine (1-7 in the THP high pressure turbine and 8-9 in the TBP low pressure turbine). Reheating also leads to an increase in the expansion work and therefore the useful work and the improvement of the working conditions of the turbines and therefore their internal efficiency since a large part of their stages operate with dry steam.

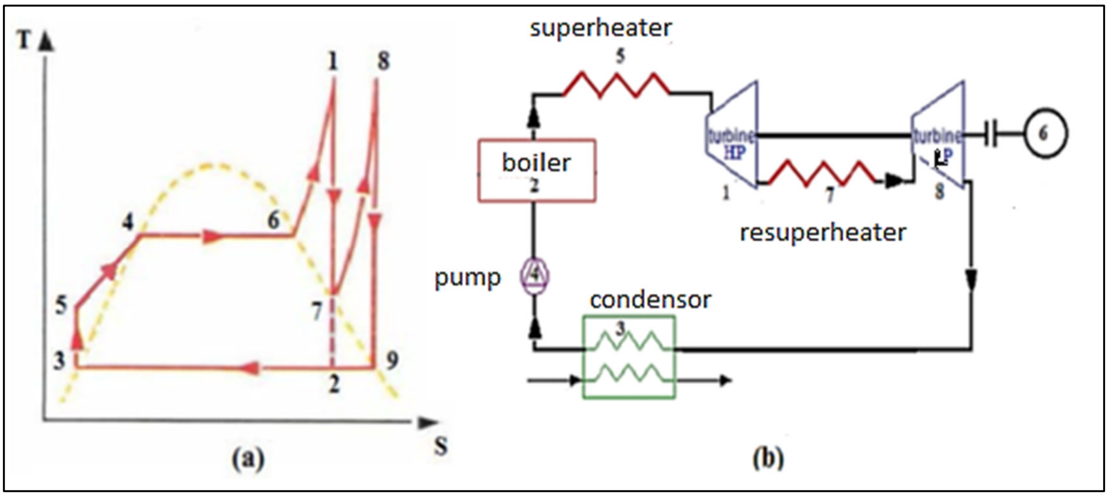


Fig. V.9. (a) Resuperheated Rankine cycle (double superheating), (**b)** Resuperheated Rankine cycle setup

The total quantity of the economizer Q_1, supplied to the motor fluid completely at constant pressure, will increase and will then be composed of a first quantity $Q_{5\text{-}4}$ (with increase in the temperature in the economizer) to bring it to the state of saturated liquid (X_4=0) with an increase in temperature followed by a second $Q_{4\text{-}6}$ to evaporate it at constant temperature ($T_4 = T_1$) until the saturation state (X_1=1), then a third $Q_{6\text{-}1}$ to superheat the saturated steam to $T_1 > T_6 = T_{sat}$ and finally a fourth $Q_{6\text{-}1}$ to reheat the saturated steam to $T_8 > T_6$ =Tsat (and $T_8 > T_7$), and therefore,

$$Q_1 = Q_{5\text{-}4} + Q_{4\text{-}6} + Q_{6\text{-}1} + Q_{7\text{-}8} = Q_{5\text{-}1} + Q_{7\text{-}8} \qquad \text{(V-28)}$$

$$Q_1 = (H_4 - H_5) + (H_6 - H_4) + (H_1 - H_6) + (H_8 - H_7) \qquad \text{(V-29)}$$

$$= (H_1 - H_5) + (H_8 - H_7) \qquad \text{(V-30)}$$

CHAPTER 6

Other engines types

VI.1. Cycle of Stirling engine

VI.1.1. General concepts

The Stirling cycle is a thermodynamic closed cycle invented in 1816 by the Scottish minister Robert Stirling. It was used as an engine and was considered at the time to be capable of replacing the steam engine since steam boilers were prone to life-threatening explosions. The counterpart of the Stirling engine, the refrigerator, was first recognized in 1832. Both machines experienced highs and lows during the nineteenth century. The principle behind the machines was almost condemned to obscurity after the invention of the internal combustion engine (gas-, petrol-, and diesel motors) and compressor refrigerators with external evaporation.

In 1938 the famous Dutch Philips Research Laboratory was looking for a means to power electricity generators for short wave communication systems in remote areas without electricity supply. The practically-forgotten Stirling engine attracted their attention.

In 1946 Philips started optimizing the Stirling cycle to be used for cryogenic cooling. The result was the development of the world conquering Stirling Cryogenerator, marking the start of significant cryogenic activities at Philips. Though the Stirling engine itself never became a commercial success, the Stirling Cryogenerator has been selling by thousands worldwide and has been incorporated in equipment and projects used from Antarctica to the North Pole.

VI.1.2 Operating principle

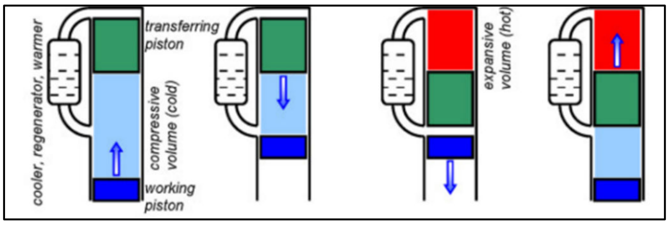


Fig. VI.1. Stirling engine processus

VI.1.3. Stirling cycle

In Stirling cycle, Carnot cycle's compression and expansion isentropic processes are replaced by two constant-volume regeneration processus.

During the regeneration process heat is transferred to a thermal storage device (regenerator) during one part and is transferred back to the working fluid in another part of the cycle.

The regenerator can be a wire or a ceramic mesh or any kind of porous plug with a high thermal mass (mass times specific heat). The regenerator is assumed to be reversible heat transfer device.

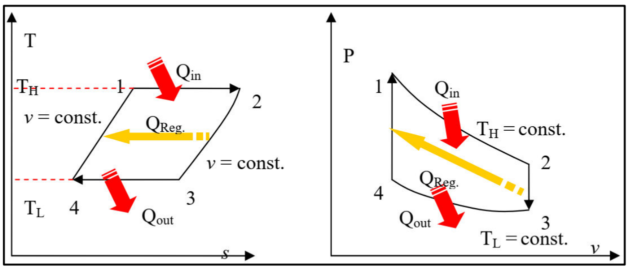


Fig. VI.2. Diagrams of Stirling cycle

1-2 Isothermal expansion : Heat addition from external source

2-3 Isochoric heat transfer : Internal heat transfer from the gas to the regenerator

3-4 Isothermal compression : Heat rejection to the external sink

4-1 Isochoric heat transfer : Internal heat transfer from the regenerator to the gas.

The only difference is that during the processes 4 → 1 and the 2 → 3, instead of the adiabatic processes, they have made it as the constant volume processes (Isochoric).

From the above T-S diagram of the Stirling cycle, it is observed that during the constant volume processes, the heat added to the system from the source and the heat rejected to sink both are same. ($T_3 = T_4$ and $T_2 = T_1$ respectively).

VI.1.3. Applications of Stirling Cycle

Stirling cycle garners widespread applications in the following areas :

- Electricity generation in remote areas
- Power source for deep space missions
- Cryogenic refrigeration for research and medical purposes
- Solar energy conversion
- Waste heat recovery in industries
- Home heating and cooling systems
- Water pumping in remote or off-grid locations.

VI.1.4. Thermal effeciency of Stirling Cycle

Work output : $W_E = [RT_1 \log(\frac{V2}{V1}) + C_V (T_1 - T_4)] - [C_V (T_2 - T_3) + RT_3 \log(\frac{V3}{V4})]$

Heat input : $Q_H = RT_1 \log(\frac{V2}{V1}) + C_V (T_1 - T_4)$

$$\eta_{th} = \frac{\text{Work Output}}{\text{Heat Input}} = 1 - \frac{C_V (T_2 - T_3) + RT_3 \log(\frac{V3}{V4})}{RT_1 \log(\frac{V2}{V1}) + C_V (T_1 - T_4)} \quad \text{(VI.1)}$$

$$C_V (T_2 - T_3) = C_V (T_1 - T_4) \text{ and } \frac{V3}{V4} = \frac{V2}{V1} \quad \text{(VI.2)}$$

$$\eta_{th\ Stirling} = 1 - \frac{T_3}{T_1} = 1 - \frac{T_L}{T_H} \quad \text{(VI.3)}$$

VI.2. Ericsson engine

VI.2.1. Advantages of Ericsson Cycle

The main advantage of the Ericsson cycle over the Carnot cycle and the Stirling Cycle is, it maintains the smaller pressure ratio for the given ratio of maximum to minimum specific volume with the higher Mean Effective pressure.

In this thermodynamic cycle, the heat addition and the heat rejection take place in both the Isobaric (Constant Pressure) and the Isothermal Processes.

The net effect is that the heat needs to be added only at constant temperature $T_3 = T_4$ and rejected at the constant temperature $T_2 = T_1$

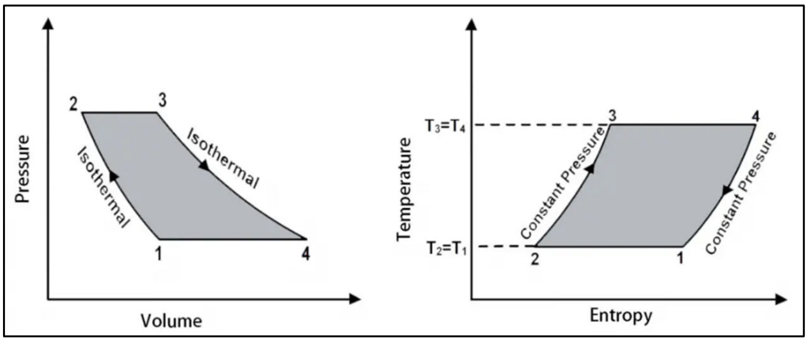


Fig. VI.3. Ericsson cycle

1 → 2 (Isothermal Compression)

2 → 3 (Isobaric Compression)

3 → 4 (Isothermal Expansion)

4 → 1 (Isobaric Expansion)

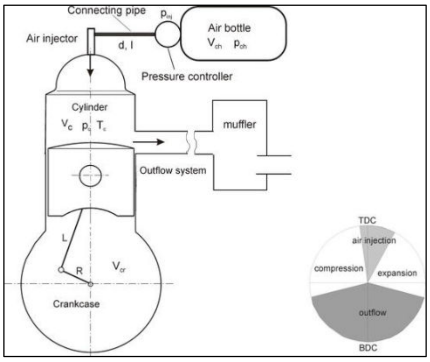
VI.2.2. Thermal effeciency of Ericsson Cycle

$$\eta_{\text{th Ericsson}} = \frac{T_3 - T_1}{T_3} = 1 - \frac{T_1}{T_3} = 1 - \frac{T_L}{T_H} \qquad \text{(VI.4)}$$

This Ericsson Cycle is used in the gas turbine engines by introducing a large number of stages with the help of heat exchangers insulators, reheaters. This cycle is not applicable to build the piston engines.

VI.3. Compressed air engine

VI.3.1. Design and operating principle



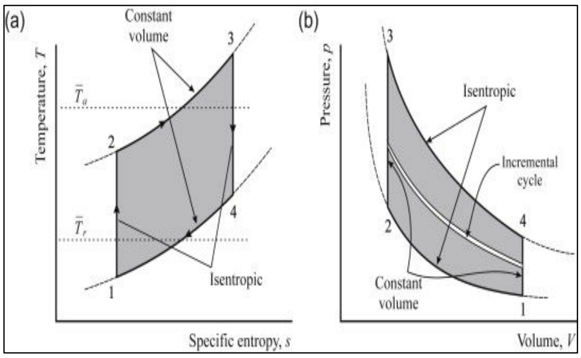


Fig. VI.4. Compressed air engine **Fig. VI.5.** Compressed air cycle
operating principle

1- The valves are connected to the storage tank by pipe at inlet and connectors are used for leak proofing in between. The outlet of valve is connected to the cylinder inlet port, the valves are controlled by electric circuit which controls the flow of compressed air and makes the piston reciprocate.
2- The piston rod is connected to the rack like arrangement which is used to rotate the sprocket connected with wheel the entire system is being supported by supporting frame.
3- The cylinder is connected to compressor which compresses the air.

The engine power is controlled only by change of the valve timing. The friction losses, compression stroke, pumping losses in the crankcase and outflow energy decrease the total engine efficiency.

Corrected Application Examples

Directed activities N^{bre} 01 : Technical thermodynamic reminder

Exercise1

Is atmospheric pressure constant? Else, establish the relationship dp= f(dz, M, R, T, P_{atm}) assuming that the air behaves like an ideal gas, and neglecting the variation of the ambient temperature (T: ambient temperature, dz: variation of l altitude, M: the molar mass of air, R: universal constant of ideal gases.

P_{atm}: Atmospheric pressure at z=0. Give Pascal's law $\frac{dp}{dz} = -\rho g$

Exercise2

A volume of air (ideal gas) of $2m^3$ at a pressure of 3 bars undergoes isothermal expansion up to the final pressure of 2 bars.

Calculate :

1. The variation in internal energy (the potential and kinetic energies variation is neglected).

2. The work provided by the air during the pressure drop.

3. The exchanged heat.

Exercise3

A mole of gas receives δQ heat quantity during a reversible elementary transformation.

δQ heat quantity can be expressed by three different relationship, depending on the state variables choice (pressure P, volume V, temperature T) :

$$dQ = C_V dT + l dV \ldots..1$$

$$dQ = C_P dT + h dp. \ldots 2$$

$$dQ = \lambda dp + \mu dV \ldots\ldots 3$$

γ *et* μ: Calorimetric Coefficients

- ✓ Express the calorimetric coefficients l, h, λ et μ depending on heat capacities Cv, Cp and partial derivatives $\left(\frac{\delta T}{dP}\right)_V$, $\left(\frac{\delta T}{dV}\right)_P$

Exercise4

A matter quantity n = 1 mol of ideal gas undergoes the following transformations succession :

(A → B) Isothermal expansion from P_A = 2 bar and T_A = 300 K up to P_B = 1 bar while remaining in contact with a temperature thermostat $T_0 = T_A$.

(B → C) Isobaric evolution up to V_C = 20.5 L always remaining in contact with the thermostat at T_0.

(C → A) Rversible adiabatic compression until returning to A state. The isentropic coefficient γ is taken equal to 1.4.

1. Represent this cycle in the Clapeyron diagram (P, V).

2. Determine the entropy between A and B.

3. Calculate the temperature in C, the work W_{BC} and the heat transfer Q_{BC} received by the gas. Deduce the exchanged entropy with the thermostat, as well as, the created entropy.

4. Are the proposed and the reverse cycles possibles ?

Directed activities N^{bre} 02 : Receiver cycle machines

Exercise1

Higher pressure P_1 (at the condenser outlet) and lower pressure P_3 (at the evaporator outlet) of a refrigeration setup, operating with the refrigerant fluid NH_3 (Ammonia) following the Carnot refrigeration cycle (isentropic expansion 1-2), are respectively equal to 11.2 atm and 2.966 atm. Calculate:

1- The quantity of heat rejected by the ammonia in the condenser Q_1
2- The quantity of heat subtracted by the ammonia in the evaporator Q_2
3- The effectiveness of this cycle:
a) According to the heat quantities Q_1 and Q_2
b) According to the temperatures T_1 and T_3
4- The refrigeration mass flow of ammonia per KW.
5- Compression power.

Exercise2

Air conditioning system equipped with R134a fluid with:

- Compressor outlet temperature (isentropic compression): + 62°C

- Compressor inlet temperature: -10°C

- Condenser outlet temperature (isobaric condensation): + 41°C

- Absolute Suction pressure : 2 bars.

- Absolute Discharge pressure : 16 bars.

At the condenser outlet, the fluid is just saturated (100% liquid). The expansion is isentropic. In our case, we obtain a mixture of 40% vapor and 60% liquid at the regulator outlet.

1- Draw the isenthalpic digram.

2- Calculate the refrigeration and heating efficiency.

Exercise3

Using the R134a diagram, determine the balance of a refrigeration installation. This concerns the evolution of 1 kg of refrigerant. An installation operating on R134a operates under the following conditions :

T_{compi}= -5°C, T_{compo}= +42°C, T_{condo}= +30°C, T_{evapi}= -12°C.

Exercise4

1-2 : Adiabatic compression, the pressure increasing from $p_1 = 10^5$ Pa to $P_2 = 2 \times 10^5$ Pa and the temperature increasing from $T_1 = 310$ K to T_2.

2-3 : in the coil in contact with the heating circuit (V 3 < V2): isobaric cooling, the temperature rising from T_2 to $T_3 = 330$ K.

3-4 : in the regulator: adiabatic expansion, the pressure going from $p_3 = p_2$ to $p_4 = p_1$, the temperature going from T_3 to $T_4 = 271$ K.

4-1 : in a coil immersed in industrial effluents : isobaric heating up to temperature T_1.

We give: Ideal gas constant: $R = 8.32$ J.mol $^{-1}$. K $^{-1}$

Specific heat capacity of air at constant pressure : $c_p = 1{,}000$ J.kg $^{-1}$. K $^{-1}$

Ratio of the specific thermal capacities of the air, at constant pressure and at constant volume $\gamma = \frac{6○o}{6○} = 1.4$

- During the isentropic transformation of an ideal gas : $pV^y = cte$

1. Represent the cycle described by the air on the Clapeyron diagram (P, V). Indicate with arrows the direction of the transformations.
2. Prouve that $T_2 = 378$ K.
3. Calculate the quantities of heat exchanged by a mass of 1 kg of air during each of the cycle transformations.
4. What is the variation in the internal energy of the air which describes the cycle ? State the first law of thermodynamics for a cycle.
5. Deduce the work W received by the mass of 1 kilogram of air, during the cycle.
6. : The heat pump efficiency $COP_{6○}$ is the ratio of the quantity of received heat by the hot source and the work received by the air, during a cycle. Calculate $COP_{6○}$.

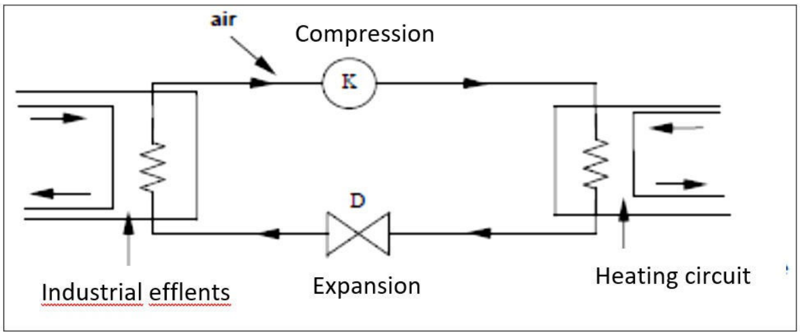


Fig. Heat pump

Directed activities N^{bre} 03 : Ideal cycles of Internal combustion engine

Exercise1

An internal combustion engine operates according to the cycle with heat supply at constant pressure. This cycle is characterized by the following parameters :

P_1= 0.9 bar ; V_2= 0.1 (m^3/Kg) ; $\varepsilon = 15$; T_{max}= 1800 °C and $\gamma = 1.4$; r = 287 J/Kg/K

1- Represent this Diesel cycle on a Clapeyron P-V diagram

2- Calculate the state variables of the driving fluid (P, V, T) at the characteristic points of the cycle 1, 2, 3 and 4

3- Calculate the subsequent expansion rate δ, and deduce the value of ε' (The prior expansion rate) $\delta = \frac{V_4}{V_3}$, $\varepsilon' = \frac{V_3}{V_2}$

4- Check the relationship of the Diesel efficiency $\boldsymbol{\eta_{th}} = 1 - \frac{1}{\varepsilon^{\gamma-1}} . \frac{(\varepsilon')^{\gamma} - 1}{\gamma\,(\varepsilon'-1)}$

5- Calculate the thermodynamic efficiency of this cycle.

Exercise2

An internal combustion engine operates according to the mixed cycle (Sabathe cycle). This cycle is characterized by the following parameters:

$\lambda = 1.5$, $\gamma = 1.4$, Q_3= 400 (KJ/Kg), $\varepsilon = 14$, T_1= 70 °C, V_2= 1 (m3/ Kg), η_{th} = 50%

1- Draw this cycle on a P-V Diagram ?

2- Calculate the state variables of the driving fluid (P, v, T) at the characteristic points of cycle 1, A, A', B, C and D ?

3- Calculate the prior expansion rate ε' ?

4- What happens to this cycle when ε'= 1 ?

5- Calculate the speed average of the piston knowing that N=3000 rpm.

We give:

Cylinder bore : D = 90 mm and piston stroke/cylinder bore ratio = 1.5?

Exercise3

The Diesel engine is an internal combustion engine whose ignition is not controlled by spark plugs but by high compression. It operates according to the eponymous cycle consisting of two isentropics, an isobaric and an isochore. More precisely, the cycle can be described in four stages :

1. a cylinder admits air alone through an intake valve into a volume VA (portion IA of the cycle).

2. The valves are closed. Fuel injection starts at point B and is progressive to point C so that the pressure remains constant ;

3. The valves are always closed and the combustion products undergo isentropic expansion by pushing the piston back to its initial position (portion CD)

The cycle is characterized by the volumetric compression ratio $\alpha = \frac{V_A}{V_B}$ and the prior expansion ratio $\beta = \frac{V_C}{V_B}$. The temperatures of the mixture in A and C are: $T_A = 293$ K and $T_C = 1220$ K.

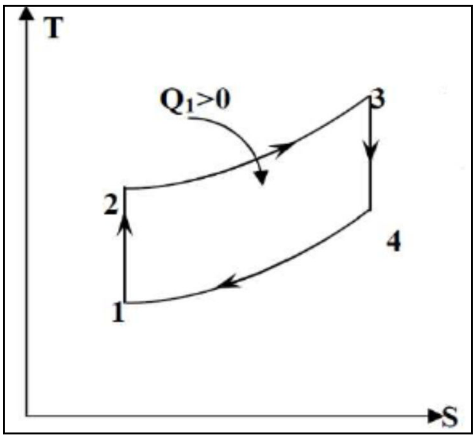
1. Draw this Diesel cycle in the Clapeyron diagram, showing the 5 points I, A, B, C, and D.
2. Identify the quantities of heat exchanged and their signs on the cycle, the work performed and their signs, and write the heat balance on a cycle.
3. Give the expression for the quantities of heat exchanged and give the expression for the efficiency η of this heat engine.

Show that the efficiency of this engine depends only on the compression ratio α and the expansion ratio β. We give: $\gamma = 1{,}4$; $\alpha = 14$ et $\beta = 1{,}55$.

Directed activities N^{bre} 04 : Gas turbine and turbojet

Prerequisites : *-Thermodynamic transformations (isentropic, isobaric, isochoric, isothermal).*
-Thermodynamic cycles
Objectives : *-Energy balance and thermal efficiency.*
-Power calculation.

Exercise1



A gas turbine system without recovery represented by the Joule cycle (entropic diagram) composed of

-Axial compressor (compression rate $\tau_c=4$, $\eta_{comp}=80\%$) draws in ambient air at temperature $T_1=27°C$.

-Combustion chamber with negligible thermal losses, $T_3=1027°C$.

-Turbine with moving wheel, efficiency $\eta turb=65\%$

-Exhaust economizer.

Gases of the cycle assumed to be perfect, $C_{pair}= C_{pgaz}=1$ KJ/Kg.K, $\beta = (\gamma-1)/\gamma = 0.4$.

1. Calculate the real compression and expansion work Wc and Wd.

2. Determine the real heat input supplied/escaped by the cycle (Q_{2r3}, Q_{4r1}).

3. Calculate the real η_{therm} of the cycle.

Exercise2

An air flow of 20 Kg/s at temperature T_1 = 30 °C enters the compressor of a gas turbine installation operating according to the Joule (Brayton) cycle. The compression ratio is 7 and the temperature at the turbine inlet is T_3 = 650 °C. The isentropic efficiencies of the turbine and the compressor are respectively 0.85 and 0.80, the efficiency of the combustion chamber is 0.80.

We consider that $C_{pair} = C_{pgaz} = 1$ KJ/Kg.K and $\gamma_{air} = \gamma_{gaz} = 1.4$. Calculate :

1. The necessary real power needed for air compression

2. The total real power produced by the turbine

3. The available power (received by the electric generator) if the efficiency of the coupling is equal to 0.90.

4. The theoretical thermal efficiency of the cycle.

5. The real efficiency of the cycle.

Exercise3

We consider a power plant gas turbine. The problem data is as follows :

- State 1 is characterized by the values : P_1 = 100 kPa and T_1 = 20 °C.
- The pressure at which combustion takes place is: $P_2 = P_3$ = 1780 kPa.
- The burned gases have, at the outlet of the combustion chamber, a temperature T_3 = 1000 °C.

– The transformations in the turbine and the compressor are assumed to be isentropic;

– The mechanical efficiencies of the turbine and the compressor are worth $\eta_{mec} = 0.98$.

Calculate :

In the case a constant value for γ ($\gamma = 1, 4$) :

1. Successive states (P, T) of the gas during the cycle.
2. Heat Q released by combustion, the engine work available and the overall efficiency of the cycle.

Directed activities N^{bre} 05 : Steam turbine

Prerequisites : *-Thermodynamic transformations (isentropic, isobaric, isochoric, isothermal).*
-Thermodynamic cycles
Objectives : *-Energy balance and thermal efficiency.*
-Power calculation.

Exercise1

we consider a steam thermal power plant operating with the Rankine cycle, the saturated steam enters the turbine at a pressure of 80 bar and the saturated liquid leaves the condenser at a pressure of 0.08 bar, the net power developed by the cycle is order of 100 MW. Determine :

1- The thermal efficiency and the steam flow rate in Kg/h.

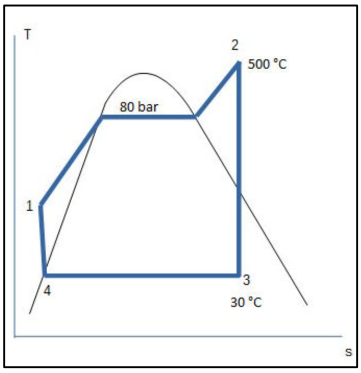
2- The theoretical power of the turbine in MW.

3- The heat output delivered by the boiler in MW.

4- The cooling power in the condenser.

5- The mass flow of **saturated cooling water** in the condenser (Kg/h) knowing that the inlet temperature is 17°C and the outlet temperature is 40°C.

Exercise2



The internal efficiency of the steam turbine is 0.9. Calculate :

1- The thermal efficiency of the cycle (reversible).

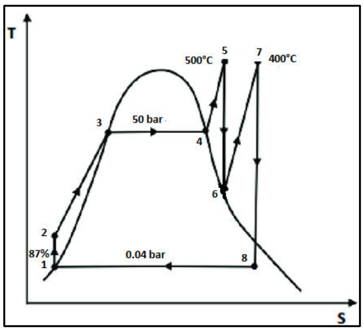
2- The thermal efficiency of the cycle (reversible) if the work of the pump is neglected.

3- The reversible power produced by the turbine if the steam flow rate is 60 Kg/h.

The real power produced by the turbine. We give :

$h_{2\,(500\,°C)}$ =3398.05 Kj/Kg, **$S_{2\,(500\,°C)}$ =6.7228 Kj/Kg**

Exercise3



Calculate :

1- The net work of the cycle.

2- The total quantity of supplied heat to the motor fluid by the steam generator SG.

3- The isentropic thermal efficiency of the cycle.

4- The specific consumption of steam.

We give : $h_{(500°C)}$=3433.75 KJ/Kg, $S_{(500°C)}$=6.985 KJ/Kg, $h_{(400°C)}$ =3275 KJ/Kg, $S_{(400°C)}$=8.033 KJ/Kg.

Solution of Directed activities Nbre 01 : Technical thermodynamic reminder

Exercise1

$$\begin{cases} \frac{dp}{dz} = -\rho g \quad Pascal\ Law\ (1) \\ pV = nRT \quad Perfect\ gases\ Law\ (2) \end{cases}$$

$$(1) \rightarrow \ dp = -\rho g dz \ (3)$$

$$(2) \rightarrow \ pV = \frac{\rho V}{M} RT \ \rightarrow p = \frac{\rho}{M} RT \quad (4)$$

$$\frac{(3)}{(4)} \ \rightarrow \ \int_{p_{atm}}^{p} \frac{dp}{p} = -\frac{M}{RT} g \int_0^z dz \ \rightarrow p = \ p_{atm} e^{-\frac{Mg}{RT} z}$$

Exercise2

Isothermal expansion, de 1 à 2, $T_2 = T_1$ et therefore the ideal gas equation reduces to,
$P_2 V_2 = P_1 V_1 = m.r.T_2 = m.r.T_1 = m.r.T = K$.
Ce qui donne :
$V_2 / V_1 = P_1 / P_2$

1. **The internal energy variation is:**

$\Delta U_{1\text{-}2} = U_2 - U_1 = m.\ C.\ (T_2 - T_1) = 0$

2. **The work provided by the air during this expansion is:**

$W_{1\text{-}2} = -\ K.\ Ln\ (V_2 / V_1) = -\ K.\ Ln\ (P_1 / P_2) = -\ P_1 V_1.\ Ln\ (P_1 / P_2)$
$= -\ 3 \times 10^5 \times 2 \times Ln\ (3/2)$
$= -\ 243 \times 103$ J
$= -\ 243$ KJ

3. **he quantity of heat exchanged is :**

The variation of internal energy $\Delta U_{1\text{-}2} = 0$, the quantity of heat exchanged is equal to the work supplied
$Q_{1\text{-}2} = -\ W_{1\text{-}2} = 243$ KJ

Exercise3

$dQ = C_V dT + l dV$……………………..(1)
$dQ = C_P dT + h dP$…………………… (2)
$dQ = \lambda dp + \mu dV$……………………... (3)

1) At constant pressure dp=0

$$(2) \rightarrow \ dQ = C_P dT$$

$$(1) \rightarrow C_P dT = C_V dT + l dV \ \rightarrow \ \left(C_p - C_V\right) dT = l dV$$

$$l = \left(C_p - C_V\right) \left(\frac{dT}{dV}\right)_p$$

$$(3) \rightarrow C_P dT = \mu dV \ \rightarrow \ \mu = C_P \left(\frac{dT}{dV}\right)_p$$

2) A volume constant dV=0

$(1) \rightarrow \ \mathrm{dQ} = \mathrm{C}_V dT$

$(2) \rightarrow \mathrm{dQ} = \mathrm{C}_P dT + hdP \ \rightarrow \ \left(\mathrm{C}_p - \mathrm{C}_V\right)dT = ldV$

$(3) \rightarrow dQ = \lambda dp$

$$\Longrightarrow \quad \mathrm{C}_V dT = \lambda dp = \mathrm{C}_P dT + hdP \ \rightarrow \ \lambda = \mathrm{C}_V \left(\frac{dT}{dp}\right)_V$$

$$\rightarrow h = \left(\mathrm{C}_V - \mathrm{C}_p\right)\left(\frac{dT}{dp}\right)_V$$

Exercise4

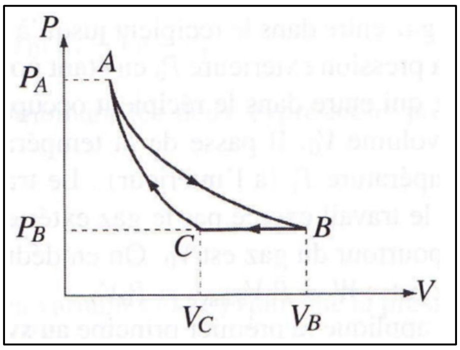
1. Drawing of cycle :

- Let's calculate its volume V_A :

$$\mathrm{V_A} = \frac{\mathrm{nRT_A}}{P_A} = 12\ \mathrm{L}$$

- **We apply again the ideal gas law with** $T_B = T_A$ because of the isothermal transformation.

$$\mathrm{V_B} = \frac{\mathrm{nRT_A}}{P_B} = 25\ \mathrm{L}$$



The transformation AB is an isotherm, the entropy variation ΔS_{AB} is therefore a function of T which is eliminated, for example:

$$\Delta \mathrm{S_{AB}} = \mathrm{n\,.\,R.\,T}_A\mathrm{.\ln}\frac{\mathrm{p_A}}{\mathrm{p_B}} = \Delta \mathrm{S_{AB}} = 1\,.8.32.300.\ln 2 = 1.73\ \mathrm{KJ/K}$$

- We calculate the entropy exchanged from the heat transfer from the first Principle,

$$\Delta \mathrm{U_{AB}} = \mathrm{W_{AB}} + \mathrm{Q_{AB}}$$

For an isothermal transformation of ideal gas,

$$\Delta \mathrm{U_{AB}} = \mathrm{C_V}\Delta \mathrm{T} = 0 \ \ , \ \ \mathrm{W_{AB}} = -\mathrm{Q_{AB}}$$

We calculate the work exchanged W_{AB} assuming the static transformation.

$$\mathrm{W_{AB}} = -\int \mathrm{P_{ext}dV} = -\int \mathrm{PdV} = -\int \mathrm{nRT}\frac{\mathrm{dV}}{\mathrm{V}} =_{\mathrm{isotherme}} -\mathrm{nRT_A}\int_{\mathrm{V_A}}^{\mathrm{V_B}} \frac{\mathrm{dV}}{\mathrm{V}}$$

Therefore $$\mathrm{W_{AB}} = -\mathrm{nRT_A}\ln\frac{\mathrm{V_B}}{\mathrm{V_A}} = \mathrm{nRT_A}\ln\frac{\mathrm{P_B}}{\mathrm{P_A}}$$

$$Q = -nRT_A \ln\frac{P_B}{P_A}$$

We deduce from this the entropy exchanged with the temperature of thermostat T_0 = TA during the transformation AB.

$$S_{exch} = \frac{Q_{AB}}{T_0} = -nRT_0 \ln\frac{P_B}{P_A}$$

We then notice that $\Delta S = S_{exch}$, c'est-à-dire que $S_{crea} = 0$: the AB transformation is reversible

$$T_C = \frac{P_B V_C}{nR} = 250\ K$$

The transformation is isobaric, so the work received is written

$$W_{BC} = -P_B(V_C - V_B) = 440\ J$$

Finally, heat transfer is deduced from the first law, for example in terms of enthalpy.

$$Q_{BC} = \Delta H_{BC} = C_p(T_C - T_B) \quad \text{Where} \quad Q_{BC} = \frac{nR\gamma}{\gamma-1}(T_C - T_B) = -1{,}6\ kJ$$

The exchanged entropy is deduced directly

$$S_{exch,BC} = \frac{Q_{BC}}{T_0} = -5{,}2\ J/K$$

To calculate the entropy created, it is first necessary to calculate the variation in entropy of the gas between B and C, which is done with the given expressions. As the transformation is isobaric, the most clever thing is to use an expression dependent on P since the associated terms compensate each other. We deduce:

$$\Delta S_{BC} = \frac{nR\gamma}{\gamma - 1} \ln\frac{T_C}{T_B} = -5{,}7\ J/K$$

Finally, we deduce the created entropy

$$S_{crexe,BC} = \Delta S_{BC} - S_{exch,BC} = -0.5\ J/K$$

The entropy created during step BC would therefore be negative, which is absolutely impossible. The propositionx cycle is therefore unrealizable. On the other hand, the reverse cycle is possible because two transformations are reversible and the third is associated with a creation of entropy, which is permitted by the second principle.

Solution of Directed activities Nbre 02 : Receiver cycle machines

Exercise1

1. Calculation of Q_1 : (saturation table for Ammonia R717), PH=11.2 bars

$Q_1 = H_1 - H_4 = 250.48 - 1484.60 = -1220.94$ KJ/Kg

2. Calculation of Q_2 : (saturation table for Ammonia R717), PB=2.966 bars

$H_2, H_3 = ?$ $S_2 = S_1; S_3 = S_4$ (isentropic compression and expansion)

Two ways to find enthalpies !

- By the thermodynamic diagrams of Ammonia (enthalpic and entropic)
- By the title in steam:

$$X_v = \frac{H_2 - H_2'}{H_2'' - H_2'} = \frac{S_2 - S_2'}{S_2'' - S_2'} = \frac{S_1 - S_2'}{S_2'' - S_2'} \ldots\ldots(1)$$

$$X_v = \frac{1.4633 - 0.8483}{5.7373 - 0.8483} = 0.125$$

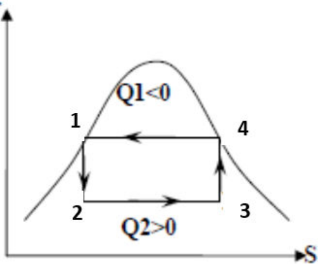
Replacing in relationship (1) :

$H_2 = X_v.\,(H_2'' - H_2') + H_2' = 0.125.\,(1450.49-159.05) + 159.05 = 320.48$ KJ/Kg

The same procedures to calculate H_3

$H_3 = 1327.028$ KJ/Kg

$Q_2 = H_3 - H_2 = 1327.028 - 320.48 = 1006.54$ KJ/Kg



The effectiveness of the Carnot cycle

$$\text{COP}_{rev} = \frac{Q_{2rév}}{|Wrev|} = \frac{Q_{2rév}}{|Q_{1rév} - Q_{2rév}|} = \frac{H_3 - H_2}{(H_4 - H_1) - (H_3 - H_2)} = \frac{1006.54}{1220.94 - 1006.54} = 4.69$$

$\text{COP}_{rév} = \frac{T_2}{T_1 - T_2}$ for a Carnot cycle $(\Delta S_{14} = \Delta S_{32})$

Exercise2

In order to draw, we have to determine the points on this diagram representing cycle transformations :

1. Isentropic compression : [1-2] $T_{out\ comp}$= 62°C

The work expended by the compressor

$$Wc = H_2 - H_1 = 432.58 - 391.32 = 35.2 \text{ kJ/kg}$$

2. Isobaric desuperheating [2-2']

$Q_{C1} = H_{2'} - H_2 = 425.32 - 432.58 = -7.26$ kJ/kg

3. Isobaric Condensation: [2'-3] --------- (vapor/liquid) à $T_{icond} = T_2 = 58^0$C

Heat transferred to the hot source

$$Q_{C2} = H_3 - H_{2'} = 284.13 - 425.32 = -141.19 \text{ kJ/kg}$$

4. Undercooling [3-3'] $T_{undercool}$ from 58°C to 41°C

$$H_{3'} = H_4{}^{=} X_v \,.\, H''_4 + (1-X_v).\, H'_4 = 268.68 \text{ kJ/kg}$$

$$Q_{C3} = H_{3'} - H_3 = 268.68 - 284.13 = -15,45 \text{ kJ/kg}$$

Xv is the Vapor quality (Xv=0.4)

5. Expansion : [3’-4] --------- isentropic transformation

The work produced by the expansion valve

$$W_{Exp} = 0 \text{ (isenthalpic } H_{3'} = H_4)$$

6. Evaporation : [4-1] --------- isobaric transformation (liquid/vapor)

Heat absorbed from the cold source

$$Q_2 = H_1 - H_4 = 391.32 - 268.68 = 122.64 \text{ kJ/kg}$$

Refrigeration efficiency (refrigeration performance)

Depending on the compression work, we find:

$$COP_F = \frac{Q_2}{W_c} = \frac{122.64}{35.2} = 3.48$$

Depending on the heat quantities, we find:

$$COP_F = \frac{Q_2}{|Q_1| - |Q_2|} = \frac{122.64}{(141.19 + 15{,}45 + 7.26) - 122.64} = 2.97$$

Refrigeration efficiency can be improved by superheating the steam at the compressor inlet making:

$$\frac{122.64}{(141.19 + 15{,}45 + 7.26) - (122.64 + X)} = 3.48$$

X= 76.50 kJ/kg

Calorific efficiency (heat performance) :

$$COP_C = \frac{Q_{C1} + Q_{C2} + Q_{C3}}{W_c} = \frac{141{,}19 + 15{,}45 + 7{,}26}{35.2} = 4.65$$

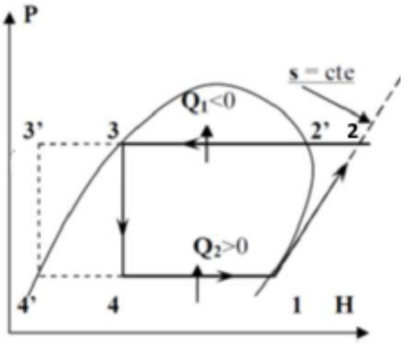


Fig. Thermodynamic diagram

Exercise3

Energy assessment of a cycle :

Using the pressure-enthalpy diagram of the refrigerant R134a we find:

Compressor assessment: compression work.

$$Wc = H_2 - H_1 = 436 - 398 = 38 \text{ kJ/kg.}$$

Desuperheating of superheated vapors at discharge (evacuation of sensible heat from the vapor from 42^0C to 37°C).

$$Q_{\text{desuperheated}} = H_3 - H_2 = 418 - 436 = -18 \text{ kJ/kg.}$$

Condenser assessment:

It is the specific enthalpy of change of state at the condenser (quantity of latent heat evacuated by 1 kg of fluid which during condensation T_C=37°C).

$$Q_1 = H_4 - H_3 = 260 - 418 = -158 \text{ kJ/kg.}$$

Subcooling of the liquid in the condenser (extracted of sensible heat from the liquid from 37^{0}C to 30°C).

$$H_5 – H_4 = 240 – 260 = - 20 \text{ kJ/kg}.$$

Expansion Assessment : The expansion of the refrigerant takes place without exchange of energy with the external environment.

$$H_6 – H_5 = 240 – 240 = 0 \text{ kJ/kg}).$$

Evaporator assessment : This is the specific enthalpy of change of state at the evaporator. (Amount of latent heat absorbed by 1 kg of evaporating fluid).

$$Q_2= H_7 – H_6 = 390 – 240 = 150 \text{ kJ/kg}.$$

Superheating of superheated steam in the evaporator and in the suction line (sensible heat gain of the steam.

$$H_1 – H_7 = 398 - 390 = 8 \text{ kJ/kg}.$$

Cycle assessment : Gains and losses

$$\Delta U = [(h_6\text{-}h_4) + (h_7\text{-}h_6) + (h_1\text{-}h_7)] - [(h_2\text{-}h_1) + (h_3\text{-}h_2) + (h_4\text{-}h_3)]$$

$$\Delta U= [150 + 8 + 38] - [-18 -158 -20] = 196\text{-}196 = 0 \text{ kJ/kg}.$$

Refrigeration performance factor (COP_F)

$$COP_F = \frac{Q_2}{W_c} = \frac{150}{38} = 3.94$$

calorific performance factor (COP_C)

$$COP_C = \frac{Q_2}{W_c} = \frac{158}{38} = 4.15$$

Exercise4

1.

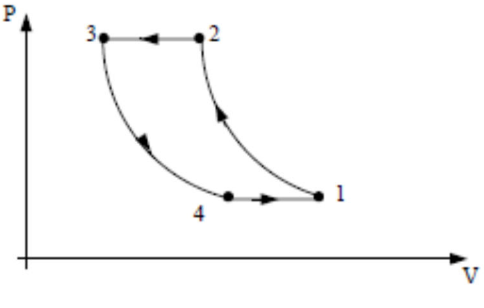


Fig. Thermodynamic transformations cycle

2. The cycle is necessarily described in the trigonometric sense because it is a heat pump (the cycle is therefore resistant: it requires an input of work to be carried out). The reversibility of the cycle implies that it is not dithermic, and therefore that the heating circuit and the industrial effluents do not constitute sources of heat: their temperature is certainly relatively constant, equal to T_1 (hot source) and T_3 (cold source), which implies that the cycle is dithermic and therefore that it is certainly not reversible.

$pV^{\gamma} = cte$ Adiabatic transformation

$\rightarrow \begin{cases} pV^{\frac{1}{\gamma}} = cte \\ PV = nRT \end{cases}$ The report : $pV^{\frac{1}{\gamma}/\gamma}.T = cte$ and $pV^{1-\gamma/\gamma}.T$

$$\rightarrow T_2 = T_1.\left(\frac{P_2}{P_1}\right)^{1-\gamma/\gamma} \approx 310.(1.10^5/2.10^5)^{\,1-\gamma/\gamma} \approx 378\,K$$

3.

$Q_{12} = 0$ et $Q_{34} = 0$ because these transformations are adiabatic.

$Q_{23} = m.cp.\,(T_3\text{-}T_2)$ because these transformations are isobaric, likewise $Q_{41} = m.cp.(T_1\text{-}T_4)$

We find $Q_{23} \approx 1\times1000\times(330 - 378) \approx -48$ kJ et Q41 $\approx 1 \times 1000 \times (310 - 271) \approx 39$ kJ

4.

On cycle $\Delta U_{cycle} = 0$ because the final temperature is always equal to the initial temperature for one cycle.

First thermodynamic principle : $W_{cycle} + Q_{cycle} = \Delta U_{cycle} \Rightarrow W_{cycle} + Q_{cycle} = 0$.

5.

$W_{cycle} = -Q_{cycle}$ and $W_{cycle} = -(Q_{23} + Q_{41}) \approx -(-48.103 + 39.103) \approx 9$ kJ

6.

$$COP_C = \left|\frac{Q_{23}}{W_c}\right| = \left|\frac{48.10^3}{9.10^3}\right| = 5.3$$

Note: for a dithermal heat pump operating between sources T_1 et T_3 we have :

$$COP_{Cmax} = T_3/\,(T_3\text{-}T_1) = 330\,/\,(330 - 310) \approx 16{,}5$$

Obtained for a reversible cycle so the cycle is irreversible.

Solution of Directed activities Nbre 03 : Internal combustion engine cycles

Exercise1

1- Clapeyron diagram P-V :

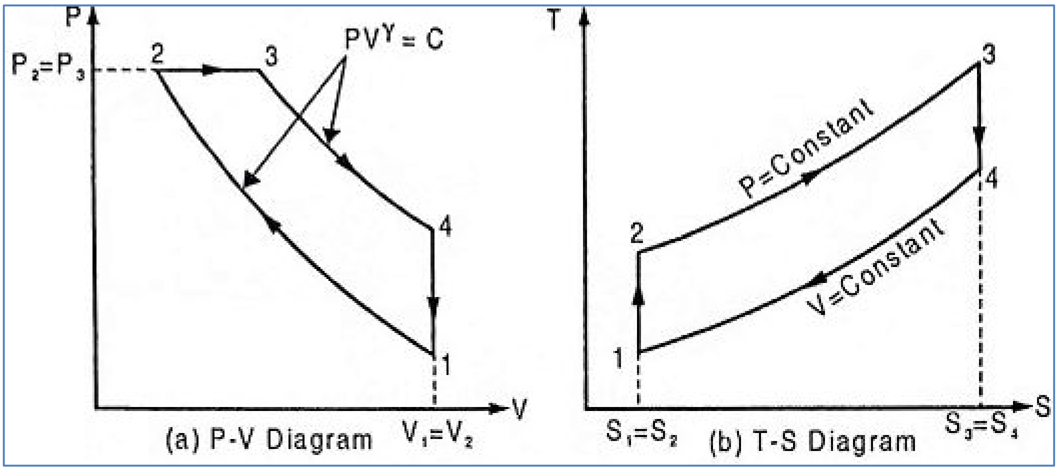


Fig. III.8. Diesel Diagram

2- Calculation of the driving fluid state variables (P, v, T) at the characteristic points of the cycle 1,2, 3 et 4 ? **r= 287 J/Kg/K**

- Point 1: P_1=0.9 bar; $V_1=V_2.\ \varepsilon = 1.5\ (m^3/Kg)$; $T_1 = \frac{P_1.V_1}{r}$= 470.38 K
- Point 2 : $V_2 = 0.1\ (m^3/Kg)$; $P_2 = P_1.\ (\frac{V_1}{V_2})^{\gamma}$= 39.88 $bars$; $T_2 = T_1.\ (\frac{V_1}{V_2})^{\gamma-1}$= 1389.58 K
- Point 3 : $T_{max} = T_3$ = 2073 K ; $V_3 = V_2.\ \frac{T3}{T2}$ = 0.149 (m^3/Kg) ; $P_3 = P_2$
- Point 4 : $V_4=V_1$; $P_4= P_3.\ (\frac{V_3}{V_4})^{\gamma}$= 1.57 $bars$; $T_4 = T_3.\ (\frac{V_3}{V_4})^{\gamma-1}$= 823.07 K

3- Subsequent expansion rate δ :

$\delta = \frac{V_4}{V_3}$= 10.06

4- Prior expansion rate ε' :

$\varepsilon' = \frac{V_3}{V_2}$= 1.49

5- Rendement du cycle Diesel :

$$\boldsymbol{\eta_{th}} = 1 - \frac{(\frac{T_4}{T_1}-1)}{\frac{\gamma T_2}{T_1}(\frac{T_3}{T_2}-1)} = 1 - \frac{1}{\varepsilon^{\gamma-1}} . \frac{(\varepsilon')^{\gamma} - 1}{\gamma\ (\varepsilon'-1)} = 0.63 = 63\%\text{w}$$

Exercise2

1. Clapeyron Diagram of Sabathe Cycle :

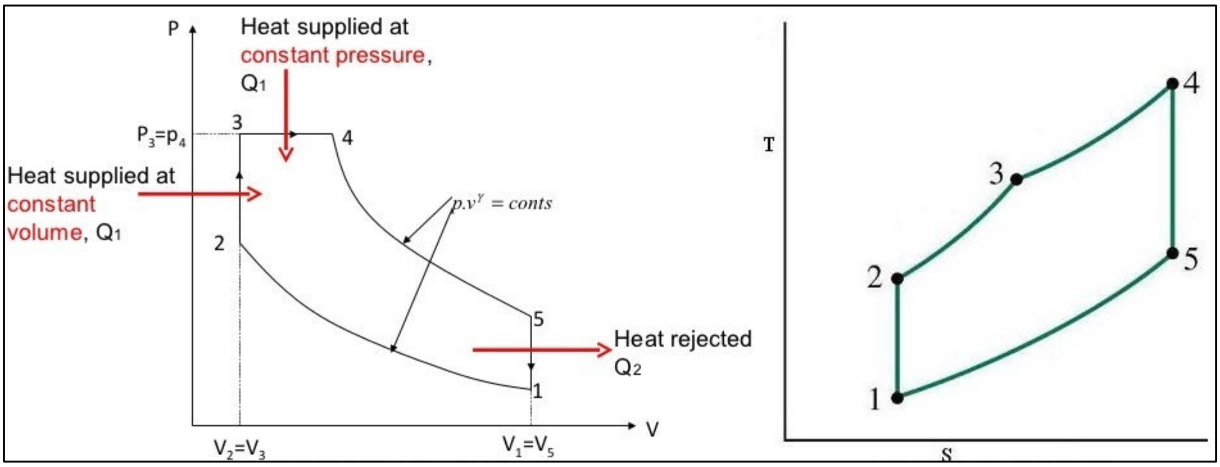


Fig. III.10. Sabathe Cycle

2. Calculation of the driving fluid state variables (P, v, T) at the characteristic points of the cycle 1, 2, 3, 4 et 5 : (λ : constant volume combustion ratio), **r= 287 J/Kg/K**
   - Point 1 : $V_1=V_2.\ \varepsilon = 14\ (m^3/Kg)$; T_1 =343 K ; $P_1 = \frac{r.T_1}{V_1} = 7031.5$ pa
   - Point 2 : $V_2 = 1\ (m^3/Kg)$; $T_2 = T_1.\ (\frac{V_1}{V_2})^{\gamma-1} = 985.7\ K$; $P_2=P_1.\ (\frac{V_1}{V_2})^{\gamma} = 2.82\ 10^5$ pa
   - Point 3 : $P_3 = \lambda.P_2 = 4.24\ bars$; $V_3 = V_2$; $T_3 = \lambda.T_2 = 1478.5\ K$
   - Point 4 : $P_4 = P_3$; $T_4 = T_5.\ (\frac{P_4}{P_5})^{\frac{\gamma-1}{\gamma}} = 2205.33\ K$; $V_4 = \frac{rT_4}{V_4} = 1.49\ (m^3/\text{Kg})$
   - Point 5 : $T_5 = T_1 + \frac{Q_3}{C_v} = T_1 + \frac{Q_3 10^3(\gamma-1)}{r} = 900.49\ K$; $V_5=V_1=14\ (m^3/Kg)$;

     $P_5 = \frac{rT_5}{V_5} = 1.846\ bars.$

3. Prior expansion rate ε'

$\varepsilon' = \frac{V_4}{V_3} = 1.49.$

If the prior expansion rate $\varepsilon' = \frac{V_4}{V_3} = 1$ **(Gasoline cycle)**

4. Calcul de la vitesse moyenne du piston :

$$v_{moy\ piston} = \frac{C_p.N}{30} = \frac{1.5.D.10^{-3}.N}{30} = 13.5\ m/s$$

Solution of Directed activities Nbre 04 : Gas turbine and turbojet

Exercise1

1- Real compression work :

$w_{r\,comp} = H_{2'} - H_1 = C_p \times (T_{2'} - T_1)$

Isentropic transformation :

$T_2 = T_1\,(P_2/P_1)^{(\gamma-1)/\gamma} = (27+273) \times (4)^{(1,4-1)/1,4}$ $\mathbf{T_2 = 522,33\ K}$

$T_{2'} = T_1 + (T_2 - T_1) / \eta_{comp} = (27+273) + [522,33 - (27+273)] / 0,80$ $\mathbf{T_{2'} = 577,91\ K}$

$w_{r\,comp} = 1 \times [577,91 - (27+273)]$ $\mathbf{w_{r\,comp} = 277,91\ KJ/Kg}$

2- Real expansion work:

$W_{r\,exp} = H_{4'} - H_3 = C_p \times (T_{4'} - T_3)$

Isentropic transformation :

$T_4 = T_3\,(P_4/P_3)^{(\gamma-1)/\gamma} = (1027+273) \times (1/4)^{(1,4-1)/1,4}$ $(P_4 = P_1\ ;\ P_3 = P_2)$ $\mathbf{T_4 = 746,65\ K}$

$T_{4'} = T_3 + (T_4 - T_3) \times \eta_{dét} = (1027+273) + [746,65 - (1027+273)] \times 0,65$ $\mathbf{T_{4'} = 940,32\ K}$

$W_{r\,exp} = 1 \times [940,32 - (1027+273)]$ $\mathbf{W_{r\,exp} = -359,67\ KJ/Kg}$

3- Thermal input/output of the cycle ($Q_{2'3}$, $Q_{4'1}$) :

$Q_{23r} = H_3 - H_{2'} = C_p \times (T_3 - T_{2'}) = 1 \times [1027+273-577,91] =$ **722,09 Kj/Kg**

$Q_{41r} = H_1 - H_{4'} = C_p \times (T_1 - T_{4'}) = 1 \times [27+273-940,32] =$ **-640,32 KJ/Kg**

4- Real cycle efficiency

$\eta_{th\text{-}real} = (w_{cycle})_{real} / (q_1)_{real} = [\,|W_{r\,exp}| - |w_{r\,comp}|\,] / (q_{2'3})$

$\eta_{th\text{-}real} = \frac{359,67 - 277,91}{722,09} = 0.11$

Exercise2

1- Necessary real power for the air compression :

$(P_{u\,comp})_{real} = \phi \times (w_{comp})_{real} = \phi \times (H_{2'} - H_1) = \phi \times C_p \times (T_{2'} - T_1)$

1-2 Isentropic transformation :

$T_2 = T_1\,(P_2/P_1)^{(\gamma-1)/\gamma} = (30+273) \times (7/1)^{(1,4-1)/1,4}$ $\mathbf{T_2 = 528,32\ K}$

$T_{2'} = T_1 + (T_2 - T_1) / \eta_{comp} = (20+273) + [528,32 - (30+273)] / 0,80$ $\mathbf{T_{2'} = 584,65\ K}$

$(P_{u\,comp})_{real} = 20 \times 1 \times [584,65 - (30+273)] = 5633\ KW$

2- Real total power produced by the turbine :

$(P_{u\,turb})_{real} = \phi \times (w_{turb})_{réel} = \phi \times (H_{4'} - H_3) = \phi \times Cp \times (T_{4'} - T_3)$

$T_4 = T_3\,(P_4/P_3)^{(\gamma-1)/\gamma} = T_3\,(P_1/P_2)\,(\gamma-1)/\gamma = (650+273) \times (1/7)^{(1,4-1)/1,4}$

$=$ **529,35 K**

$T_{4'} = T_3 + (T_4 - T_3) \times \eta_{turb} = (650+273) + [529,35 - (650+273)] \times 0,85 =$ **588,4 K**

$(P_{u\,turb})_{real} = 20 \times 1 \times [(588,4 - (650+273)]$

$= -6692\ KW =$ **- 6,692 MW**

$|(P_{u\,turb})_{real}| = 6692\ KW =$ **6,692 MW**

3-1 Useful power of the cycle :

$|(P_{u\,cycle})_{real}| = [\,|(P_{u\,turb})_{real}| - |(P_{u\,comp})_{real}|\,] =$ **1059 KW**

3-2 Useful mechanical power available on the generator tree :

$\left|(P_{mec})_{\text{real}}\right| = \left|(P_{u\,t})_{\text{real}}\right| \times \eta_{gen} = 1059 \times 0,9 =$ **953,1 KW**

4- Reversible efficiency of the cycle :

$\eta_{\text{th-theo}} = (w_{\text{cycle}})_{\text{theo}} / (q_1)_{\text{theo}} = [\,|(w_{\text{turb}})_{\text{theo}}| - |(w_{\text{comp}})_{\text{theo}}|\,] / (q_1)_{\text{theo}}$

$= [(H_3 - H_4) - (H_2 - H_1)] / ((H_3 - H_2)$

$= [(T_3 - T_4) - (T_2 - T_1)] / ((T_3 - T_2)$
$= [(650+273 - 529,35) - (528,32 - (273+30))] / (650+273 - 528,32)$
$=$ **0,427**

5- Real efficiency of the cycle with thermal losses in the combustion chamber :

$\eta_{\text{th-real}} = (w_{\text{cycle}})_{\text{real}} / (q_1)_{\text{real}} = [\,|(w_{\text{turb}})_{\text{real}}| - |(w_{\text{comp}})_{\text{real}}|\,] / (q_1)_{\text{real}}$
$= [(H_3 - H_{4'}) - (H_{2'} - H_1)] / [(H_3 - H_{2'}) / \eta_{c\,ch}]$
$= [(T_3 - T_{4'}) - (T_{2'} - T_1)] / [(T_3 - T_2) / \eta_{ch.c}]$
$= [(650+273 - 588,4) - (584,65 - (273+30))] / [(650+273 - 584,65)/0,80] =$ **0,125**

<u>Exercise3</u>

Characteristic table of states (**For constant specific heats)** :

State	Pressure (bars)	Temperature (°C, K)	
1	1	20	293.15
2	17.8		
3	17.8	1000	1273.15
4	1		

1. The transformations (1 → 2) et (3 → 4) are isentropic, transformation law is $p.v^{\gamma}$=cte

$\frac{T_2}{T_1} = \frac{P_1}{P_3}^{\frac{\gamma-1}{\gamma}}$ $\frac{T_4}{T_3} = \frac{P_4}{P_3}^{\frac{\gamma-1}{\gamma}}$ **T_2= 667.35 K ; T_4=559.26 K**

2. Effective work : ($C_{pair} = C_{pgaz} = 1$ KJ/Kg.K**)**

$$W_{eff} = \eta_{mec,T} C_{p\,gaz}(T_3 - T_4) - \frac{1}{\eta_{mec,C}} C_{p\,air}(T_2 - T_1) = \mathbf{320.95\ KJ/Kg}$$

Heat released by the combustion :

$$Q = C_{p\,gaz}(T_3 - T_2) = \mathbf{611.86\ KJ/Kg}$$

3. Thermal efficiency of the cycle :

$$\eta_{Cycle} = \frac{W_{eff}}{Q} = 0.525$$

Solution of Directed activities Nbre 05 : Steam turbine

Exercise1

1. The thermal efficiency and the steam flow rate in Kg/h.

Point1 (Saturated liquid, P_1=0.08 bar) : h_1=171.68 Kj/Kg and V_1=1.0085 dm^3/Kg

Point2 (Saturated liquid, P_2= P_3=80 bar) : $h_2= h_1+ V_1 (P_2- P_1)$ Kj/Kg

h_2=171.68+1.0085x10^{-3}(80-0.08)x10^5=**179,74 Kj/Kg**

Point3 (Saturated steam, P_3=80 bar) : h_3=2757.72 Kj/Kg and S_3=5.74 Kj/Kg

Point4 (P_1= P_4=0.08 bar) : S_4= S_3=5.74 Kj/Kg

$$X_v = \frac{h_4 - h_4'}{h_4'' - h_4'} = \frac{S_4 - S_4'}{S_4'' - S_4'} = \frac{S_3 - S_4'}{S_4'' - S_4'} = \frac{5.74 - 0,5853}{8,2379 - 0,5853} = 0.67 \text{ } (S_4 = S_3 \text{ isentropic transformation})$$

$$h_4 = X_V h_4'' + (1 - X_V) h_4' = 0.67.2575,73 + (1 - 0.67)171,68 = 1\,782,40 \text{ Kj/Kg}$$

$$\eta_{th} = \frac{\text{turbine work} - \text{pump work}}{\text{heat received}} = 1 + \frac{Q_f}{Q_C} = \frac{(h_4 - h_1)}{h_3 - h_2} = 1 - \frac{1782.4 - 171.68}{2757.72 - 179.74} = 0.37$$

Flow rate: $W_{net} = W_{Cycle} = |W_T - W_P| = |(h_4 - h_3) - (h_2 - h_1)|$

$$P_{net} = Q_m W_{net} \to Q_m = \frac{P_{net}}{|(h_4 - h_3) - (h_2 - h_1)|}$$

$$= \frac{100.10^3}{|(1\,782,40 - 2757,72) - (179,74 - 171,68)|} = 103.4\ kg/s = 103.4x3600 = 372.18x10^5 Kg/h$$

2. Theoretical power of the steam turbine in MW

$$|P_T| = Q_m W_T = Q_m |h_4 - h_3| = 103.4|1\,782,40 - 2757,72| = 100.84 \text{ MW}$$

3. Theoterical power delivred by steam generator SG

$$P_{net} = Q_m (h_3 - h_2) = 103.4(2757,72 - 179,74) = 266.56 \text{ MW}$$

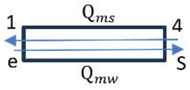
4. Cooling power of the condenser

$$\dot{Q}_{ms} = Q_m (h_1 - h_4) = -166.54\ MW$$

5. Flow rate of the saturated cooling water :

$$\dot{Q}_{ms} = \dot{Q}_{mw} \to Q_{mw} . (h_1 - h_4) = Q_{ms} . (h_{out} - h_{in})$$

$$Q_{mw} = \frac{|\dot{Q}_{ms}|}{(h_{out} - h_{in})} = \frac{166540}{167.57 - 71.38} . 3600 = 6.23.10^6 Kg/h$$



Condenser

Exercise2

1. The steam title X_V (Turbine output)

$$X_v = \frac{h_3 - h_3'}{h_3'' - h_3'} = \frac{S_3 - S_3'}{S_3'' - S_3'} = \frac{S_2 - S_3'}{S_3'' - S_3'} = \frac{6.7228 - 0,4348}{8,4530 - 0,4348} = 0.77 \text{ } (S_2 = S_3 : \text{isentropic transformation})$$

$$h_3 = X_V h_3'' + (1 - X_V) h_3' = 0.77.2555,92 + (1 - 0.77)125,22 = 1996,86 \text{ Kj/Kg}$$

Point4 (Saturated liquid, P_4=0.0425 bar) : h_4=125.22 Kj/Kg and V_4=1.0047 dm^3/Kg

Point1 (Saturated liquid, P_1=P_2=80 bar) : $h_1= h_4+ V_4 (P_1- P_4)$ Kj/Kg

h_1=125.22 +1.0047x10^{-3}(80-0.0425)x10^5=**133,25 Kj/Kg**

Point2 (sperheated steam, P_2=80 bar, T_2=500 °C) : **h_2 =3398.05 Kj/Kg**, **S_2 =6.7228 Kj/Kg**

2. The reversible cycle effciency

$$\eta_{th} = \frac{|W_T|-W_P}{Q_C} = \frac{|h_3-h_2|-(h_1-h_4)}{h_2-h_1} = \frac{|1996,86-3398,05|-(133,25-125,22)}{3398.05-133,25} = 0.426$$

3. The reversible cycle effciency (pump work neglected)

$$\eta_{th} = \frac{|W_T|}{Q_C} = \frac{|h_3-h_2|}{h_2-h_1} = \frac{|1996,86-3398,05|}{3398.05-133,25} = 0.43$$

4. The reversible turbine power

$|P_T| = Q_m\, W_T = Q_m\, |h_3 - h_2| = 60|1996,86 - 3398,05| = 84,07$ MW

5. The irreversible turbine power

$|P_T|_{ir} = Q_m\, W_T = Q_m\, |h_3 - h_2|.\eta_T = 60|1\,782,40 - 2757,72|.\,0,9 = 75,66$ MW

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ملخص

تلعب الآلات البسيطة والحرارية دورًا أساسيًا في تطوير التكنولوجيا وتنفيذ المهام اليومية. الآلات البسيطة، مثل الروافع والبكرات والمستويات المائلة، هي لبنات البناء الأساسية التي، عند دمجها، تشكل آلات أكثر تعقيدًا. إنها تسهل أداء العمل عن طريق تقليل مقدار القوة اللازمة لتحريك الأشياء، مما يسمح بتنفيذ المهام الصعبة بكفاءة أكبر وجهد أقل.

أما الآلات الحرارية فهي الأجهزة التي تحول الطاقة الحرارية إلى عمل ميكانيكي. تشمل الأمثلة الكلاسيكية المحركات البخارية ومحركات الاحتراق الداخلي. وكانت هذه الآلات ضرورية خلال الثورة الصناعية، حيث وفرت الطاقة اللازمة لتحريك القطارات والسفن والمصانع. إن فهم كيفية عمل هذه الآلات لا يساعدنا على تقدير الابتكارات التكنولوجية في الماضي فحسب، بل يسمح لنا أيضًا بتطبيق هذه المبادئ في الاختراعات الجديدة وفي حياتنا اليومية.

Abstract

Simple and thermal machines play a fundamental role in the development of technology and in carrying out everyday tasks. Simple machines, such as levers, pulleys, and inclined planes, are the basic building blocks that, when combined, form more complex machines. They facilitate the performance of work by reducing the amount of force needed to move objects, allowing difficult tasks to be executed with greater efficiency and less effort.

Thermal machines, on the other hand, are devices that convert thermal energy into mechanical work. Classic examples include steam engines and internal combustion engines. These machines were essential during the Industrial Revolution, providing the energy needed to move trains, ships, and factories. Understanding how these machines work not only helps us appreciate the technological innovations of the past, but also allows us to apply these principles in new inventions and in our daily lives.