ROZWÓJ POTENCJAŁU I OFERTY DYDAKTYCZNEJ POLITECHNIKI WROCŁAWSKIEJ

Wrocław University of Technology

Automotive Engineering

Anna Janicka, Czesław Kolanek, Wojciech Walkowiak

APPLIED THERMODYNAMICS INTERNAL COMBUSTION ENGINE

Laboratory

Wrocław 2011

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Contents:

Greek Alphabet

Alpha	Α	α
Beta	Β	β
Gamma	Γ	γ
Delta	Δ	δ
Epsilon	Ε	ε
Zeta	Ζ	ζ
Eta	Η	η
Theta	Θ	θ
Iota	Ι	ι
Kappa	Κ	κ
Lambda	Λ	λ
Mu	Μ	μ
Nu	Ν	ν
Xi	Ξ	ξ
Omicron	Ο	ο
Pi	Π	π
Rho	Ρ	ρ
Sigma	Σ	σ
Tau	Τ	τ
Upsilon	Υ	υ
Phi	Φ	φ
Chi	Χ	χ
Psi	Ψ	ψ
Omega	Ω	ω

Prefixes for the International System of Units SI

The list of the prefixes for the International System of Units (in French: *Système International d'Unités*) officially approved by the International Bureau of Weights and Measures, in French *Bureau International des Poids et Mesures*, BIPM (in Polish: Miedzynarodowe Biuro Miar i Wag).

Official SI Prefixes

Name	Symbol	Multiple	Multiple Name
yotta (Greek *okto* – eight)	Y	10^{24}	septillion
zetta (Latin *septem* – seven)	Z	10^{21}	sextillion
exa (Greek *ex* – six)	E	10^{18}	quintillion
peta (Greek *penta* – five)	P	10^{15}	quadrilion
tera (Greek *teras* – monster)	T	10^{12}	trillion
giga (Greek *gigas* – enormous)	G	10^{9}	billion
mega (Greek *megas* – giant)	M	10^{6}	million
kilo (Greek *khilioi* – thousand)	k	10^{3}	thousand
hector (Greek *hekaton* – hundred)	h	10^{2}	hundred
deca (Greek *deka* – ten)	da	10^{1}	ten
		10^{0}	one
deci (Latin *decimus* – tenth)	d	10^{-1}	one tenth
centi (Latin *centum* – hundred)	c	10^{-2}	one hundredth
milli (Latin *mille* – thousand)	m	10^{-3}	one thusandth
micro (Greek *mikros* – small)	µ	10^{-6}	one millionth
nano (Greek *nanos* – dwarf)	n	10^{-9}	One billionth
pico (Italian *piccolo* – small)	p	10^{-12}	one trillionth
femto (Danish. *femten* – fifteen)	f	10^{-15}	one quadrillionth
atto (Danish *atten* – eighteen)	a	10^{-18}	one quintillionth
zepto (Latin *septem* – seven)	z	10^{-21}	One sextillionth
yocto (Greek *okto* – eight)	y	10^{-24}	one septillionth

1. INTRODUCTION

Thermodynamics, as a field in mechanics, is the first discipline in thermal techniques, including not only thermal power engineering (boilers, turbines, steam engines, pumps), but also other industries, like refrigeration engineering, cryogenics, internal combustion engines, etc.

Classical thermodynamics divides into statistical, technical and chemical; the two latter qualify as applied thermodynamics because on top of basics in theory they also include basic usage in techniques and chemistry.

Phenomena in thermodynamics are described in a statistical or phenomenological way.

Thermodynamic phenomena, taking place in continuum, are described using the phenomenological method, resulting directly from an experiment. They can also be described using the statistical method. Media containing small number of particles or media of low density are better to be described with the statistical methods. In this case, the results of the statistical contemplation are experimentally tested.

The fundamental laws of thermodynamics have been known since XIX century in a form of the first, second and third law of thermodynamics. Later on there was also the zeroth law of thermodynamics introduced to define the temperature compensation relation. The Onsager a reciprocal relation is regarded the Fourth Law of Thermodynamics.

The basic concepts are accepted as evident. Depictions, in some way specific for most of authors, as well as axiomatics, regard the development of the formal thermodynamics. The subject matter development consists in learning new laws and establishing the interrelations between the basics and applications of thermodynamics.

The term thermodynamics derives from Greek words:

τερμοσ (warm)
δυναμiς (force);

introduced into studies when the notion of work as a result of force and the force itself were still indistinguishable.

The devices (power plants, refrigerators, compressors, pumps, furnaces, heat exchangers, engines and other machines), that function basing on heat and force, are called thermodynamic devices. Because the relation between heat, force and energy is quite close, the term “power engines” applies as well.

In those devices, the important role of energy conductor falls onto so called thermodynamic factor, which can be any physical body - usually liquid, like gas or fluids. Therefore the interrelation laws and qualities of those factors are always deeply studied.

Basic terminology

The surrounding world is built of matter. The matter that has the rest mass is called a substance (G). The matter without the rest mass is called radial matter, e.g. photons of electromagnetic radiation (they have got mass but no substance).

The substance quantity can be measured by the number of molecules – gram-molecule or rest mass in standardized thermal conditions (so called kilogram).

Since in technology speed not greater then speed of light (1/100) is quite common, and the standardized thermal conditions have no influence on mass, in practice substance **G** equals mass **m.**

Thermodynamics uses terminology differing from other terminology in other science fields therefore requires a separate presentation. Units of measure applying to a entire object are marked with capital letters, e.g. $\boldsymbol{V}$, $\boldsymbol{Q}$, $\boldsymbol{U}$, $\boldsymbol{I}$ etc. Units measure, depending on the object quantity, so called extensive quantities, referring to the substance amount $\boldsymbol{m}$, are marked with a small letter, e.g. $\boldsymbol{v} = V/m$, $\boldsymbol{q} = Q/m$, $\boldsymbol{u} = U/m$, $\boldsymbol{i} = I/m$ etc. and additionally they have a term for all specific measures, e.g. **v** – specific volume, $\boldsymbol{i}$ – enthalpy, the only exemption is $\boldsymbol{q}$, called individual heat or kilogram heat, because the term *specific heat* is commonly used to describe another thermodynamic unit.

If a given unit, dependant on time, relates to it, it is usually referred to as a stream of measure or its intensity and marked with an apostrophe (like time derivative in Mathematics), e.g.: $Q' = Q/\tau$ heat stream, I' – enthalpy stream, V' – volume stream or volume intensity flow, $m' = m/\tau$ – mass stream or mass intensity flow.

1.1 Table compiles markings and units for the key physical measures used in thermodynamics. It also contains remarks regarding applied terminology, which is not always explicit in the field of thermodynamics.

Table 1.1. Key markings and measures

Physical Measure	Symbol	Unit	Derivative Unit	Remarks
heat	Q $q=Q/m$ $Q'=Q/\tau$	J J/kg W	kJ kJ/kg kW	 q – individual heat Q’ – heat stream
work	L	J	kJ, Wh	
energy	E	J	kJ	
power	N	W	kW	
time	τ	s	h	h - hour
mass	m	kg	Mg	
force	K	N	kN	
volume	V $\upsilon=V/m$ V'	m^3 m^3/kg m^3/s		 specific volume volume intensity flow
pressure	p	Pa	MPa	
temperature	T	K	°C	

I. UNDERLYING ISSUES

2. BASIC CONCEPTS

2.1. State marks

To determine an object and differentiate it from others, the object's qualities are used, so called marks, like length, width, colours, temperature, pressure, smell, etc.

Among those there are some that can be described in numbers, e.g. length, width, temperature, pressure, etc. - they are called **quantitative marks** and they are quantities. The other marks, impossible to be described with numbers, e.g. shape, smell, colour, are called **qualitative marks** – they are not quantities.

Quantitative marks include marks that never change, e.g. gas constant, Avogadro constant, and those that may vary, like pressure, temperature or volume. Quantitative marks able to change are important for determining object's state and for that reason they are called **state marks**.

Two objects are in the same state only if the group of all state marks describing the objects is adequately equal, which means that the **object's state** is determined by the group of state marks.

State marks, taken as independent variables are called state **parameters**, and the dependent marks – **state functions**. Not all state marks are independent from one another. Describing a random object's state, it is possible to freely choose only a certain number of state marks, the rest of them depends on the chosen ones. This minimal number of parameters that can be chosen independently to describe any state is called **absolute state function system** (e.g. for ideal gas it is p-υ, T-s, or i-s). The state function described by an absolute state function system is called **state equation** (e.g. for ideal gas it is $v=RT/p$)

2.2. Pressure

Pressure is a very important state mark. If pressure of surface F is marked as K, than the local pressure p is:

$$p = \frac{dK}{dF} \tag{2.1}$$

If pressure is spread evenly than local pressure p is:

$$p = \frac{K}{F} \tag{2.2}$$

The first pressure unit is **1Pa** (pascal), which is 1 N/m^2. This unit is too small to stand for most of the pressures present in technology therefore its multiples are used, e.g. megapascal (**1 MPa** = 10^6 Pa). **1 bar** = 10^5 Pa is an auxiliary unit.

Due to frequent usage of measuring devices, called differential manometers, which measure pressure by the difference in manometer fluid level, it is allowed to present pressure in mm on a column of fluid. This kind of pressure is marked ***h***, except for when its represents the surroundings pressure, called barometric pressure, because than it is marked ***b***. Fluid column pressure of 1mm Hg at 0 °C with the usual gravitation acceleration is marked as 1 thorium (**1Tr**), after Torricelli.

The value of measured pressure depends on not only the factor state, but also on the conditions in which the measurement has been taken.

Pressure measured with a manometer $\boldsymbol{p_m}$ defines the relation between $\boldsymbol{p}$ and $\boldsymbol{p_o}$:

$$\boldsymbol{p_m} = \boldsymbol{p} - p_o \tag{2.3}$$

given:

p_m – manometric pressure,
p_o – barometric pressure,
p – absolute pressure (pressure found in vaccum).

Depending on whether the measured pressure $\boldsymbol{p}$ is lower or higher from $\boldsymbol{p_o}$, the manometer can show positive manometric pressure $\boldsymbol{p_{m1}}$ (called excess pressure) negative pressure p_{m2}, presented in the pressure diagram (Fig. 2.1).

Gas absolute pressure is calculated by adding manometric pressure to environment pressure. Environmental pressure is measured usually separately, using barometers, which work in a similar way to Torricellie's tube, or devices which base on a material elasticity phenomenon, called aneroids.

The value of manometer measured gas pressure also depends on whether its macroscopic velocity varies from zero or is static.

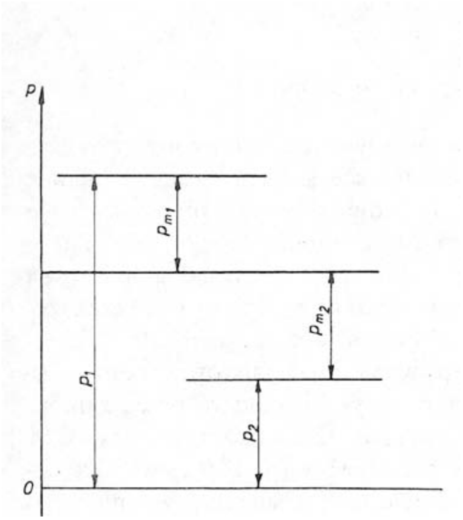
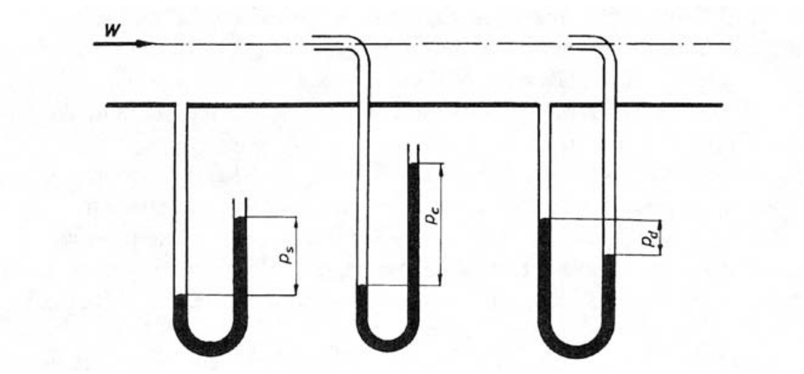


Figure 2.1. Manometric and absolute pressure on the pressure chart

If the pressure measurement is taken when the object is in motion, depending on the method used (Figure 2.2), there are three types of (manometric or absolute) pressure distinguished: static, dynamic and total pressure.

If the pivot of the pressure conducting impulse tube is directed towards the fluid velocity vector „w", the devise connected to the tube will measure the static pressure. If the tube pivot is consistent with the direction of the vector „w" and the tube exit is reversed to the vector direction, the devise connected to the tube will measure the total pressure. The devise connected to the both impulse tubes will show the dynamic pressure.



Figur. 2.2. Measuring of the in-motion fluid pressure: p_s *– static pressure,* p_d *– dynamic pressure,* p_c *– total pressure*

As it results from this method:

$$p_c = p_s + p_d \tag{2.4}$$

If the flow is disturbed, the static pressure can be measured only when none of the velocity "w" factors affects the impulse tube or the manometric fluid.

The flow theory suggests the following interrelation between dynamic pressure and velocity:

$$p_d = \frac{\gamma w^2}{2g} \quad , \tag{2.5}$$

in which

γ- specific weight of fluid, N/m^3,
w –flow velocity, m/s,
g - gravitational accelaration, m/s^2.

The interrelation (2.5) is used to define the fluid velocity by direct measurement of dynamic pressure p_d, as a subtraction p_c-p_s using so called accumulating tubes – the most frequently used are Prandtl tube, focusing on dynamic pressure p_d, and Pitot tube, focusing on total pressure p_c. (Figure 2.3).

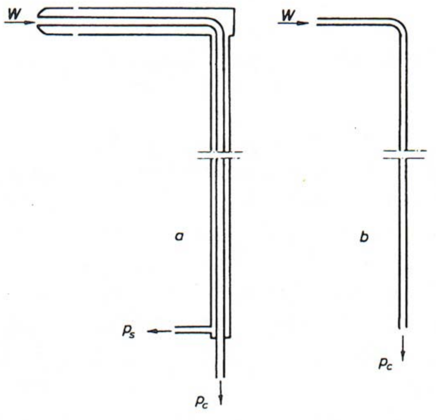


Figure 2.3. Accumulative tubes a – Prandtl tube, b – Pitot tube

Fluid pressure, as a state mark (resulting from the microscopic movement of the fluid particulates), is the total static pressure. It occurs in equations of state and all the relating to ones.

2.3 Temperature

Temperature is a strong thermodynamic measure. Thermodynamic scale *T* is the primary temperature scale, relating to the notion of entropy *S* and internal energy *U* by the means of fragmentary derivative, given the constant volume *V*, as in the following equation:

$$T = (\delta U/\delta S)_v$$

When choosing the temperature scale it is important to establish the start (zero) on the scale and the unit.

To measure temperature, any physical system can be used, provided that at least one of its state marks changes with the temperature change, and the rest of them stay approximately unchanged. This measure is called a thermometric parameter.

Another important thing when choosing the temperature scale is the presence of two constant temperature points. Experiments have proven that there are some processes occurring at set temperature. They include melting, boiling (at constant pressure) or so called triple point – gas, liquid and solid coexisting at the same time. Processes of this kind could be used as the foundation for finding the characteristic points of temperature.

Many experiments prove that perceptible temperature change like hot – cold, lead to altering some of the characteristics. The easiest way to describe such changes, given the correlation between characteristic x and quality X, is as a direct proportion:

$$t=Ax \qquad (2.6)$$

Thermodynamic property introduced by R. Clausius

or a linear equation:

$$t = A'x + B' \qquad (2.7)$$

or as a square equation:

$$t = A''x^2 + B''x + C'' \qquad (2.8)$$

To create a temperature scale for the equation in example 2.6 it is necessary to define one temperature point, in 2.7 – two points, in 2.8 – three points.

Thermoelectric thermometers (thermocouples) have become widely used for temperature measuring employing the thermoelement phenomena. The thermoelement consists of two thermoelectrodes, made of different metals or metal alloys; they are connected end to end, forming a measuring weld and the other sides of the electrodes are the voltage measurement points (Figure 2.4).

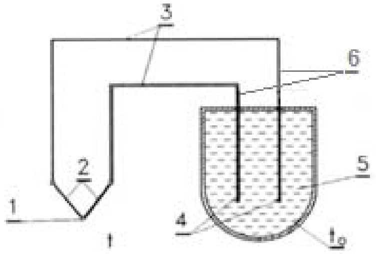


Figure 2.4. Thermoelectric thermometer diagram: 1 – measurement weld, 2 – thermoelectrodes, 3 – compensatory leads, 4 - so called loose ends, 5 – thermostat, 6 – milivoltometer connection, t - measured temperature, t_o - temperature of reference

Measuring temperature using this type of thermometers is based on the phenomenon of creating electric power in a closed circuit comprising of two thermoelectrically diverse conductors while the measurement weld and the loose ends are connected to sources of opposite temperatures.

If the temperature of the measurement weld (t) is higher than the temperature of the galvanometer connected loose ends (t_o), the closed circuit will produce thermoelectric force (TEF) E proportional to temperature difference ($t-t_o$):

$$E = f(t-t_o), \tag{2.9}$$

Value of which depends only on the type of material used to make the thermoelements and the temperature difference.

The type of material also influences the range for measured temperature. Mostly used materials include:

- platinum-platinum/rhodium,
- nickel-chromium/nickel
- copper-constantan,
- Iron-constantan, .

The basic thermoelement data are presented in Table 2.1 and Figure 2.5.

Table 2.1. Qualities of some of the thermoelements:

Electrode material	Thermoelectric force given t=100ºC i t_o=0ºC, mV	Application limits, ºC		
		continuous measuring		momentary measuring
		lower	upper	
platinum-	0,64	-20	+1300	+1600

platinum/rhodium,				
nickel-chromium/nickel	4,10	0	+1000	+1300
copper-constantan,	4,25	-200	+400	+600
Iron-constantan,	5,37	-200	+600	+900

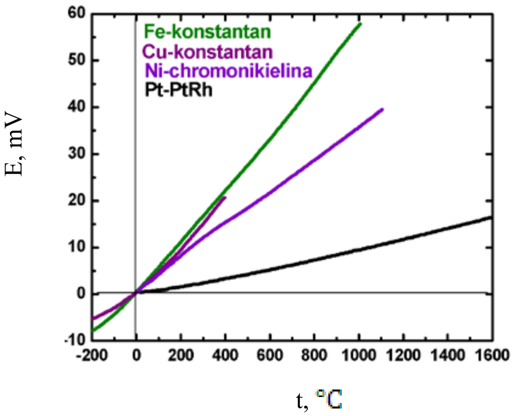


Figure 2.5. Thermocouple characteristics (electric current generated at the bonds between the thermocouples while one of the bonds is put in icy water)

The diagram for the thermoelectric thermometer measuring system with a magnetoelectric milivoltometer is presented in Figure 2.6.

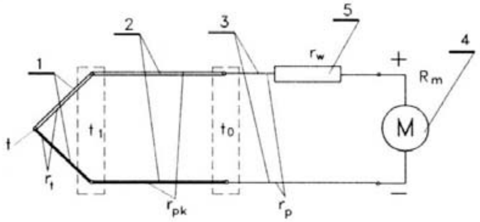


Figure 2.6. The thermoelectric thermometer measuring system with a magnetoelectric milivoltometer: t – measuring weld temperature, t_1 – temperature of the thermoelectrode weld, t_o – junction temperature, 1 – thermoelectrodes, 2 – compensatory leads, 3 – junctions, 4 – milivoltometer, 5 – compensatory resistor (in case of using a different to the compensatory method)

It is to be remembered that milivoltometer measures electric voltage, not thermoelectric force of a thermoelement.

The entire circuit resistance comprises of measuring instrument resistance R_m and the external circuit resistance R_z (thermoelement resistance r_t, compensatory lead resistance r_{pk}, junction resistance r_p, compensatory resistance r_w).

Following the description of Figure 2.6., it can be assumed:

$$R=R_m + R_z \qquad (2.10a)$$

$$R_z=r_t + r_{pk} + r_p + r, \qquad (2.10b)$$

The correlation between TEF (E) and the voltage connected to the milivoltometer clamps U can be calculated with this formula:

$$E= U (R/R_m) \qquad (2.11)$$

It results from this formula that E can be calculated by measuring U, knowing the value of the milivoltometer internal resistance R_m (stated at the device information plate) and the resistance value of the thermoelement measuring circuit R_m. If the resistance R_z is much lower than R_m ($R_z \ll R_m$), it is sufficient to calculate the resistance value for R_z at a chosen temperature and take for constant at the specific temperature range; this will give a precise enough result for E. It is safe to assume, that

$$(R_m + R_z)/R_m = C, \qquad (2.12)$$

And the formula (2.11) transforms into:

$$E = U\,C \qquad (2.13)$$

One of the thermoelectric thermometer advantages is small surface of a junction connected to the object of which the temperature is being measured, as well as the relaxation time. The disadvantages enlist the necessity for temperature compensation (keeping the cold junction at constant temperature by cooling it down in an ice bath or with electronic methods).

2.4 Thermodynamic factor

Thermodynamic calculations use the following thermodynamic factors:

1. Ideal gas is a theoretical gas that obeys the ideal gas law (stated by Emile Clapeyron) and its specific heat at constant volume is constant c_v= const. Other qualities are discussed further on in this publication. Ideal gas does not exist in nature.

2. Semi-ideal gas – is gas that does not obey the ideal gas law, but its specific heat at constant volume is presented with this temperature function: $c_\upsilon = c_\upsilon(T)$. This kind of gas does not exist in nature.

3. Van der Waals gas – it does not obey the ideal gas law but it has more qualities of real gas than the previous two. This kind of gas does not exist in nature.

4. Real gas – it does not obey the ideal gas law or the Van der Waals model; specific heat c is an equation of many variables, although, depending on the factor state and the accuracy of the problem to be solved, under some circumstances, the real gas can be treated as ideal or semi-ideal gas.

a) If the real gas has a physically homogenous gas phase, the distance between the molecules is big (adequately mild temperatures and low pressures) and the calculations are only approximate, it is accepted to treat real gas as ideal;

b) if the real gas has a homogenous gas phase, temperature fluctuations are so big, that, at this temperature range, it is impossible to take the specific heat c_υ for constant heat and the calculations are more accurate (e.g. structural), it is accepted to treat real gas as semi-ideal gas (real gas is similar to semi-ideal gas only if $p>0$).

2.5. The equation of state for ideal and semi-ideal gas

Boyle-Mariotte’s Law propose a gas relation that forms into the ideal gas law:

$$pv = RT \tag{2.14}$$

given:

p – pressure (absolute), N/m^2,

υ – specific gas volume, m^3/kg,
T – absolute gas temperature, K,
R – individual gas constant, J/(kg • K).

If both sides of that equation are multiplied by mass **m**, it gives:

$$pV = mRT, \quad (2.15)$$

given:
V – total gas volume.

If gas amount is given in kilo moles ($m = nM$), then the equation (2.15) changes into:

$$pV = nMRT, \quad (2.16)$$

given:
n – number of gas kilo moles
M – mole number (mass)

Product MR for all gases is constant and is called **molar gas constant** (also called universal or ideal gas constant):

$$MR = 8315 \frac{J}{kmol \cdot K} \quad (2.17)$$

Considering (2.16) and (2.17) it gives the ideal gas law equation:

$$pV = n\ 8315\ T \quad (2.18)$$

Defining the **specific mole volume** as $\Phi = V/n$, the equation (2.18) equals:

$$p\Phi = 8315\ T, \quad (2.19)$$

it says that one kilogram of any gas in given thermal conditions (i.e. at the same p and T) takes up the same space (Avogadro's law).

Assuming that the accepted physical conditions are as normal – pressure 1013 hPa (760 T) and temperature 273,2 K – the equation (2.19) says that in those conditions 1 kilogram of any gas takes up 22,4 m^3.

That gives the formula allowing converting the number of kilo moles into conventional (normal) cubic meters (um^3):

$$V_u = 22{,}4\ n \quad (2.20)$$

The mixture composition (Figure 2.7) can be determined using so-called volumetric, weight (mass) and molar shares.

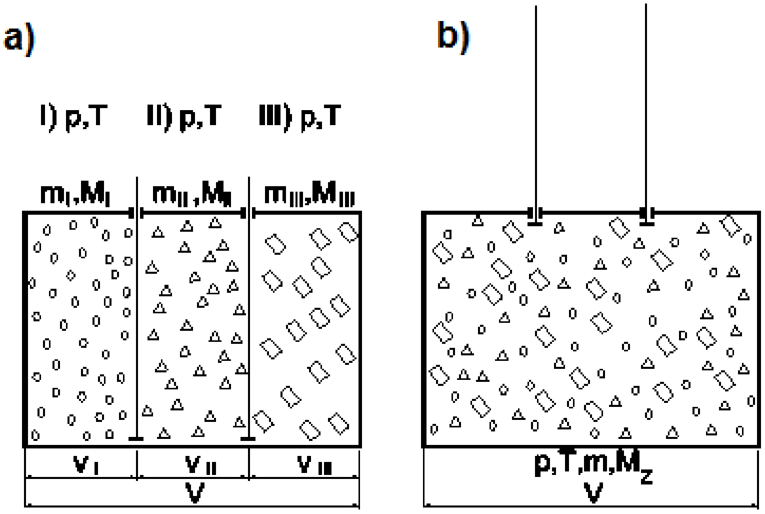


Figure. 2.7. Mixing components: a – state before mixing, b - state after mixing.

The **volumetric share** of a give mixture component is defined as **a ratio of the component volume to the mixture volume**.

Those are measured in the same thermal conditions. It can be written down as follows:

$$r_i = \left(\frac{V_i}{V}\right)_{pT} \tag{2.21}$$

$$\sum r_i = \sum\left(\frac{V_i}{V}\right)_{pT} \tag{2.22}$$

The **weight (mass) share** of a given component is a ratio of its weight ($\boldsymbol{m_i}$) to the mixture weight ($\boldsymbol{m}$), in kilograms:

$$g_i = \frac{m_i}{m} \tag{2.23}$$

$$\Sigma g_i = \Sigma \frac{m_i}{m} = 1 \tag{2.24}$$

The **mole share** of that component **(z)** is a ratio of its amount (***n_i***) in kilo moles to the mixture amount **(n)** in kilomoles:

$$z_i = \frac{n_i}{n} \tag{2.25}$$

$$\Sigma z_i = \Sigma \frac{n_i}{n} = 1 \tag{2.26}$$

If the mixture complies with the ideal gas law than the mole share equals volumetric share:

$$z_i = \frac{n_i 8315T}{n8315T} = \frac{pV_i}{pV} = \left(\frac{V_i}{V}\right)_{pT} \tag{2.27}$$

and

$$z_i = r_i$$

The air mixture can be approximately presented as:

$r_{O2} = 0,21$ $g_{O2} = 0,23$

$r_{N2} = 0,79$ $g_{N2} = 0,77$

To be able to apply the ideal gas law to gas mixtures in a form of equations (2.14 or 2.15), a supplementary gas constant ***R_z*** or supplementary mixture mole number has to be calculated ***M_z***.

IT IS ASSUMED THAT THE PRODUCT OF THE SUPPLEMENTARY GAS CONSTANT AND THE SUPPLEMENTARY MOLE NUMBER EQUALS THE **IDEAL GAS CONSTANT.**

The ideal gas law applied to a gas mixtures and any mixture component:

$$pV = mR_zT \tag{2.28}$$
$$pV_i = m_iR_iT \tag{2.29}$$

After the equation transformation (2.29)

$$p\sum V_i = T\sum m_iR_i, \tag{2.30}$$

and after dividing both sides of the equation by the equation (2.28):

$$\frac{p\sum V_i}{pV} = \frac{T\sum m_iR_i}{TmR_z} = 1 \tag{2.31}$$

remembering that $\Sigma Vi = V$.

The supplementary gas constant equals:

$$R_z = \frac{\sum m_iR_i}{m} = \sum g_iR_i$$

given:

$$g_i = m_i / m$$

is the weight share (mass).

In a similar way it can be proven, that

$$M_z = \sum r_iM_i \tag{2.32}$$

therefore supplementary mole number equals the sum of the products of the mixture component volumetric shares and their mole numbers.

2.7. Thermodynamic processes

There is a gaseous object in a cylinder (Fig. 2.8). Measurements describing this object (e.g. p and T) have a definite value all the time.

Set of all those measurements describing the object in any moment determine the state of the object.

The number of the transitional states (from the beginning state 1 till the final state 2) is called a **process**.

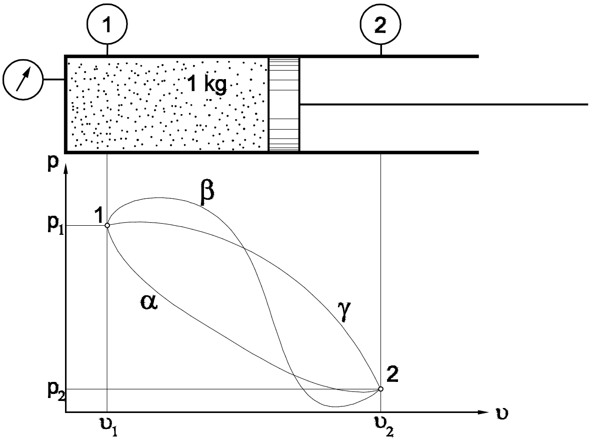


Fig. 2.8. The object states and the processes in a system of coordinate's $\mathbf{p} - \boldsymbol{\upsilon}$

If any object what in a beginning state presented in the point 1 (Fig 2.8), i.e. its pressure was p_1, specific volume υ_1, and in the end the pressure and volume changes to p_2 and υ_2, which corresponds with point 2, it means that the object has gone through the states of transitional pressures between p_1 and p_2 as well as transitional specific volumes between υ_1 and υ_2.

It means that for an object state to change, a process has to occur.

From a number of accepted processes, there are a few that are called characteristic processes, during which states change in a complex way, e.g.:

- \- for constant pressure – isobaric process,
- \- for constant volume – isochoric process,
- \- at constant temperature – isothermal process,
- \- for constant enthalpy – isoenthalpic (isenthalpic) process, etc.

3. WORK

The concept of work was introduces into mechanics to describe the effect of any force acting through a distance and causing movement (Fig. 3.1).

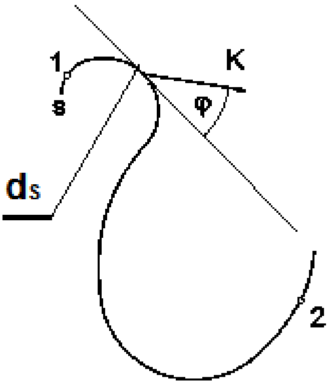


Figure 3.1. Work of the concentrated force ***K*** *at a distance* ***s*** *between states 1 and 2*

The definitional formula is:

$$L_{1-2} = \int K \cos\varphi \, ds \qquad (3.1)$$

where:
K - force,
φ – angle between the force vector and the momentary displacement, s - displacement.

Figure 3.2 shows a cylinder-piston system filled up with a thermodynamic factor, in which there is balance between the force K, acting on the piston rod and the mass force K_Q. There is no friction in this device.

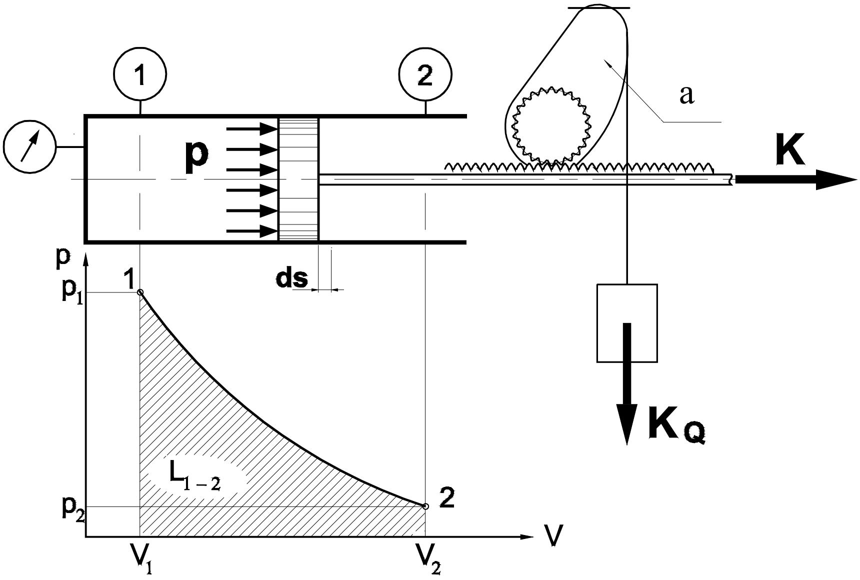


Figure 3.2. Cylinder-plunger system: 1, 2 – border states of a process: a – cam

The absolute pressure of the thermodynamic factor inside the cylinder acts on the piston rod with concentrated force *K*. Moving the rod, according to the equation (3.1), the following work is done:

$$L_{1-2} = \int K ds, \qquad (3.2)$$

Force *K* can be described as:

$$K = pF \qquad (3.3)$$

given F – a rod surface.

If the product *Fds* = *dV* (compare Fig. 3.2), the relation defining **absolute work** is:

$$L_{1-2} = \int_1^2 pdV. \qquad (3.4)$$

The equation (3.4) is the **primary equation** in thermodynamics and it defines the **work of the volume change**.

If the device (Fig. 3.2) is in an environment at pressure $\boldsymbol{p_0}$, the displacement is caused not by force ***K*** but ***K'*** (compare Fig. 3.3):

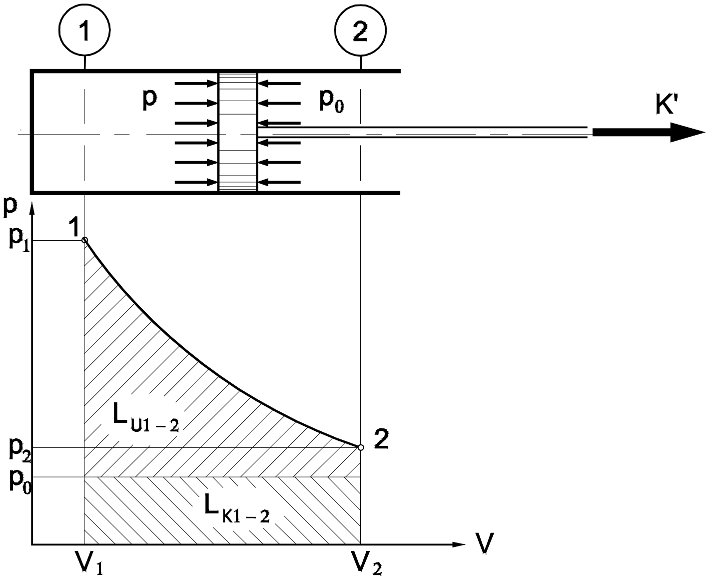


Figure 3.3. Useful work of the process

Work **used** to lift the weight $\boldsymbol{K_Q}$ is:

$$L_{u1-2} = \int_1^2 K'ds \tag{3.5}$$

Given $K' = (p - p_o)F$, $Fds = dV$, then

$$L_{u1-2} = \int_1^2 (p - p_0)dV \tag{3.6}$$

Equation (3.6) determines so-called **useful work**.

The integral on the right hand side of the equation (3.6) can be presented as a difference of integrals:

$$L_{u1-2} = \int_1^2 pdV - \int_1^2 p_0 dV \tag{3.7}$$

In this equation

$$\int_1^2 pdV$$

Is the known **absolute** work, and at constant environment pressure the integral

$$\int_1^2 p_0 dV = p_o(V_2 - V_1)$$

and it represents the work of gas L_{k1-2}, used to compress the environment, therefore the equation (3.7) could change into

$$L_{u1-2} = L_{1-2} - L_{k1-2} \tag{3.8}$$

In a real cylinder there is always some friction, on the inside as well as inside the mechanism, e.g. the piston rod rubbing against the walls of the cylinder.

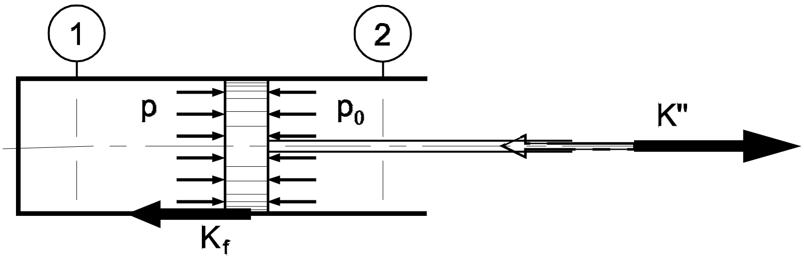


Figure 3.4. The influence of the friction K_f on the cylinder-piston system.

If the force K_f represents the summary of the **generalized frictional force** occurring in the cylinder-piston system (including friction inside the mechanisms as well as inside the fluid), after the movement from position 1 to 2, the work done to overcome the **external** force acting on the gas will be decreased by the work done to overcome the **frictional** force:

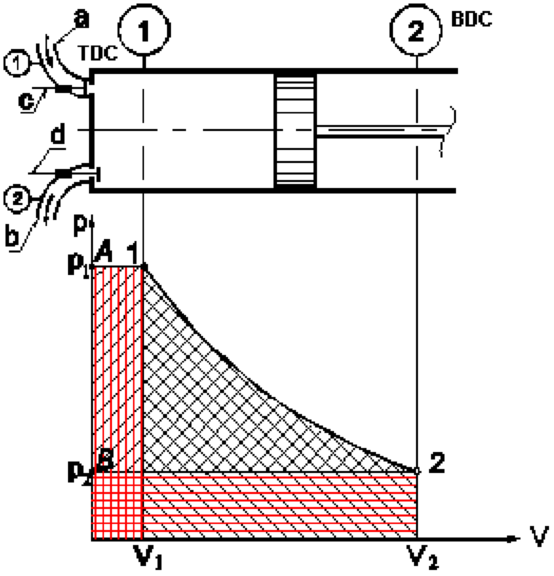
$$L_{z1-2} = L_{u1-2} - L_{f1-2} \tag{3.9}$$

The **thermodynamic factor** is delivered into technical devices and flow engines through **isobaric “input”** and **isobarically** removed by so-called „**exhale**” Both processes influence the quantity of the work done, it can be observed in the functioning of a so-called **ideal flow engine** (Fig. 3.5).

The ideal flow engine characteristics:

- compressive capacity equals zero,
- there is no friction of any kind inside an engine, so there is no pressure drop in the valves,
- during the input and exhale the heat does not travel,
- the piston is tight.

The work cycle of such an ideal flow engine is illustrated by Fig. 3.5.



Figur. 3.5. The ideal flow engine diagram: BDC – bottom dead centre, TDC – top dead centre, a – connecting lead for the thermodynamic factor in state 1, b – connecting lead for the thermodynamic factor in state 2, c – inlet valve, d – outlet valve,:

▨ $\equiv L_{A-1} = p_1V_1$ ▧ $\equiv L_{2-B} = p_2V_2$ ▥ $\equiv L_{1-2} = \int_1^2 pdV.$ ▤ $\equiv L_{t1-2} = -\int_1^2 V$

Any volume and pressure changes occurring during the work cycle have characteristic names, connected to the change nature, as in Fig. 3.6.

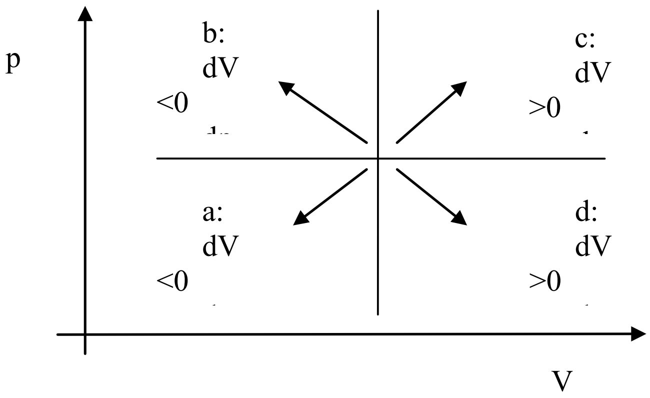


Figure 3.6. Thermodynamic processes in the system of coordinates p-V:
dV>0 – expansion, dV<0 – compression, dp>0 – reduction, dp<0 – decompression

The work L_{A-1} and $L_{1-2,}$ done while the factor expansion, is **positive** (*dV*> 0) and work done while compression is **negative** (*dV< 0)*.

The algebraic sum of those works is defined by *A-1-2-B* field and in this case it is positive, i.e. work done by a machine is "received" – a machine is **an engine**. Devices requiring some "input" into their driving force, like ventilators, compressors, etc., are called **working machines**.

The sum of work in the **isobaric input**, **1-2 process** and **isobaric exhale** is called **technical work**.

$$L_{t1-2} = L_{A-1}+L_{1-2}-L_{2-B} = L_{1-2}+ p_1V_1 -p_2V_{2,} \qquad (3.10)$$

which has the following mathematical equivalent:

$$L_{t1-2} = -\int_1^2 Vdp. \qquad (3.11)$$

The integral value (3.11) can be calculated if 1-2processes are given.

In every **flow** machine or device the work is always equals **technical work** as a total of input, change and exhale works.

If the work done in flow machines is encumbered with friction, that the friction is present in every work phase, i.e. during input, change and exhale.

This is **friction** work $\boldsymbol{L_{fc}}$ per cycle.

In flow machines (engines and working machines) **external** work $\boldsymbol{L_z}$ equals technical work diminished by work $\boldsymbol{L_{fc}}$,, which is:

$$L_z = L_{tl\text{-}2} - L_{fc}, \tag{3.12}$$

In working machines the **external work** $\boldsymbol{L_z}$, which is the algebraic total of the **negative technical work** diminished by the **friction work**, treated as a positive quantity, is called **motive work** $\boldsymbol{L_n}$:

$$\boldsymbol{L_n = - L_z} \tag{3.13}$$

4. ENERGY BALANCE

The rule for the energy saving:

„ THE ENERGY OF AN ISOLATED SYSTEM IS CONSTANT”,

where "isolated" means " not exchanging energy with the environment".

To isolate a system from the environment, the **concept of a balance shield** was introduced.

Objects inside the shield are a **system** - outside - **environment**. The balance shield is a **theoretical** enclosed surface – it does not change or restrict any of the phenomena.

An exemplary energy balance is presented by Fig. 4.1.

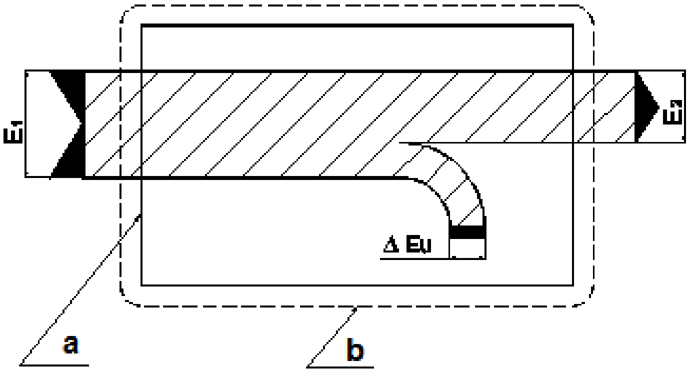


Figure. 4.1. The energy balance presented as a Stankey diagram: a – balanced system, b – balance shield

If the system exchanges energy with the environment, the **system energy increase (change)** Δ***Eu*** during the phenomenon can be calculated using the energy balance equation:

$$\Delta E_u = E_1 - E_2 , \qquad (4.1)$$

given:
E_1 – so-called „imported energy" into the system,
E_2 - „exported energy" from the system.

Transferring energy through a pipeline is commonly applied method for importing/exporting the energy from the system using another medium.

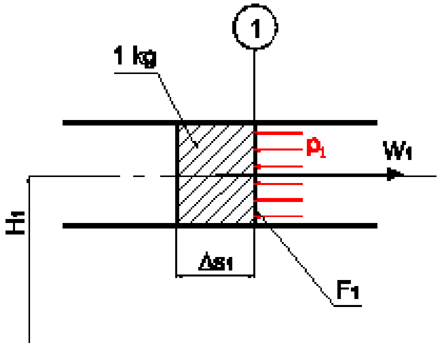


Figure. 4.2. Energy transfer through a pipeline

If the factor coming onto the system has volume ***m***, than the quantity of the energy $\boldsymbol{E_r}$ delivered with the factor is proportional to volume ***m***.

$$E_r = m\, e_r$$

The factor **mass *m*=1kg**, coming into the system at height ***H*** and with speed ***w***, brings kinetic energy.

$$m\, w^2/2$$

as well as relative potential energy

$$m\, g\, H$$

To transfer a **factor** with mass ***m*=1kg** through a section ***F*** at pressure ***p***, that factor has to be moved by a path elementΔs, which means that **work (pressing)** has to be applied

$$p\, F\, \Delta s/m = p\, V/m$$
$$p\, V/m = p\, \upsilon$$

where $V = F\, \Delta s$

The factor mass ***m*=1kg** also brings in the energy related to the state of object's microparticles, their motion, so-called internal energy ***u***,–that is the amount of the actual energy $\boldsymbol{e_r}$ delivered with the factor is:

$$e_r = u + p\upsilon + \frac{w^2}{2} + gH. \qquad (4.4)$$

where ***u*** – actual internal energy.

The internal energy U is an extensive thermodynamic function (state function). It is safe to assume:

$$U = m\, u$$

In many cases in technology, if $w < 50$ m/s and the relative height is $H < 50$ m

$$\frac{w^2}{2} + gH << u + p\upsilon \qquad (4.5)$$

that is

$$e_r \approx u + p\upsilon \tag{4.6}$$

The sum of u+pv is treated as a definitional equation of (actual) **enthalpy *i***:

$$\boldsymbol{i = u + p\,\upsilon} \tag{4.7}$$

Enthalpy is also the state function, since $\boldsymbol{u}$ as well as $\boldsymbol{p}$ and are functions or state marks.

The equation (4.4) transforms into:

$$e_r = i + \frac{w^2}{2} + gH \tag{4.8}$$

and respecting conditions (4.5.)

$$E_r \approx mi = I \tag{4.9}$$

The **energy** imported through the **pipeline** by the factor is approximately equal the **enthalpy** of that factor.

5. FIRST LAW OF THEROMDYNAMICS

In a cylinder there is gas which is in a process.

As a result of external heat $Q_{z1\text{-}2}$ added to the gas, the state has been changed and the piston has been moved 1-2 (gas performed external work $L_{zl\text{-}2}$)

The energy balance for the system is presented in Fig. 5.1.

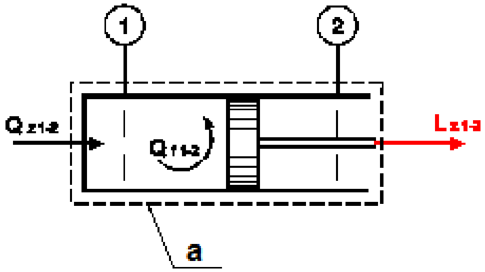


Figure 5.1. Cylinder-plunger-gas system: a – balance shield

In this particular case the imported energy is $E_1 = Q_{z\,l\text{-}2}$; the exported energy is $E_1 = Q_{z\,l\text{-}2}$ therefore, basing on the equation (4.1) it is:

$$\Delta E_u = Q_{z1\text{-}2} - L_{z1\text{-}2}\,, \qquad (5.1)$$

With the **semi-static** (almost static) piston movement and the same energy of the cylinder walls and the piston:

$$\Delta E_u = U_2 - U_1\,,$$
(5.2)

where:
U_2 - internal gas energy at the end of the transition,
U_1 - internal gas energy at the start of the transition.

The two equations (5.1) and (5.2) give a relation:

$$Q_{z1\text{-}2} = U_2 - U_1 + L_{zl\text{-}2}\,,$$
(5.3)

which is called the **first law of thermodynamics**, written down using the external heat and external work.

If in the system, presented in Fig. 5.1 appears some friction, than the following equation is in place:

$$Q_{f1\text{-}2} = L_{f1\text{-}2} \qquad (5.4)$$

Work used to overcome the friction work while displacement 1-2 equals friction heat.

Adding equation (5.4) to (5.3) bilaterally gives:

$$Q_{z\,1\text{-}2} + Q_{f\,1\text{-}2} = U_2 - U_1 + L_{z\,l\text{-}2} + L_{f\,l\text{-}2}. \qquad (5.5)$$

The sum of the external heat and the heat created by friction inside the system gives **heat of transition $Q_{1\text{-}2}$**.

The sum of $L_{z1\text{-}2} + L_{f1\text{-}2}$, according to equations (3.9) and (3.8), equals the **absolute work** of transition $L_{l\text{-}2}$, as the result equation (5.5) forms:

$$Q_{1\text{-}2} = U_2 - U_1 + L_{1\text{-}2}\,,$$

(5.6)

the mostly used form of the equation of **the first law of thermodynamics (ILT)**.

If the absolute work is presented with the transformed equation (3.10)

$$L_{1\text{-}2} = L_{t\,1\text{-}2} + p_2\, V_2 - p_1\, V_1 \qquad (5.7)$$

and put into (5.6) then

$$Q_{\,1\text{-}2} = U_2 + p_2\, V_2 - U_1 - p_1\, V_1 + L_{t\,l\text{-}2}\,, \qquad (5.8)$$

can be written down as

$$Q_{1\text{-}2} = I_2 - I_1 + L_{\,t\,1\text{-}2} \qquad (5.9)$$

This equation is called **the second form of the first law of thermodynamics (IILT)**.

Both of the ILT equations (5.6.) and (5.9.) can be written as differentials:

$$\text{đ}\,Q = dU + \text{đ}\,L\,, \qquad (5.10)$$
$$\text{đ}\,Q = dI + \text{đ}\,L_t\,, \qquad (5.11)$$

Where $\text{đ}\,Q$, $\text{đ}\,L$i $\text{đ}\,L_t$ are differential equations, used to calculate adequately heat, absolute work and technical work (e.g. $\text{đ}\,L = pdV$, a $\text{đ}\,L_t = -Vdp$). dU and dI are adequately differentials of the internal energy and enthalpy.

The **internal energy depends** on the substance **state** and it changes only when the **extreme states** change.

Heat and work depend not only on the extreme states changes, but also **on the path** on which the states transform, meaning **a process**.

Heat and work are not energy, but their partial changes (constituent): work comes from acting on forces, heat comes from temperature changes and other with a similar effect.

6. ZEROTH LAW OF THERMODYNAMICS

There is a container with adiabatic (stopping any heat exchange with the environment) walls, containing systems A, B and C. Systems A and B are in unbalanced states and therefore they will thermally contact the third system C, but will not contact each other at the same time (Fig. 6.1).

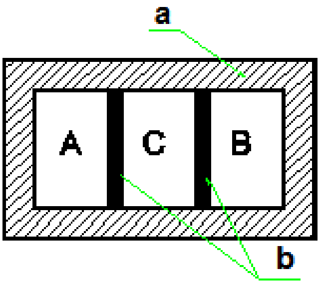


Figure. 6.1. Systems in the adiabatic container: a – adiabatic wall, b – diathermic wall

After some time they will reach a stable state, with all the parameters reaching a certain level and not changing over time.

Systems *A* and *C* are in balance, so are *B* and *C*.

Intuitively, it is safe to assume, that **systems *A* and *B* are in thermal balance** despite the lack of the physical contact.

A postulate describing this relation is called

ZEROTH LAW OF THERMODYNAMICS

(called „zeroth” because it was formulated after the „first law of thermodynamics” got established). There are no experimental facts contradicting it.

The zeroth law of thermodynamics goes:

If two thermodynamic systems are each in thermal equilibrium with a third system, then they are in thermal equilibrium with each other.

7. SPECIFIC HEAT (SPECIFIC HEAT VOLUME)

The definitional formula for the specific heat is based on the formula for the average specific heat $c_{1\text{-}2}$ of the process occurring between the extreme states 1-2

$$c_{1-2} = \frac{q_{1-2}}{\Delta T} \qquad (7.1)$$

$$c = \lim_{\Delta T \to 0} \frac{q_{1-2}}{\Delta T} = \frac{đq}{dT} \qquad (7.2)$$

Basing on the formula (7.2) it is possible to write a general formula for heat transformation:

$$Q_{1-2} = \int_1^2 mcdT \qquad (7.3)$$

The heat transformation can be calculated basing on the formula (7.1)

$$Q_{1-2} = mc_{1-2}(T_2 - T_1). \qquad (7.4)$$

7.1 Ideal gases

Ideal gas obeys the ideal gas law; the specific heat for that gas is constant at constant volume

$c_v = idem$

as well as at constant pressure

$c_p = idem$

From the gas theory in kinetics it results, that increasing the temperature of 1 kilomole by 1K at constant volume requires increase of the kinetic energy of the particles by *MR/2* per each degree of freedom in the advanced and rotary movement. In the case of ideal (and semi-ideal) gas:

$$Mc_p - Mc_\upsilon = MR$$
$$Mc_p / Mc_\upsilon = k$$

where: k – adiabatic exponent.

Numeric illustration of the quantities in question is shown in Table 7.1.

Table 7.1. The degrees of freedom, molar heat and adiabatic exponents for various gases

Gas	The degrees of movement freedom		Mc_υ	Mc_p	$k=Mc_p / Mc_\upsilon$
	advanced movement	rotary movement	kJ/kmol K		
monoatomic	3	0	12,5	20,7	1,667
biatomic	3	2	20,7	29,0	1,400
triatomic or more	3	3	24,9	33,2	1,333

7.2. Semi-ideal gases

The semi-ideal gas obeys the ideal gas law and the specific heat of the isochoric process of such gas is only a temperature function:

$$c_\upsilon = f(T) \qquad \text{or} \qquad c_\upsilon = \varphi(t), \qquad (7.5)$$

Since the semi-ideal gas

$$c_p - c_\upsilon = R,$$

c_p is only the temperature function for this gas.

With this function interrelation given $c_\upsilon = \varphi(t)$ (Fig. *7.1) it is possible to calculate the heat of the isochoric process.*

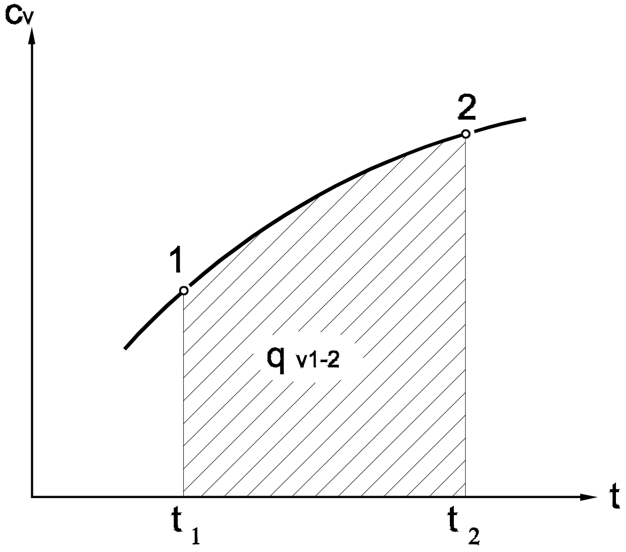


Figure. 7.1. Heat of the isochoric process of the semi-ideal gas.

In practice, calculations apply the concept of the average specific heat of the isochoric process (or isobaric), assuming one of the states as an accepted definite state, the other – free variable state.

In the definite state it is assumed 0°C that the gas temperature equals and in the free state it is ***t***.

The average specific heat of such process is described as:

$\boldsymbol{c_{\upsilon,0-t}}$ or $c_\upsilon\Big|_0^t$

According to the equation (7.1)

$$c_\upsilon\Big|_0^t = \frac{q_{\upsilon,0-t}}{t-0} = \frac{q_{\upsilon,0-t}}{t} \qquad (7.6)$$

Using the relation (7.6) heat, required by the semi-ideal gas for it to increase its temperature form 0°C to ***t***, can be calculated.

$$q_{\upsilon,0-t} = c_\upsilon \Big|_0^t t \qquad (7.7)$$

8. INTERNAL ENENRGY, ENTHALPY

The internal energy of the thermodynamic factor is its state function. It is specifically determined for a given object, given the complete system of the state parameters.

8.1. Ideal and semi-ideal gas

To calculate ideal and semi-ideal gas

$$U=U(p,T,m) \qquad (8.1)$$

per 1 kg of the object

$$u = u(p, T) \qquad (8.2)$$

or (given, that $p\,\upsilon = R\,T$):

$$u = u(T, \upsilon) \qquad (8.3)$$

Differentiating the equation (8.3), gives

$$du = \left(\frac{\partial u}{\partial T}\right)_\upsilon dT + \left(\frac{\partial u}{\partial \upsilon}\right)_T d\upsilon \qquad (8.4)$$

The first law of thermodynamics in a form of differentiation per 1 kg of an object is

$$dq = du + p\,d\upsilon \qquad (8.5)$$

Replacing *du* with the equation (8.4) and adding up all the *du* elements it gives

$$dq = \left(\frac{\partial u}{\partial T}\right)_\upsilon dT + \left[\left(\frac{\partial u}{\partial \upsilon}\right)_T + p\right] d\upsilon \qquad (8.6)$$

Dividing both sides of (8.6) by *dT* and replacing the specific heat ***c*** of the process with *dq/dt*, it gives:

$$c = \left(\frac{\partial u}{\partial T}\right)_\upsilon + \left[\left(\frac{\partial u}{\partial \upsilon}\right)_T + p\right] \frac{d\upsilon}{dT}$$

(8.7)

For the isochoric process **(**υ = idem a $c = c_\upsilon$) the equation (8.7) changes into

$$c_\upsilon = \left(\frac{\partial u}{\partial T}\right)_\upsilon \qquad (8.8)$$

This formula is used to calculate the internal energy of the ideal and semi-ideal gas.

After integrating (8.8) it is:

$$u = \int c_\upsilon dT + f(\upsilon) \qquad (8.9)$$

Basing on the Gay-Lussac and Joule's experiment, conducted on air in conditions making it very similar to an ideal or semi-ideal gas (it obeys the ideal gas law), it is safe to state that internal gas energy does not depend on its volume.

Since ***u*** does not depend on *v*, the function ***f(v)*** in the equation (8.9) will equal the value of the constant integrated u_o:

$$u = \int c_\upsilon dT + u_o \qquad (8.10)$$

The formula (8.10) shows that the internal energy of the ideal and semi-ideal gas is a solely temperature function. Constant u_0 is the absolute value of the internal gas energy at a certain state of reference.

Technical calculations require the usage of the relative internal energy, calculated with the given **T = 0, U = 0.** In this situation the formula for the relative internal energy of the ideal and semi-ideal gas is:

$$u = \int_0^T c_\upsilon dT \qquad (8.11)$$

Since for the ideal gas c_υ= idem,

$$u = c_\upsilon T \qquad (8.12)$$

Using the formula (8.7)

$$c = \left(\frac{\partial u}{\partial T}\right)_\upsilon + \left[\left(\frac{\partial u}{\partial \upsilon}\right)_T + p\right]\frac{d\upsilon}{dT}$$

for the **isobaric** process and combining it with the relation (8.8)

$$c_\upsilon = \left(\frac{\partial u}{\partial T}\right)_\upsilon$$

it gives:

$$c_p = c_\upsilon + \left[\left(\frac{\partial u}{\partial \upsilon}\right)_T + p\ \right]\left(\frac{d\upsilon}{dT}\right)_p \tag{8.13}$$

Since ideal and semi-ideal gas

$$\upsilon = \frac{RT}{p} \tag{8.14}$$

at constant pressure

$$\left(\frac{d\upsilon}{dT}\right)_p = \frac{R}{p} \tag{8.15}$$

At the same time, Gay-Lussac and Joule’s experiment proves, that at constant temperature in a calorimeter, so at constant gas temperature as well:

$$\left(\frac{\partial u}{\partial \upsilon}\right)_T = 0 \tag{8.16}$$

Formulas (8.15) and (8.16) in the equation (8.13) prove that the difference between the specific heat in the isobaric and isochoric process is constant.

$$c_p - c_\upsilon = R \tag{8.17}$$

According to the definitional equation, the enthalpy for every thermodynamic factor is

$$i = u + p\,\upsilon \tag{8.18}$$

For the ideal gas $u = c_\upsilon T$, and $p\upsilon = RT$, as a result

$$i = (c_\upsilon + R)T \tag{8.19}$$

which means that for the ideal gas the formula describing enthalpy is

$$i = c_pT. \tag{8.20}$$

8.2 Liquids and solids

In case of liquids and solids, with small volumetric expansion, the actual heat at the constant volume is practically the same as the actual heat at the constant pressure, which is

$$c_\upsilon \approx c_p = c(t). \qquad (8.21)$$

In correspondence with the equation (7.1)

$$q_{1-2} = c_\upsilon \Big|_{t_1}^{t_2} (t_2 - t_1) \qquad (8.22)$$

Applying the ILT to the object at the volume ***υ* = idem**, it gives

$$q_{1-2} = u_2 - u_1 \qquad (8.23)$$

($l_{1-2} = 0$ because $d\upsilon = 0$)

From (8.22) and (8.23) it results

$$u_2 - u_1 = c_\upsilon \Big|_{t_1}^{t_2} (t_2 - t_1) \qquad (8.24)$$

Taking the relative value of the internal energy in state 1 at the given temperature $\boldsymbol{t_1 = 0°\mathrm{C}}$ as equal to $\boldsymbol{u_1 = 0}$; for any state 2 at the temperature $\boldsymbol{t_2 = t}$ it gives the formula describing the relative internal energy of solids and liquids

$$u = c \Big|_{0}^{t} t \qquad (8.25)$$

Ignoring the specific heat changeability and the temperature, which can be done only in case of less precise calculations, or when the factor temperature changes are not vital, it is allowed to use the approximate equation for the internal energy of the solids and liquids

$$\boldsymbol{u = ct} \qquad (8.26)$$

9. SPECIFIC THERMODYNAMIC PROCESSES

9.1 Isochoric process

The line copying the isochoric process in a complete parameter system is called an isochore (Fig. 9.1).

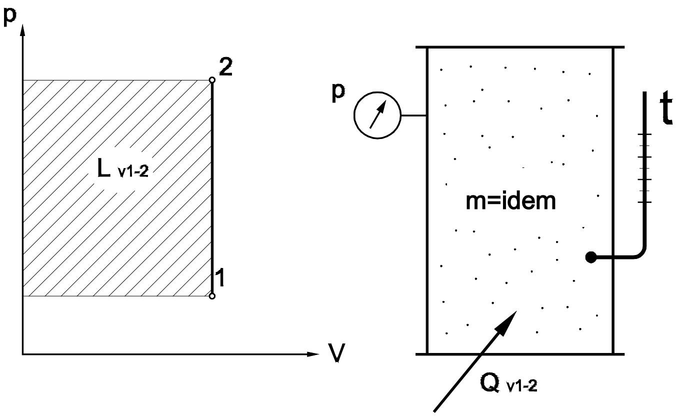


Figure 9.1. Isochoric processes in the system of coordinates p-V

Absolute work of the process (dV=0)

$$L_{\upsilon 1-2} = \int_1^2 p dV = 0 \tag{9.1}$$

Useful work of the process

$$L_{t\upsilon 1-2} = -\int_1^2 V dp = V(p_1 - p_2) \tag{9.2}$$

Heat of the isochoric process: basing on the ILT ($L_{\upsilon 1-2}$=0) is

$$Q_{\upsilon 1-2} = U_2 - U_1 \tag{9.3}$$

and from the formula (7.3)

$$Q_{\upsilon 1-2} = \int_1^2 m c_\upsilon dT \tag{9.4}$$

9.2 Isobaric process

The isobaric process (*p*=idem) can occur in a cylinder with a movable piston (Fig. 9.2). The weight is chosen to give the desired cylinder pressure p. As a result of heat, the gas volume increases from V_1 to V_2, and the temperature from t_1 to t_2. The absolute work is L_{pl-2}, the technical work is L_{tp1-2}, and the process heat is Q_{p1-2},;

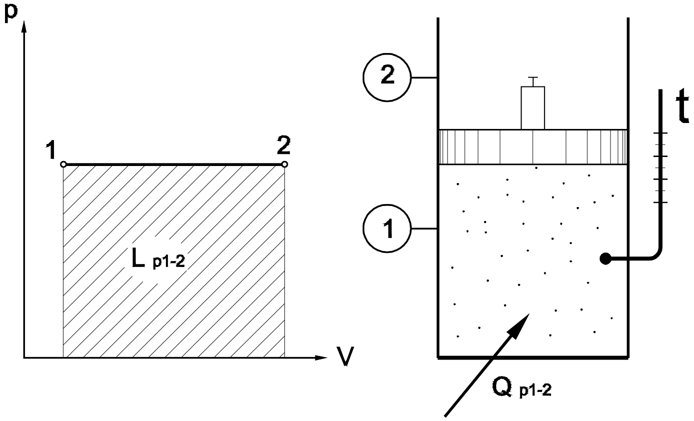


Figure 9.2. Isobaric processes in the system of coordinates p-V

$$L_{p1-2} = \int_1^2 pdV = p(V_2 - V_1) \qquad (9.5)$$

$$L_{tp1-2} = -\int_1^2 Vdp = 0, \qquad (9.6)$$

The heat of the isobaric process, basing on the second form of ILT (L_{tp1-2}=0) is

$$Q_{p1-2} = I_2 - I_1 \qquad (9.7)$$

and from the formula (7.3):

$$Q_{p1-2} = \int_1^2 mc_p dT \qquad (9.8)$$

9.3 Isothermal process

If the gas volume increases, and the focus is to keep it at the same level **(T = idem)**, it has to be adequately heated up.

The **isothermal process** (Fig. 9.3) uses the absolute work $L_{T1\text{-}2}$ and the transferred heat $Q_{T1\text{-}2}$; the technical work $L_{tT1\text{-}2}$ equals.

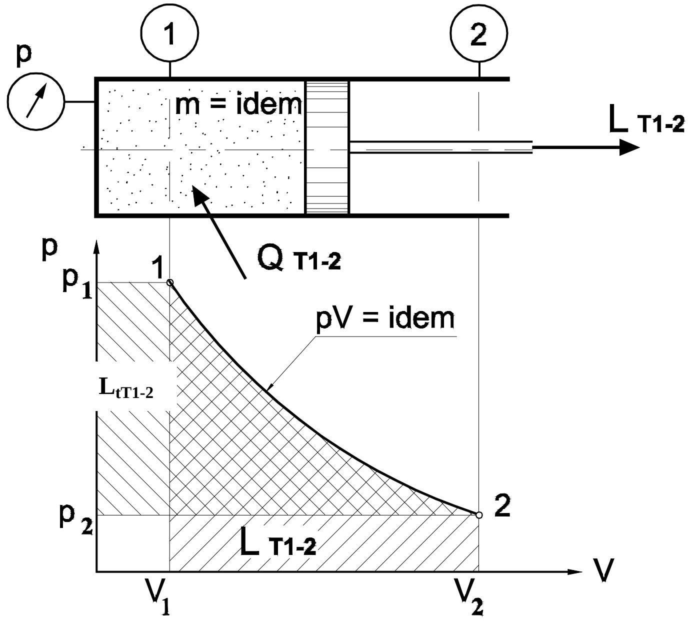


Figure 9.3. Isothermal processes in the system of coordinates p-V

The absolute work $L_{T1\text{-}2}$ of the isothermal process

$$L_{T1-2} = \int_1^2 p dV \qquad (9.9)$$

Solving the integral requires finding the relation $p = p(V)$ given $T =$ idem. For this purpose the factor is assumed to be an ideal or semi-ideal gas, where

$$pV = mRT \qquad (9.10)$$

If T is constant, given that m and R are constant, then

$$pV = \text{idem} \qquad (9.11)$$

According to (9.11) the isotherm p -V is illustrated by an equilateral hyperbole.

After putting ***p* = idem / *V*** into (9.9) it gives

$$L_{T1-2} = idem\int_1^2 \frac{dV}{V} \qquad (9.12)$$

Because **idem = *p₁V₁* =*p₂V₂***, that gives

$$L_{T1-2} = p_1V_1 \ln\frac{V_2}{V_1} \qquad (9.13)$$

or

$$L_{T1-2} = p_1V_1 \ln\frac{p_1}{p_2} \qquad (9.14)$$

Technical work L_{tT1-2} of the isothermal process

Since the result of (9.11) is ***V* = idem/*p*,** therefore

that is $$L_{tT1-2} = -\int_1^2 Vdp = -idem\int_1^2 \frac{dp}{p} \qquad (9.15)$$

$$L_{tT1-2} = -p_1V_1(\ln p_2 - \ln p_1)$$

$$L_{tT1-2} = p_1V_1 \ln\frac{p_1}{p_2} \qquad (9.16)$$

Comparing (9.14) and (9.16), it results in $L_{T1-2} = L_{tT1-2}$ for the isothermal process of the ideal and semi-ideal gas (the absolute work equals technical work of the process).

Heat in the isothermal process

Using ILT formula in the analysed process results in

$$Q_{T1-2} = U_2 - U_1 + L_{T1-2} \qquad (9.17)$$

For ideal and semi-ideal gases $U = U(T)$, which means at constant temperature $U_2 - U_1 = 0$, and the equation (9.17) transforms into

$$Q_{T1-2} = L_{T1-2}$$

$$Q_{T1-2} = p_1V_1 \ln\frac{p_1}{p_2} \qquad (9.18)$$

Using (7.3) to calculate the heat of the isothermal process,

$$Q_{1-2} = \int_1^2 mcdT.$$

using (6.1) on the specific heat,

$$c_{1-2} = \frac{q_{1-2}}{\Delta T}$$

it gives the indeterminate symbol, since the specific heat of the isothermal process ($\Delta T=0$) is infinite $c_T = \infty$.

9.4 Adiabatic process

In the adiabatic reversible process (Fig. 9.4)

$$q_{1\text{-}2}= 0; \qquad dq = 0$$

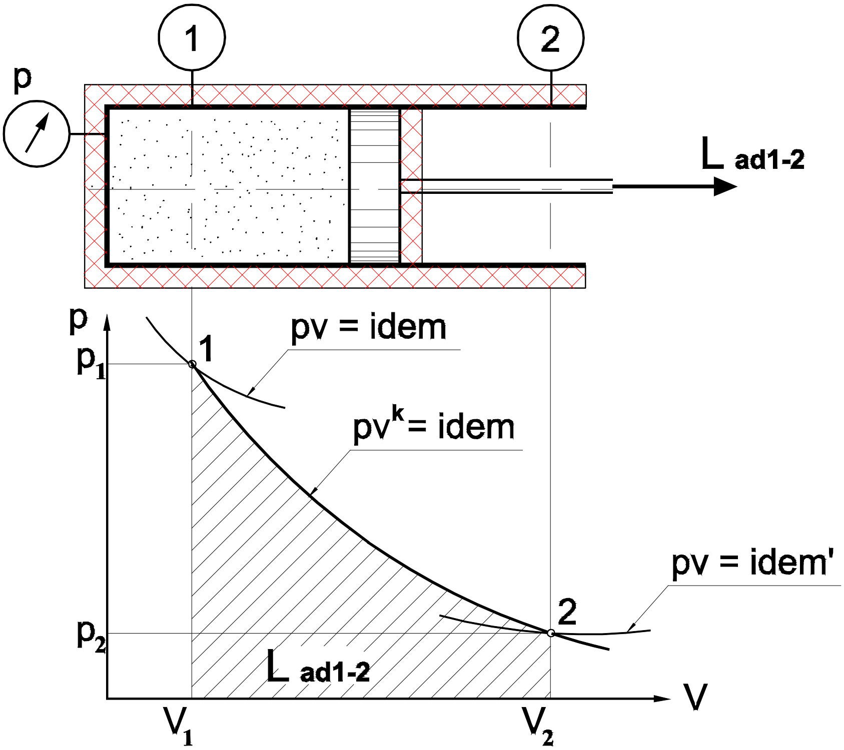


Figure 9.4. Adiabatic processes in the system of coordinates p-V

Applying ILT in the adiabatic process (***dq*=0**) results in

$$0 = du + pdv \qquad (9.19)$$

The assumption is: the factor is an ideal or semi-ideal gas.

Differentiating $p\upsilon = RT$ gives:

$$dT = \frac{1}{R}(pd\upsilon + \upsilon dp) \qquad (9.20)$$

Putting (9.20) into (9.19) and exercising $\boldsymbol{du = c_\upsilon dT}$ (8.8), it results in:

$$0 = c_\upsilon \frac{1}{R}(pd\upsilon + \upsilon dp) + pd\upsilon \qquad (9.21)$$

it transforms into

$$0 = \left(\frac{c_\upsilon}{R} + 1\right)pd\upsilon + \frac{c_\upsilon}{R}\upsilon dp \qquad (9.22)$$

that is

$$0 = (c_\upsilon + R)pd\upsilon + c_\upsilon \upsilon dp \qquad (9.23)$$

The equation (9.23) is bilaterally divided by the product $\boldsymbol{pv}$, given $\boldsymbol{c_\upsilon + R = c_p}$; it gives:

$$0 = \frac{c_p}{c_\upsilon}\frac{d\upsilon}{\upsilon} + \frac{dp}{p} \qquad (9.24)$$

Given the ratio and the integrated equation (9.24), (for ideal gases $\boldsymbol{k}$ = idem) it gives the relation

$$p\upsilon^k = \text{idem} \qquad (9.25)$$

or

$$pV^k = \text{idem}' \qquad (9.26)$$

The equations (9.25) and (9.26) are called equations of the **adiabatic curve of the ideal gas** and the factor ***k*** is called **an adiabatic exponent**.

Verification of integration:
Applying the algorithm on (9.25) it gives

$$\ln p + k \ln \upsilon = idem \qquad (9.27)$$

differentiating leads to the form equal to the equation (9.24)

$$\frac{dp}{p} + k\frac{d\upsilon}{\upsilon} = 0 \qquad (9.28)$$

The absolute work of the adiabatic process

The adiabatic equation leads to

$$p = \frac{idem}{V^k} \qquad (9.29)$$

and after replacing ***p*** from the general work formula with equation (9.29)

$$L_{1-2} = \int pdV \qquad (3.4)$$

it gives

$$L_{ad1-2} = \int_1^2 \frac{idem}{V^k} dV \qquad (9.30)$$

that is

$$L_{ad1-2} = \frac{idem}{1-k}(V_2^{1-k} - V_1^{1-k}) \qquad (9.31)$$

Since

$$idem = p_1 V_1^k = p_2 V_2^k \qquad (9.32)$$

therefore adequately multiplying the phrase from the equation (9.13) by idem it gives the absolute work formula in the adiabatic process of the ideal gas.

$$L_{ad1-2} = \frac{1}{k-1}(p_1 V_1 - p_2 V_2) \qquad (9.33)$$

Using the adiabatic equation and the gas state equation, (9.33) can be reduced to

$$L_{ad1-2} = \frac{mR}{k-1}(T_1 - T_2) \qquad (9.34)$$

or

$$L_{ad1-2} = \frac{p_1 V_1}{k-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{(k-1)/k}\right] \qquad (9.35)$$

Technical work of the adiabatic process:

Technical work of the adiabatic process:

The *k*th root of the adiabate is

$$p^{1/k}V = idem \tag{9.36}$$

that is

$$V = \frac{idem}{p^{1/k}} \tag{9.37}$$

hence

$$L_{t,ad1-2} = -\int_1^2 Vdp = -\int_1^2 \frac{idem}{p^{1/k}}dp. \tag{9.38}$$

After the integration

$$L_{t,\,ad1-2} = \frac{idem}{1-\frac{1}{k}}(p_1^{1-1/k} - p_2^{1-1/k}). \tag{9.39}$$

Multiplying adequately the phrase in parentheses from the right side of the equation (8.39) by

$$idem = p_1^{1/k}V_1 - p_2^{1/k}V_2$$

it gives the formula describing the technical work of the adiabatic process

$$L_{t\,ad1-2} = \frac{k}{k-1}(p_1V_1 - p_2V_2) \tag{9.40}$$

Comparing (9.33) to (9.40) it follows

$$L_{t\,ad\,1\text{-}2} = \boldsymbol{k}L_{ad\,1\text{-}2} \tag{9.41}$$

The specific heat of the adiabatic process**($\boldsymbol{Q_{1\text{-}2}}$=0)** according to (7.1) equals zero

$$c_{ad} = \frac{0}{T_2 - T_1} = 0 \tag{9.42}$$

9.5 Polytropic process

The polytropic process is a process in which the relation between ***p*** and ***V*** is

$$pV^n = idem \tag{9.43}$$

where n = idem (any constant from - ∞ to +∞).

The absolute work in a process

$$L_{n1-2} = \int_1^2 p dV = \int_1^2 \frac{idem}{V^n} dV \tag{9.44}$$

After the integration

$$L_{n1-2} = \frac{idem}{1-n}(V_2^{1-n} - V_1^{1-n}) \tag{9.45}$$

Since

$$idem = p_1 V_1^n = p_2 V_2^n \tag{9.46}$$

multiplying adequately the equation by **idem** it gives – changing the phrase order in the denominator of the first right hand side factor – the relation describing the absolute work of the polytropic process.

$$L_{n1-2} = \frac{1}{n-1}(p_1 V_1 - p_2 V_2). \tag{9.47}$$

The equation (9.47) can be written in other forms useful in solving problems:

$$L_{n1-2} = \frac{mR}{n-1}(T_1 - T_2) \tag{9.48}$$

$$L_{n1-2} = \frac{p_1 V_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{(n-1)/n}\right] \tag{9.49}$$

$$L_{n1-2} = \frac{p_1 V_1}{n-1}\left[1 - \left(\frac{V_1}{V_2}\right)^{(n-1)}\right] \tag{9.50}$$

Technical work of the polytropic process

The *n*th root of the equation (9.43)

$$pV^n = idem$$

it gives

$$V = \frac{idem}{p^{1/n}} \tag{9.51}$$

For the polytropic process

$$L_{tn1-2} = -\int_1^2 \frac{idem}{p^{1/n}}\,dp \tag{9.52}$$

After the integration

$$L_{tn1-2} = \frac{idem}{1-\frac{1}{n}}\left(p_1^{1-1/n} - p_2^{1-1/n}\right) \tag{9.53}$$

given:

$$idem = p_1^{1/n}V_1 = p_2^{1/n}V_2 \tag{9.54}$$

Multiplying adequately by **idem** and transforming it, it gives the formula for the technical work of the polytropic process

$$L_{t\,n1-2} = \frac{n}{n-1}\left(p_1V_1 - p_2V_2\right) \tag{9.55}$$

Comparing (9.55) with (9.47) results in a **relation** between the **absolute and technical polytropic process**:

$$L_{t\,n1-2} = nL_{n1-2} \tag{9.56}$$

Specific heat c_n of the process

According to (7.2), adjusted for the polytropic process

$$c = \lim_{\Delta T \to 0} \frac{q_{1-2}}{\Delta T} = \frac{\text{đ}q}{dT} \tag{7.2}$$

and the first law of thermodynamics

$$c_n dT = du + p\,d\upsilon \tag{9.57}$$

Assuming, the factor contains the ideal and the semi-ideal gas, where ***du = c_υ dT (8.8),*** the result is

$$c_n dT = c_\upsilon dT + p\,d\upsilon \tag{9.58}$$

it can be transformed into

$$0 = (c_\upsilon - c_n)dT + p\,d\upsilon \qquad (9.59)$$

Differentiating the Clapeyron's relation

$$pd\,\upsilon + \upsilon\,dp = R\,dT \qquad (9.60)$$

using this equation in (9.59) it leads to

$$0 = (c_\upsilon - c_n)(pd\upsilon + \upsilon dp)\frac{1}{R} + pd\upsilon \qquad (9.61)$$

after transformation it is

$$0 = \frac{c_\upsilon - c_n}{R}\upsilon dp + \left(\frac{c_\upsilon - c_n}{R} + 1\right)pd\upsilon \qquad (9.62)$$

The equation (9.62) can be transformed using the relation

$$c_\upsilon + R = c_p \qquad (9.63)$$

into

$$0 = \upsilon dp + \frac{c_p - c_n}{c_\upsilon - c_n}pd\upsilon \qquad (9.64)$$

Dividing (9.64) by the product ***pv*** gives the differential relation between ***p*** and ***v***

$$\frac{c_p - c_n}{c_\upsilon - c_n}\frac{d\upsilon}{\upsilon} + \frac{dp}{p} = 0 \qquad (9.65)$$

integrated, given $(c_p — c_n)/(c_\upsilon — c_n)$ is not the ***V*** nor ***p*** function, is

$$p\upsilon^{(c_p - c_n)/(c_\upsilon - c_n)} = idem \qquad (9.66)$$

Since in the polytropic process

$$p\upsilon^n = idem \qquad (9.67)$$

therefore

$$\frac{c_p - c_n}{c_\upsilon - c_n} = n \qquad (9.68)$$

this, with some transformations, leads to the formula for the specific heat of the polytropic process

$$c_n = \frac{c_p - nc_\upsilon}{1-n}, \tag{9.69}$$

it can be reduced to

$$c_n = c_\upsilon \frac{k-n}{1-n}, \tag{9.70}$$

The equation (9.70) proves, that the specific heat of the polytropic process of the ideal gas, where c_υ and k are constant, is also constant. For the semi-ideal gas, the specific heat of the polytropic process is a function focusing only on the temperature.

Heat Q_{n1-2} of the process

Using ILT formula in the polytropic process results in

$$Q_{n1-2} = U_2 - U_1 + L_{n1-2} \tag{9.71}$$

or

$$Q_{n1-2} = I_2 - I_1 + L_{tn1-2} \tag{9.72}$$

or using the formula for the specific heat of the process

$$Q_{n1-2} = \int_1^2 mc_n dT \tag{9.73}$$

Comparison of the chosen polytropic processes

Assuming adequate values of the polytropic index n in the equation (9.43) as wells as in (9.70), describing the specific heat of the polytropic process, it gives the record of the characteristic processes (presented in Fig. 9.5), where:

$n = O$, p = idem - the isobaric process at the specific heat $c_n = c_p$

$n=1$, pV=idem- the isothermal process of the ideal and semi-ideal gas at the specific heat $c_n=\infty$,

$n = k$, pV^k = idem – the adiabatic process of the ideal process at the specific heat $c_n= 0$

$n = \pm \infty$, V = idem – the isochoric process at the specific heat $c_n=c_\upsilon$.

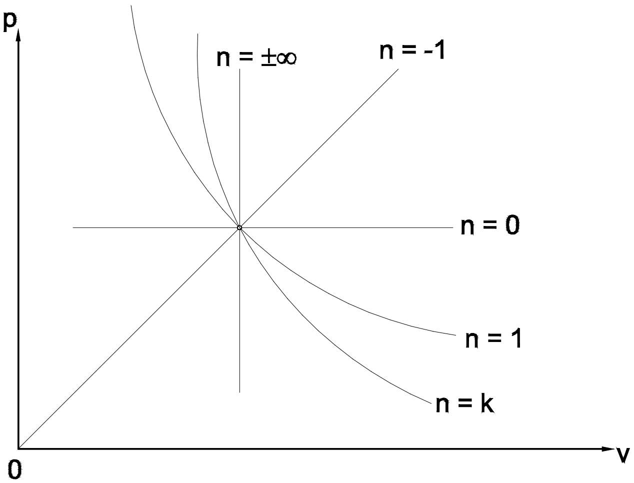


Figure. 9.5. Polytropic processes in the system of coordinates p-V

The chosen relations between the parameters describing the extreme states 1-2 in the polytropic processes

Polytropic equation can be expressed as

$$pV^n = \text{idem} \tag{9.74}$$

or, with the volume ousted, as the Clapeyron's relation

$$Tp^{(1-n)/n} = \text{idem} \tag{9.75}$$

or as

$$TV^{n-1} = \text{idem} \tag{9.76}$$

given the pressure in (9.74) is shown as *mRT/V.*

Using the mentioned forms of polytropic equations of the extreme state 1 and 2 of the process, it leads to the following relation:

$$\frac{T_1}{T_2} = \left(\frac{p_2}{p_1}\right)^{(1-n)/n} \tag{9.77}$$

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{(n-1)} \tag{9.78}$$

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{n/(n-1)} \qquad (9.79)$$

$$\frac{V_2}{V_1} = \left(\frac{T_1}{T_2}\right)^{n/(n-1)} \qquad (9.80)$$

10. ENTHROPY

Work in any process (the absolute as well as the technical) can be presented with the state parameter system ***p-v*** (Fig. 10.1) as

$\int \boldsymbol{\text{đ}}l = \int p\,dv$

or

$\int \boldsymbol{\text{đ}}l = \int v\,dp$

(corresponding fields under the transition line 1-2).

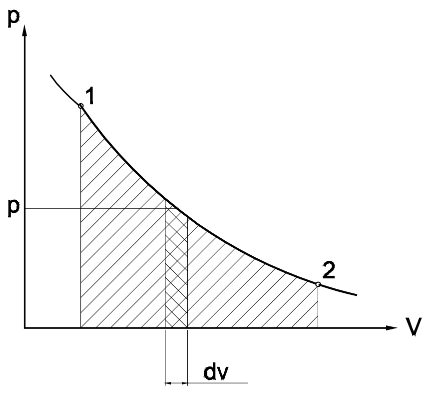


Figure. 10.1. The absolute work mapping

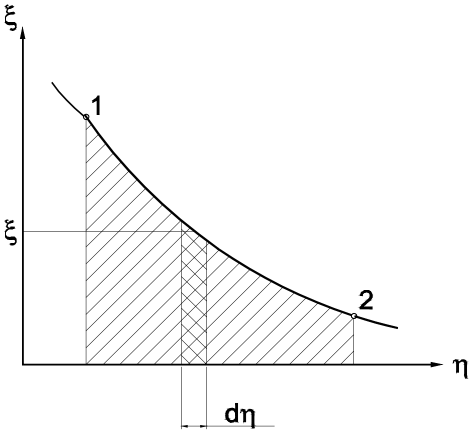


Figure. 10.2. Mapping of the heat of transition

Is it possible to map the **HEAT** of transition in a similar way?

It is assumed, that the marks ***ξ*** i ***η*** (Fig. 10.2) form a frame of reference, where the field underneath the process line in this frame is the **heat of transition**. The differential formula ***ξdη*** will be applied to calculate the heat of transition - ***đq***:

$$đq = \xi d\eta \qquad (10.0)$$

The differential of ILT is

$$đq = du + pd\upsilon \qquad (10.1)$$

according to the assumed requirements,

$$\xi d\eta = du + pd\upsilon \qquad (10.2)$$

In case of the ideal and semi-ideal gas $du = c_\upsilon dT$, while pressure ***p*** is

$$p = \frac{RT}{\upsilon} \qquad (10.3)$$

therefore for the ideal and semi-ideal gas

$$\xi d\eta = c_\upsilon dT + R\frac{T}{\upsilon}d\upsilon \qquad (10.4)$$

In the case of ideal gas, where c_υ is not dependant on ***T*** and is constant, the equation (10.4) is a first-order diferential.

This formula can be reduced to a diferential using so-called integrating factor, which in this case is the phrase **l/T**. Multiplying the equation (10.4) bilaterally by **l/T** it results in

$$\frac{\xi}{T}d\eta = c_\upsilon\frac{dT}{T} + R\frac{d\upsilon}{\upsilon} \qquad (10.5)$$

The right side of the equation (10.5) is the function differential

$$c_\upsilon \ln T + R \ln\upsilon$$

This means, that the ***ξ*** parameter is the object's absolute temeperature ***T***

$$\xi = T \qquad (10.6)$$

therefore

$$d\eta = c_\upsilon\frac{dT}{T} + R\frac{d\upsilon}{\upsilon} \qquad (10.7)$$

which after the integration is

$$\eta = c_\upsilon \ln T + R\ln\upsilon + const \qquad (10.8)$$

The ***η*** function in thermodynamics is marked ***s*** and called the **specific enthropy**. Particularly the formula for the ideal gas enthropy is

$$s = c_\upsilon \ln T + R \ln \upsilon + const \qquad (10.9)$$

The equation (10.9) states, that ***s*** is specified for the set value of ***T*** and ***v***, which means that enthropy is the function of the state of an object; thus enthropy is a state mark of an object.

To calculate the absolute numeric value of enthropy with the equation (10.9), it is vital to know the constant value of the integral.

Technology uses the relevant value of enthropy.

It is assumed, that with the agreed state parameters p_0, T_0, υ_0 enthropy s_0 equals zero.

Diferentiating the equation (10.7), starting from the accepted state, in which enthropy equals zero, up to any state with ***p, T, v,*** parameters, in which enthropy equals ***s***, it gives

$$s = c_\upsilon \ln \frac{T}{T_0} + R \ln \frac{\upsilon}{\upsilon_0} \qquad (10.10)$$

The equation (10.10) calculates the numeric value of enthropy, provided that the state of the ideal gas is defined with ***T, v***. This equation can be reduced to one, with the object's state defined by ***p, T***, which is: basing on the Clapeyron's relation with the parameters ***p, v, T*** and accepted ***p_0, υ_0, T_0***, forming the following equation

$$\frac{p\upsilon}{p_0\upsilon_0} = \frac{T}{T_0} \qquad (10.11)$$

after applying an algorythm and then tidying it up

$$\ln \frac{\upsilon}{\upsilon_0} = \ln \frac{T}{T_0} - \ln \frac{p}{p_0} \qquad (10.12)$$

Putting (10.12) into (10.10) and transforming it, it gives

$$s = (c_\upsilon + R) \ln \frac{T}{T_0} - R \ln \frac{p}{p_0} \qquad (10.13)$$

Since $c_\upsilon + R = c_p$, the final enthropy formula, with the ideal gas state defined by parameters ***p, T,*** is

$$s = c_p \ln \frac{T}{T_0} - R \ln \frac{p}{p_0} \qquad (10.14)$$

Given that the gas state is defined by two parameters p and v, applying the Clapeyron's relation (equation 10.11) and an algorithm, it gives

(10.15)

and after inserting it into (9.10)

$$s = c_{\upsilon}\ln\frac{p}{p_0} + c_{\upsilon}\ln\frac{\upsilon}{\upsilon_0} + R\ln\frac{\upsilon}{\upsilon_0} \qquad (10.16)$$

and tidying it up, it gives a formula for the **enthropy** of the ideal gas, where the state id defined by ***p, v***:

$$s = c_{\upsilon}\ln\frac{p}{p_0} + c_p\ln\frac{\upsilon}{\upsilon_0} \qquad (10.17)$$

Thanks to the enthropy properties, the heat of a random process can be expressed with the equation

$$Q_{1-2} = \int_1^2 TdS \qquad (10.18)$$

Since ***S = ms***, the specific enthropy ***s*** can be applied to calculate the process heat.

$$Q_{1-2} = \int_1^2 mTds \qquad (10.19)$$

11. MAPPING THE PROCESSES CHARACTERISTIC TO THE T-s SYSTEM

Isochoric process

Describing the flow of the isochoric process in the T-s system means finding the interrelation between T and s, where υ = idem (Fig. 11.1). The equation (10.10) can be used for this purpose:

$$s = c_{\upsilon}\ln\frac{T}{T_0} + R\ln\frac{\upsilon}{\upsilon_0}$$

which, after the transformation, is:

$$s = c_{\upsilon}(\ln T - \ln T_0) + R(\ln\upsilon - \ln\upsilon_0)$$

With v = **idem** the equation is

$$s = c_v \ln T + idem \qquad (11.1)$$

this means, that in the *T*-*s* system the isochore for the ideal gas is an exponential function.

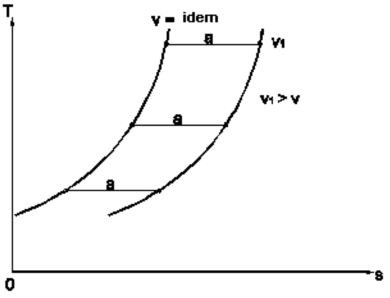


Figure. 11.1. Isochoric process in the T-s system

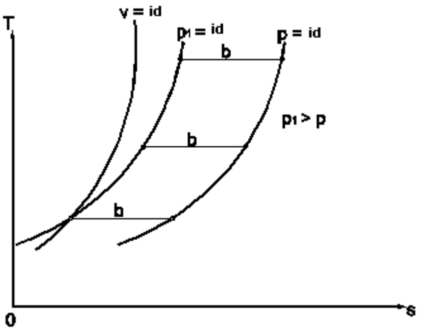


Figure. 11.2. Isobaric process in the T-s system

With the higher value of $v_1 > v$ in the equation (11.1), the value of the constant idem **idem' > idem** will increase to the value, which means the parallel movement of the isochore v_1 = **idem** in relation to v = **idem**.

Isobaric process

To map the flow of the isobaric process of the ideal gas, the equation (10.14) can be applied,

$$s = c_p \ln \frac{T}{T_0} - R \ln \frac{p}{p_0}$$

if p is constant, it is as follows

$$s = c_p \ln T + idem \qquad (11.2)$$

As a result, the isobar is an exponential curve (Fig. 11.2), but at a different angle. While mapping the isobar at the pressure $p_1 > p$, it is sure that idem will decrease by b (Fig. 11.2). The isobar, the exponential function p_1, will automatically move by the length of b.

Isothermal process

Isothermal process

Isothermal process in the T-s system (Fig. 11.3) is a straight line with the formula T = idem.

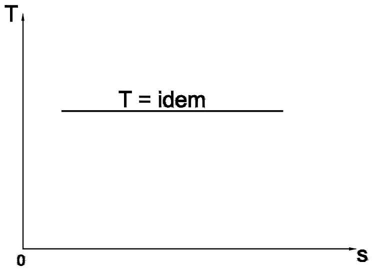


Figure. 11.3. Isothermal process in the *T*-*s* system

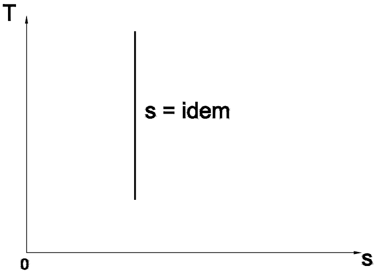


Figure. 11.4. Adiabatic process in the *T*-*s* system

Adiabatic process

The process heat $\boldsymbol{q_{1-2}}$ consists of two parts: heat supplied from the outside source $\boldsymbol{q_{z1-2}}$, and the friction heat $\boldsymbol{q_{f1-2}}$. The adiabatic process (Fig. 11.4) takes place only when $\boldsymbol{q_{z1-2}=0}$. The heat of the adiabatic process $\boldsymbol{q_{1-2}= 0}$ equals zero only when there is no friction - $\boldsymbol{q_{f1-2} = 0}$. Looking into the adiabatic process without any friction, it is known, that

$$Q_{ad1-2} = \int_1^2 TdS \qquad (11.3)$$

that is

$$TdS = 0 \qquad (11.4)$$

Given $T > 0$,

$$dS = 0 \qquad (11.5)$$

hence

$$S = idem \qquad (11.6)$$

The adiabatic process without friction is a process with constant enthropy, called an **isentropic** process.

Polytropic process

The flow of the polytropic process (Fig. 11.5) in the *T*-*s* system can be mapped using the flows of some characteristic processes.

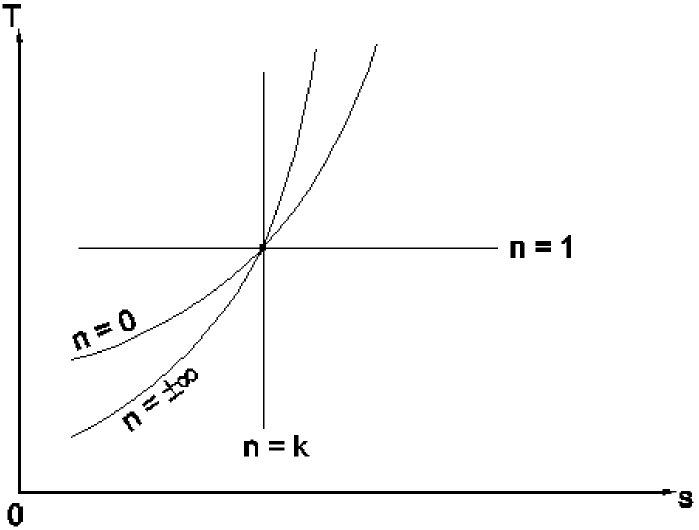


Figure. 11.5. The flow of the polytropic process in the T-s system

12. CYCLES

A process, in which the end state equals the beginning state, is called a cycle.

The graphic illustration of the circulation in the state parameter system is a closed curve. A cycle process can be modelled by so-called cycle engines or cycle devices.

There is **a power cycle (engines)** - a diagram shows a clockwise rotatory direction, and **a heat pump cycle (refrigerators)** – a diagram shows an anti-clockwise rotatory direction.

Power cycle engines and devices

Power cycle engines and devices

A power cycle engine (Fig. 12.1) has one characteristic feature - no valves. An ideal power cycle engine is tight and there is no friction. Any thermodynamic factor can be a circular factor.

When the piston is in the TDC, the gas state is defined by point A in the p-V diagram.

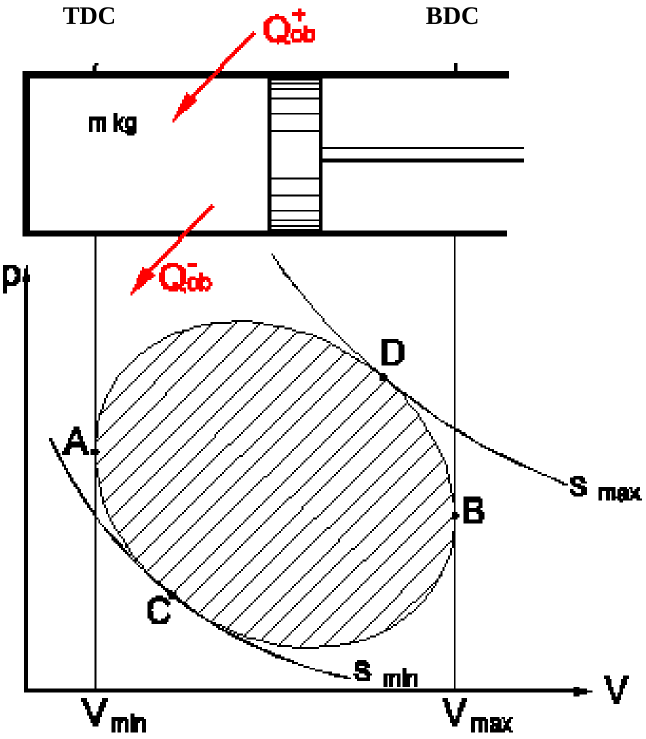


Figure. 12.1. The model of a cycle and a cycle engine in the p-V diagram

The gas, during the expansion from state A to D, has to be heated and it receives the heat $Q_{A\text{-}D}$. Further expansion from the state D all the way to piston BDC marked with the point B requires cooling and the heat equals $Q_{D\text{-}B}$.

After the crankshaft passes BDC, the gas enclosed inside the cylinder starts to compress. Compress from state B to C requires cooling. The heat of cooling is $Q_{B\text{-}C}$.

The final compression phase from state C to A requires heating, where the supplied heat equals $Q_{C\text{-}A}$.

The total supplied heat $\boldsymbol{Q_{C\text{-}A} + Q_{A\text{-}D} = Q^{+}_{ob}}$ is the positive part of the cycle heat. The cooling heat from state D to C, though state B, equals $\boldsymbol{Q_{D\text{-}B} + Q_{B\text{-}C}}$ and is the negative part of the cycle heat. $\boldsymbol{Q^{-}_{ob}}$.

The absolute cycle heat is $\boldsymbol{Q_{ob}}$.

$Q_{ob} = Q^{+}_{ob}\ \mathbf{Q}^{-}_{\mathbf{ob}}.$

The cycle and all the cycle heat types can be presented by a T-S diagram (Fig. 12.2 and 12.3).

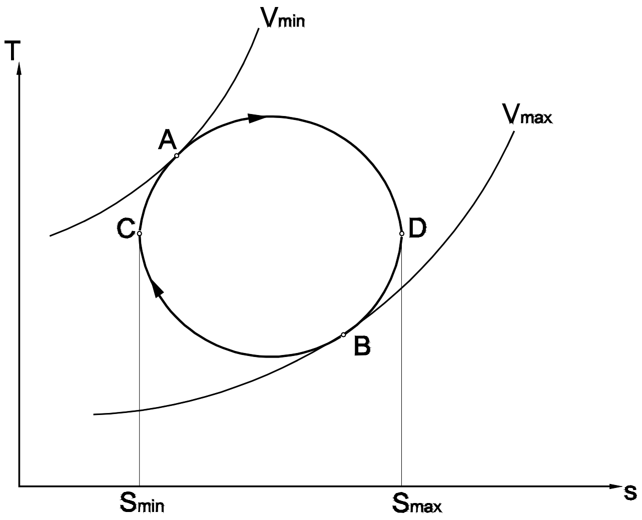


Figure. 12.2. A cycle mapped in the T-S diagram.

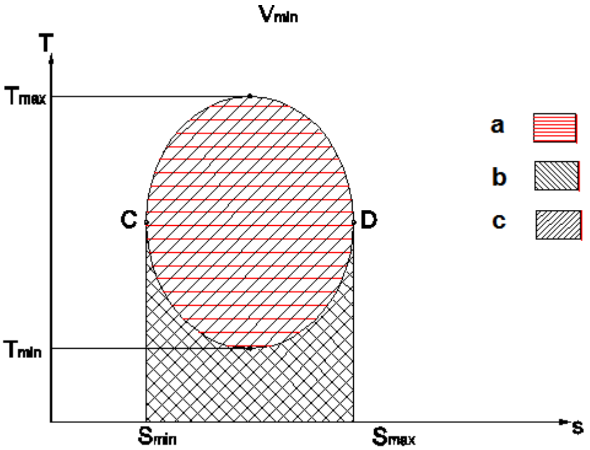


Figure. 12.3. Cycle heat in the T-S diagram. a – cycle heat Q_{ob}, b - negative heat Q^{-}_{ob} (cooling), c – positive heat Q^{+}_{ob} (heating)

The cycle falls between V_{max} and V_{min}, S_{max} and S_{min}, as well as T_{max} and T_{min}.

Work done during the 1 cycle of the engine is L_{ob}.

The efficiency of the power cycles (engines)

The work done by a power cycle is positive, that means a device with such cycle produces work. The work done during expansion L_{ex} in this cycle is bigger than the work done during compression L_k(Fig. 12.4). In that cycle the heating heat is also greater than the cooling heat, which means that the cycle heat is also positive.

Between the cycle heat and cycle work there is a relation, which can be found by applying ILT to the cycle formula.

$$Q_{1-2} = U_2 - U_1 + L_{1-2} \qquad (12.1)$$

Since in the cycle $Q_{1-2} = Q_{ob}$ and $L_{1-2} = L_{ob}$ state 1 equals state 2, $U_2 = U_1$, that means

$$Q_{ob} = L_{ob} \qquad (12.2)$$

The cycle heat equals the cycle work

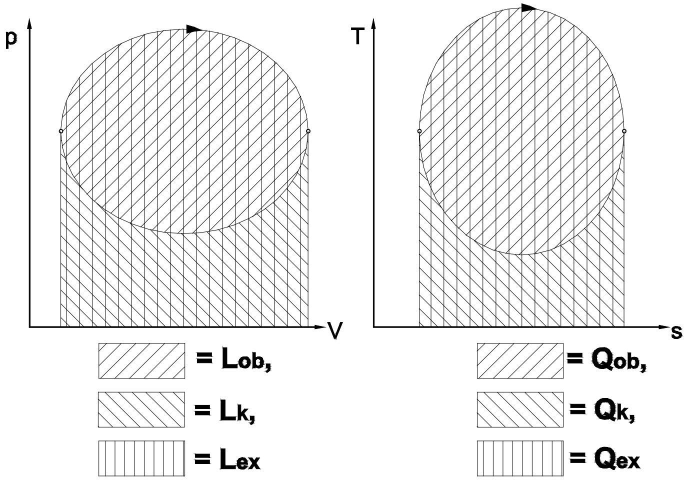


Figure 12.4. Engine cycle in the p-V and T-S diagram

To create heat during a power cycle, fuel or other forms of energy are used.

The concept of EFFICIENCY, in general sense, is the ratio of effects (results) to input (costs).

In case of a power cycle engine the effect is work and the input – heat, which means, the cycle efficiency is the ratio of the cycle work to the positive part of heat in the cycle:

$$\eta_{ob} = \frac{L_{ob}}{Q_{ob}^{+}} = \frac{Q_{ob}^{+} - Q_{ob}^{-}}{Q_{ob}^{+}} = 1 - \frac{Q_{ob}^{-}}{Q_{ob}^{+}} \qquad (12.3)$$

The bigger is the cycle efficiency, the lower the cooling heat is and the higher the heating heat.

Carnot cycle

The Carnot Cycle is a system with two isotherms and two isentropes (Fig. 12.5).

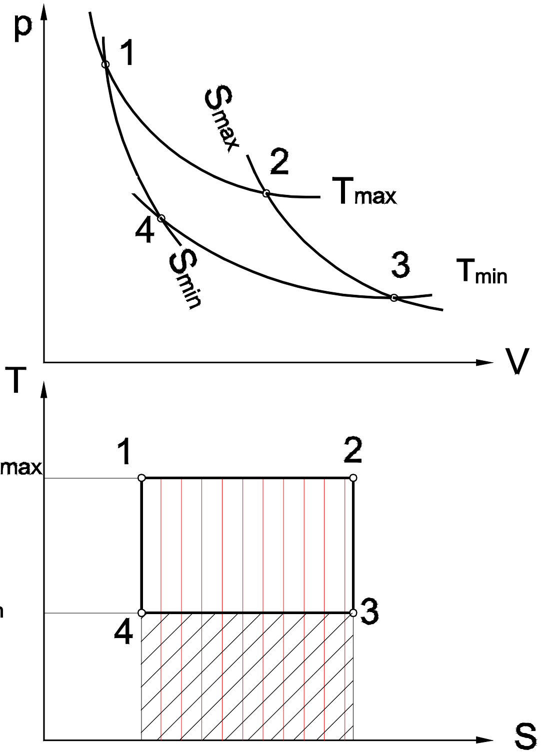


Figure 12.5. Carnot cycle in the p-V and T-S diagram.

Heating heat in the Carnot cycle

Q_c^{+} $Q_c^{+} = T_{max}\,(S_2 - S_1)$

Cooling heat

Q_C^- $Q_C^- = T_{min}\,(S_3 - S_4)$

The efficiency of the Carnot power cycle

$$\eta_c = 1 - \frac{Q_C^-}{Q_c^+} = 1 - \frac{T_{\min}(S_3 - S_4)}{T_{\max}(S_2 - S_1)} \qquad (12.4)$$

since $S_2 - S_1 = S_3 - S_4$, as a result

$$\eta_c = 1 - \frac{T_{\min}}{T_{\max}} \qquad (12.5)$$

Any free cycle has a corresponding Carnot cycle that can be assigned to that free cycle (Fig. 12.6).

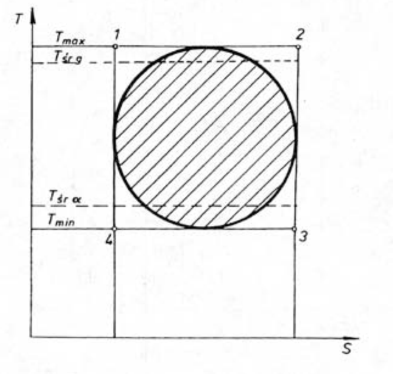


Figure 12.6. A free cycle (circular as an example) and a corresponding cycle (square 1-2-3-4).

This is the cycle 1-2-3-4, occurring at the temperatures T_{max} and T_{min}.

Given:

$T_{śr.g}$ – temperature for the average height of the heating heat field in the cycle,

$T_{śr.d}$ – temperature for the average height of the cooling heat field in the cycle,

it can be calculated (compare Fig. 12.7)

$$T_{śr.g}(S_2 - S_1) = Q_{ob}^+ \qquad (12.6)$$

$$T_{śr.d}(S_3 - S_4) = Q_{ob}^- \qquad (12.7)$$

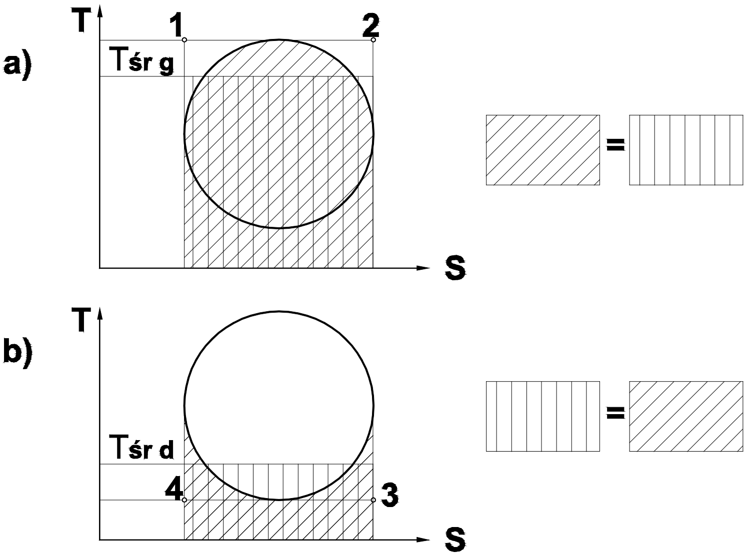


Figure 12.7. Calculating the average cycle temperatures

The **free cycle** efficiency can take a form of

$$\eta_{ob} = \frac{Q_{ob}^{+} - Q_{ob}^{-}}{Q_{ob}^{+}} = 1 - \frac{T_{śr.d}}{T_{śr.g}} \qquad (12.8)$$

The following inequality is true:

$$\eta_{ob} = 1 - \frac{T_{śr.d}}{T_{śr.g}} \langle 1 - \frac{T_{\min}}{T_{\max}} = \eta_c \qquad (12.9)$$

that means, the most efficient is the Carnot cycle corresponding with the specific free cycle (at T_{max} and T_{min} given for that free cycle).

13. SECOND LAW OF THERMODYNAMICS (II LT)

The second law of thermodynamics has more than one definition. They are logically similar but put in different words. According to the laws of logics, theorems are equal only if denying one of them causes fallacy of the other one.

According to **Ostwald,** the second law of thermodynamics is as follows:

It is not possible to build a perpetuum mobile of second type.

Perpetuum mobile of the second type (**PM II**) is an engine or a power cycle device with the **efficiency cycle equal unity**.

***Perpetuum** mobile of the first type (**PM I**) – a machine that can work “**for free**”* [2].

The formula **(12.3)**

$$\eta_{ob}=\frac{L_{ob}}{Q^{+}_{ob}}=\frac{Q^{+}_{ob}-Q^{-}_{ob}}{Q^{+}_{ob}}=1-\frac{Q^{-}_{ob}}{Q^{+}_{ob}}$$

defining the cycle efficiency, indicates, that the efficiency reaches unity (i.e. 100%), only when the cooling heat of the cycle equals zero.

The PM II can be presented on the T-S diagram as an engine working with only the **top heat source** (Fig. 13.1) – with no cooling involved.

For comparison only, Figure 13.2 presents the work of a real engine, requiring a cooling system (for the bottom heat source).

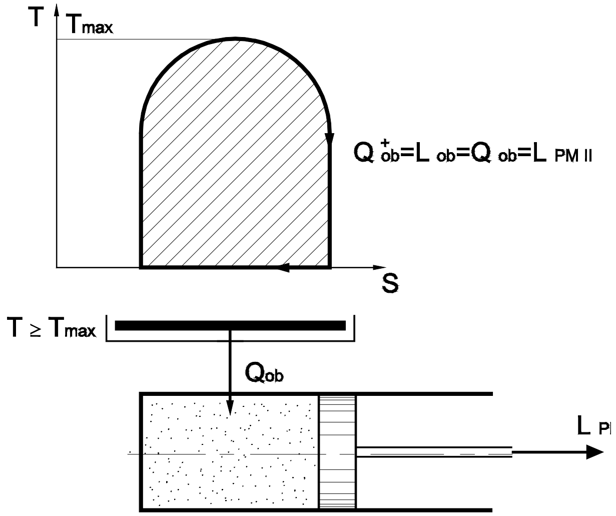


Figure. 13.1. Perpetuum mobile of the II type

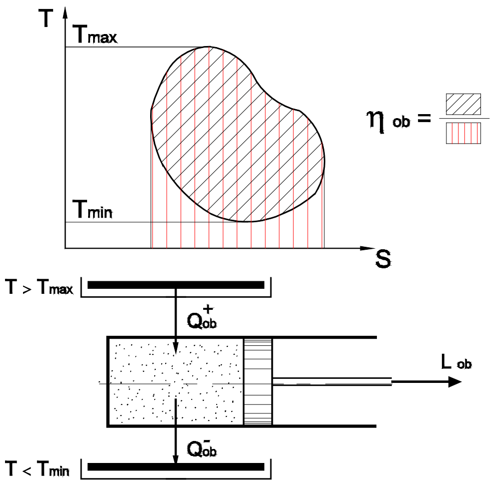


*Figure. 13.2. Diagram for the work of a real engi*ne

Clausius had a different definition of the II LT:

Heat does not travel spontaneously (e.g. through a direct contact) **from an object of lower temperature onto an object of higher temperature.**

Transferring heat from a lower temperature object onto the higher temperature object requires compensation and it **can happen in a heat-pump cycle.**

Transferring heat from a cooler to a hotter object causes the decrease in the total enthropy of the two objects. This can not happen through a direct contact. If a phenomena of this kind occurs that means a third or even more objects are involved. The increase in the enthropy of the agent objects compensates the decrease of the enthropy of the two objects in question. That is called "compensation".

Given an extreme situation, where heat is transferred between two objects without the temperature difference, **building** II P M **would be possible only if** it was possible to cool the circular factor with a body at temperature 0K.

It results from the third law of thermodynamics (III LT), that it is impossible to reach temperature of 0 kelvins.

Enthropy has and additive character and an **enthropy of the system of objects** S_{syst} is a sum

$$\sum_{i=1}^{n} S_i$$

entropies of all the *n* **elements of the systems taking part** in the phenomenon.

A mathematical definition of **II LT is so-called the enthropy increase rule**, defined by Clausius as:

In a real system of objects only those phenomena are possible which come with the increase in those object's enthropy.

Calling the **increase in the object's entropy** π**,** it leads to:

$$\pi = S_{syst2} - S_{syst1} = \sum_{i=1}^{i=n} \Delta S_i \rangle 0. \qquad (13.1)$$

Reversible and irreversible processes

A process is reversible if, using a differential environment change, it is possible to create a process reverse to the current one, i.e. following the same path but in the opposite direction.

In reality **all the processes are irreversible**, but conducting experiments carefully enough, it is possible to get close to the reversible processes.

A strictly **reversible** process is a simple and quite useful theoretical concept; its relation to the real processes is the same as the abstract ideal gases to the real ones.

In an adiabatic process, heat **does not enter nor exit** the system. An **adiabatic** process can be either a **reversible** or **irreversible** process – the definition does not exclude any of them.

During the adiabatic compression of a factor inside the cylinder, the gas temperature increases, since if the process temperature $Q = 0$, according to I LT, work L done by moving piston changes into internal energy increase ΔU in the system.

Different speeds of the piston movement give different work values L. Only in case of the reversible processes (where p has a precise value) work is as follows:

$$\int pdV,$$

i.e. it equals the area under the adequate curve in the **p-V** diagram.

Accordingly, ΔU and the corresponding temperature change ΔT will be different for the reversible and irreversible processes.

There are a number of other processes that can take place and bring the system back to its original state of balance; this is called a **circular process**, or a **cycle**.

If all the following processes of the cycle are reversible, that means it is a **reversible cycle**.

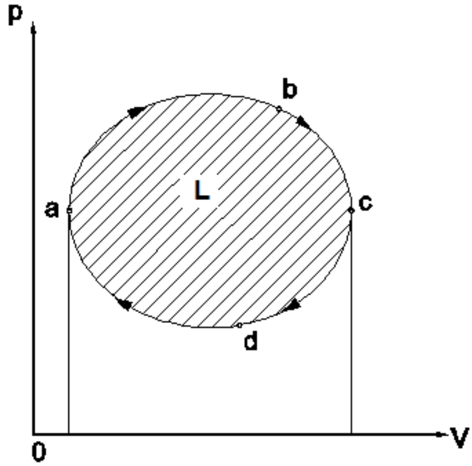


Figure. 13.3. Reversible cycle

Figure 13.3. presents a reversible cycle on a ***p-V*** diagram. The ***abc*** curve represents the system decompression and the area under the curve is the work done during that decompression.

Following the ***cda*** curve the system returns to its original state. In this case the system is being compressed and the area under the curve is the work done during the compression. The **resultant** of the work done by the system equals the area within the curve. That work is **positive**.

If the cycle followed the reverse path, i.e. the gas was decompressed along the ***adc*** curve and compressed along the ***cba*** curve, the resultant of that work would ***equal***, with a ***minus***, the work from the previous example.

The increase in enthropy rule

After conducting a number of observations and experiments, it was established as follows:

1. *In all the* ***irreversible processes, the sum of enthropy*** *of all the participating objects* ***increases****. (The sum of the enthropy increases of all the participating objects is greater than zero)*[1]*)*
2. *In all the* ***reversible processes, the sum of entropy*** *of all the participating objects* ***stays constant****. (The sum of the enthropy increases of all the participating objects equals zero).*
3. *In all the* ***reversible processes, the sum of entropy*** *of all the participating objects* ***stays constant****. (The sum of the enthropy increases of all the participating objects equals zero).*

Assuming that the ***sum of all the enthropy increases*** *of the participating objects is* π*, the entropy increase rule is as follows:*

$\pi>0$ *irreversible processes,*

$\pi = 0$ *reversible processes,*

$\pi< 0$ *impossible processes.*

---

[1]) *it does not mean that enthropy of a single participating object cannot decrease (entropy increase of a single object might be negative).*

[2]) *considering the enthropy increase rule, the system of participating objects has to be separated from the rest of the objects (not included in the experiment).*

II. SELECTED ISSUES

14. INTERNAL COMBUSTION ENGINE CIRCULATION (THEORETICAL AND COMPARATIVE)

Thermodynamic processes taking place inside the internal combustion engines have not been fully studied nor explained yet. High temperature and pressure, variable physicochemical state of an object, a very short time per a particular process, they all comprise major disruption in the occurring phenomena assessment. This type of evaluation can be carried out by comparing the real circulation (occurring inside internal combustion engines) with the theoretical ones - a result of theoretical deliberations, which, after multiple assumptions and simplifications, produced an easy to analyse circulation form, representing a cyclic process of transforming heat energy into mechanical one.

Depending on the number of simplifying assumptions applied, there are **theoretical work circulations** and **comparative circulations** with a real factor. To analyse those, it is common to use the thermodynamic factor equations and the basic equations of the characteristic thermodynamic processes. The real circulation, occurring in an internal combustion engine, can be completed in a form of a diagram only through the research using an indicator. Such a circulation is called

an indicator diagram.

A theoretical circulation and a comparative one with the real factor are also called "a theoretical indicator diagram" and "a comparative indicator diagram with the real factor" – but those terms are less correct. If the theoretical circulations, the theoretical circulations with the real factor and the real circulations are presented in a ***p-V*** diagram, they stand for work. That work, proportionate to the area of the enclosed cyclic circulation, is called adequately: theoretical work, comparative diagram work with the real factor, and the indicated work.

The indicated work is the real circulation work - the internal work of an engine. With the internal work defined as work done by an ideal machine, with no friction resistance. On occasions, the term "indicated work" suggests work calculated basing the comparative diagram with the real factor, but in that case the term replaces "comparative circulation work".

Theoretical internal combustion engine circulation

The theoretical circulation corresponds with the thermodynamic process, employing a constant volume of gas, Heat goes in and out of external heat sources (called upper and lower) through the walls enclosing gas inside the cylinder. Heat can enter the system in an isochoric, isobaric or isochoric and isobaric way.

During the heat transfer the cylinder walls do not put up any heat resistance (diathermic walls). During compression and decompression, on the other hand, they act as perfect insulation. As a result all the processes, connected to the volume changes, are adiabatic.

There is no friction between the cylinder sleeve and the bush.

Considering the ideal gas a factor means assuming that the specific heat remains constant at constant pressure as well as at constant volume. Accordingly, the internal gas energy is a linear function of temperature.

Assuming, that this process is extremely slow - the gas speed is close to zero – it results in a reversible thermodynamic process.

Theoretical circulation, the most precise match for work an internal combustion engine, is a mixed circulation with an isochoric and isobaric heat addition. It is called the **Sabathe cycle**.

Calculating the parameters of a theoretical circulation

The gas quantity in the circulation is free. It can be assumed that it is in ***m*** kg or, expressing mass units in kilo moles, ***n*** kmol.

The initial gas parameters in point 1 are p_1, V_1, T_1 (Fig. 14.1).

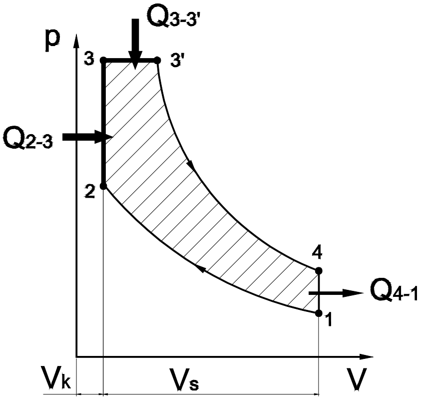


Figure. 14.1. Sabathe cycle

Between points 1 and 2 gas gets adiabatically compressed and at point 2, at volume V_2 it reaches temperature of T_2 and pressure at p_2.
From the point 2 to 3, gas is hit with heat at $Q_{23} = mc_v(T_3-T_2)$, and from 3 to 3' - isobarically - heat at $Q_{33'} = mc_p(T_{3'}-T_3)$.
Between points 3 and 4, the factor inside the cylinder is adiabatically decompressed, between 4 and 1 isochoric heat removal occurs at $Q_{41} = mc_v(T_4-T_1)$.
Because the circulation is closed, there is no mass exchange with the environment.
During compression and decompression the cylinder walls are tightly insulated.
Thermal parameters of the points 2, 3, 3' , and 4 are calculated using the adiabatic equations as well as equations for the gas state.

1-2 compression:

Using an adiabatic equation:

$$pV^k = idem,$$

where k = idem is:

$$p_1V_1^k = p_2V_2^k$$

hence:

$$p_2 = p_1(V_1/V_2)^k$$

Given V1/V2 = ε (geometrical compression ratio), the result is:

$$p_2 = p_1\varepsilon^k$$

Given this relation, the temperature in point 2 is evaluated using the state equations

$$p_1V_1 = mRT_1,$$
$$p_2V_2 = mRT_2.$$

The result is following:

$$T_2 = T_1\varepsilon^{k-1}. \qquad (14.1)$$

If is the ratio of pressures p_3/p_2, ρ is the ration of volumes $V_{3'}/V_3$, then

$$p\ p\phi = p_3/p_2, \qquad \rho = V_{3'}/V_3 \qquad (14.2)$$

Since the 2-3 is an isochoric transformation, V = idem, it is safe to conclude

$$p_3 = p_2\,\phi,$$

and taking into consideration that $\boldsymbol{p_2 = p_1\varepsilon^k}$**,**

$$\phi p^3 = p_1\phi\,\varepsilon^k.$$

Correspondingly

$$\phi T_3 = T_1\square\phi\,\varepsilon^{k-1}.$$

(14.3)

From the isobaric process flow of 3-3’ it follows that

$$\phi p_3 = p_{3'} = p_1\phi\,\varepsilon^k$$

as well as

$$\phi T_{3'} = T_1\phi\,\rho\,\varepsilon^{k-1}. \qquad (14.4)$$

It is possible to evaluate the point 4 parameters using an adiabatic equation for the decompression process

$$p_{3'}\,V^k_{3'} = p_4\,V^k_4$$

where

$$p_4 = p_{3'}\,(V_{3'}/\,V_4)^k \qquad (14.5)$$

Because the 3-4 decompression equals the volume ratio $V_4/V_{3'}$, the relation is

$$\frac{V_4}{V_{3'}} = \frac{V_4}{V_3}\frac{V_3}{V_{3'}}$$

As a substitute

$$\varepsilon = \frac{V_1}{V_2} = \frac{V_4}{V_3} \qquad \text{and} \qquad \rho = \frac{V_{3'}}{V_3}$$

It gives an equation for the decompression ratio:

$$\varepsilon_r = \frac{V_4}{V_{3'}} = \frac{\varepsilon}{\rho} \qquad (14.6)$$

It results from (14.6), that the decompression ratio ε_r is lower than the compression ratio ε. Substituting (14.6) for (14.5) gives:

$$p_4 = \frac{p_{3'}}{\varepsilon_r^k} = p_{3'}\frac{\rho^k}{\varepsilon^k} \qquad (14.7)$$

Substituting the right side of the equation (14.4) for ***p3'***, gives

$$\phi\rho p_4 = p_1 \phi\, \rho^k.$$

(14.8)

Another interrelation can be derived in a similar way

$$\phi\, \rho T_4 = T_1 \phi\, \rho^k.$$

(14.9)

In the table below, there are all the equations for pressures and temperature for points 2, 3, 3' and 4, put together in correlation with the pressure and temperature for point 1.

Table 14.1. Comparison between equations for pressures and temperatures for points 2, 3, 3' and 4

Point 1	**2**	**3**	**3'**	**4**
p_1	$p_2 = p_1\varepsilon^k$	$p_3 = p_1\,\phi\,\varepsilon^k$	$p_3 = p_1\,\phi\,\varepsilon^k$	$p_4 = p_1\phi\,\rho^k.$
T_1	$T_2 = T_1\varepsilon^{k-1}$	$T_3 = T_1\phi\,\varepsilon^{k-1}$	$T_3 = T_1\phi\,\rho\,\varepsilon^{k-1}$	$T_4 = T_1\,\phi\,\rho^k$

Deriving the equation for the Sabathe' cycle efficiency leads to a statement that the difference between the heat energy supplied in total to 2-3 and 3-3' and the energy drained from 4 and 1 equals technical work, illustrated with the crosshatched area of the diagram 14.1. The cycle efficiency, equal to the ratio of the heat turned into work to the added heat, is

$$\eta_t = \frac{mc_v(T_3-T_2)+mc_p(T_{3'}-T_3)-mc_v(T_4-T_1)}{mc_v(T_3-T_2)+mc_p(T_{3'}-T_3)}$$

Replacing T_2, T_3, $T_{3'}$, T_4 with the equations from the Table 14.1, after simplifying it and substituting $\boldsymbol{k = c_p / c_v}$, the new relation is:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}}\frac{\varphi\rho^k - 1}{\varphi - 1 + k\varphi(\rho - 1)} \tag{14.10}$$

Considering the isochoric and isobaric heat addition, the presented cycle is called a mixed cycle. It is the best match for the thermodynamic process of the spontaneous ignition engine with a direct fuel injection. The exceptions are the following:

b) with the isochoric heat addition (given ρ=1),
d) $\phi$with the isobaric heat addition (given ϕ=1),

A process with an isochoric heat addition, also called the Otto cycle, is applied as a comparable pattern of thermodynamic processes in spark-ignition engines.
ρTheoretical efficiency in this case is obtained by substituting $\boldsymbol{\rho = 1}$ in (14.10), where

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \tag{14.11}$$

ηThis formula, being a special case of (14.10), gives the highest η value for the chosen compression ratio.
Theoretical efficiency, with the isobaric heat addition, can be obtained by substituting =□1 in (14.10).

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}}\frac{\rho^k - 1}{k(\rho - 1)} \tag{14.12 }$$

A cycle with the isobaric heat addition shows similarities to the combustion process, which occurs in big slow-speed compression-ignition engines. Combustion in out-of-use air-blast injection engines is very alike. This cycle is called the Diesel cycle.

ηComparing efficiencies of the Sabathe' Cycle (formula 14.10), Otto cycle (formula 14.11) and Diesel cycle (formula 14.12), where the combustion ratio $\boldsymbol{\varepsilon}$ is the same - $\boldsymbol{\eta_t}$ reaches the highest value for the Otto cycle, medium for the Sabathe' one and the lowest for the Diesel one.

Figure 14.2 shows the comparison of the Sabathe, Otto and Diesel diagrams, at the same ε value.

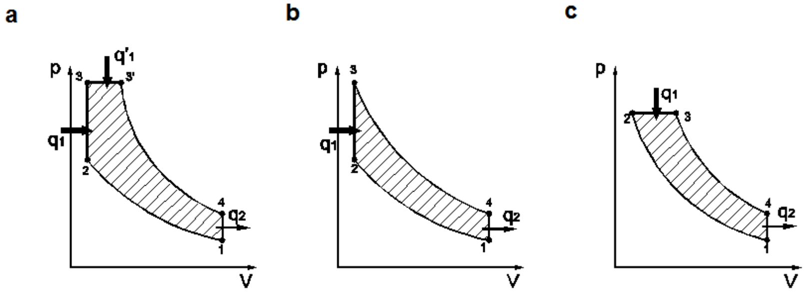


Fig. 14.2. The cycle comparison a-Sabathe', b-Otto and c-Diesel

Analysing the formulas for the theoretical efficiency η_t results in a conclusion that its value increases with the compression ratio ε. If the compression ratio is heading towards infinity the efficiency η_t is heading towards unity. The curve η_t, presented in Fig. 14.3, using the simplest formula (14.11), shows a high increase for small ε values. The large values show a slower increase. The horizontal line, corresponding with the ordinate equalling unity, is called asymptote η_t.

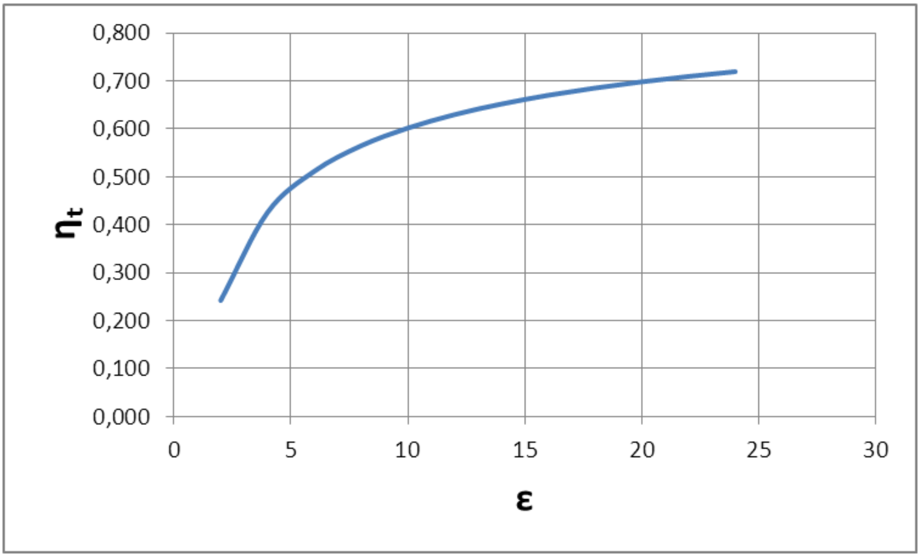


Figure. 14.3. Theoretical efficiency

The range **6 < ε < □10** approximately corresponds with the typical engine solutions with spark ignition, whereas **13 < ε < 23□** corresponds with spontaneous ignition. The selected ranges and the border values are estimated and should not be taken for final and impassable.

In case of the spontaneous ignition engines, the value of the compression ratio ε, is usually higher than 15. The choice comes from the necessity for an easy starting (the minimum required air temperature at the end of compression process for a cold engine is 350-400 ^{o}C). For the multi-fuel engines the air temperature at the end of the compression should be even higher – above 600 ^{o}C. It is necessary to bring the compression ratio up to $\varepsilon \geq 20$.

Comparative work circulation with the real factor

The comparative circulation with the real factor, Fig. 14.4, is comparable with the Sabathe cycle. Nevertheless, there are some significant differences between the two of them, than need to be discussed in detail.

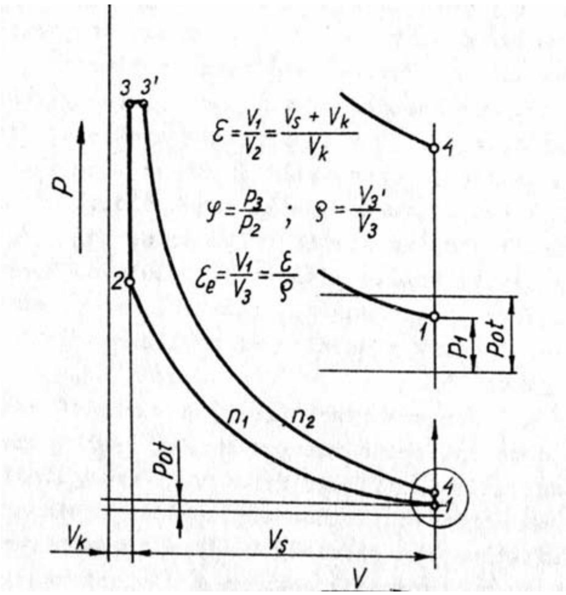


Figure. 14.4. Comparative circulation with the real factor

Gas, a thermodynamic factor in a comparative circulation, is a semi-ideal gas, unlike in the Sabathe cycle, which is based on ideal gases. Gases, when they are at much higher pressure and temperature than required for gas to turn into liquid and, at the same time, lower than the ones that cause dissociation, have qualities of semi-ideal gases. The specific heat of semi-ideal gases is contingent on temperature. If the temperature range is narrowed down to positive temperatures up to 2500 ^{o}C, it is observable that, at the constant volume c_v and constant pressure c_p, the specific heat increases constantly. As a result, the higher the temperature - the lower the value of the adiabatic exponent of the semi-ideal gases. The root formulas are

$$k = \frac{c_p}{c_v} \qquad \text{and} \qquad R = c_p - c_v,$$

resulting in

$$k = 1 + \frac{R}{c_v} \quad (14.16)$$

Since the individual gas constant **R** depends only on the gas type, not the temperature, it is obvious from (14.16) that, if the specific heat increases, the adiabatic exponent value drops.

Evaluating the comparative circulation, it is assumed, that the heat travels from gas into the cylinder walls and vice versa, in contrast with the theoretical circulation. At the start of the compression process, heat moves from the heated cylinder walls towards the cooler gas. At the end, the heat path is reverse, because the gas temperature is higher than the temperature of the cylinder walls.

It is quite a complicated process, which, for the calculative purposes, is replaced with a polytropic index n_1.

The index value is:

$$n_1 = 1{,}35\text{-}1{,}40$$

During the entire decompression process the heat travels from the hot gas towards much cooler cylinder walls. Still, simultaneously to this occurrence, at the beginning of the decompression (pass point 3 in Fig 14.4) there is the effect of dieseling, which gives extra heat.

A complicated decompression process as it is, can be explained as a series of polytropic processes with different polytropic indexes. It is accepted to take on just one replacement polytropic index during the calculations. The value of it is

$$n_2 = 1{,}22\text{-}1{,}28.$$

It can be observed that the decompression polytropic index is much lower in value comparing to the compression one for the following reasons:

- Decompression temperature is much higher than the compression one The adiabatic exponent of the semi-ideal gas drops as the specific heat increases (formula 14.16)
- There is a substantial amount of fumes in the factor. Fumes contain carbon dioxide (CO_2) as well as (overheated) water vapour. Both components, carbon dioxide and water vapour especially, are

characterised by very low adiabatic index. Dieseling at the beginning of the decompression is very important because, as its result, significant amount of heat is introduced into the factor despite the cooling effect that the cylinder walls would have. This reaction causes the decompression polytropic index to drop, which continuously stimulates drop in the average index.

The engine speed increase substantially affects the increase of n_1 and decrease of n_2, since the heat flow towards the cylinder walls is weaker.

The value of the linear dimension (mainly the cylinder diameter) is also vital. If the D diameter goes up, the ratio of the cooling surface and the factor volume grows smaller. Consequently, there is less heat moving per factor volume unit, which causes increase of n_1 and decrease of n_2.

It is to be remembered that, not like in the theoretical process, in the comparative circulation there is a variable factor number during combustion.

It is important to realise, before the calculations regarding the comparative circulation with the real factor even starts, that even if they are a step towards establishing the real processes that occur inside a running cylinder, they cannot be used as proof due to simplified assumptions, which have to be used to make the complex calculations even possible. In practice, even if the course of one thermal parameter of the gas inside the cylinder – meaning the pressure – qualify as approximately correct, it is impossible to get such accuracy with temperature flow, at least not during combustion. The reason for it is that the combustion process is assigned to the mixed isochoric-isobaric heat addition, which is wrong. The maximum temperature values are not taken into consideration since they are too high in comparison to the ones corresponding with the real circulations at the same indicated pressure and the same maximum pressure.

Calculating the average indicated pressure p_i. Definition of p_i and p_e

The average indicated pressure p_i is constant pressure, which, during one piston cycle, does work equal work represented in an indicator diagram

It can be estimated through comparative measuring of the indicator diagram by a planimeter. The value of p_i can also be calculated using theoretical interrelations.

Knowing p_i pressure, given η_m value (mechanical efficiency), allows calculating effective pressure $\boldsymbol{p_e}$ or establishing $\boldsymbol{\eta_m}$, basing on $\boldsymbol{p_e}$ determined using other methods (e.g. engine testing on a test bed).

$$p_e = p_i \, \eta_m$$

The area of the comparative diagram can be compared to indicated work.

$$L_i = V_s \, p_i$$

In the comparative diagram (Fig. 14.5)

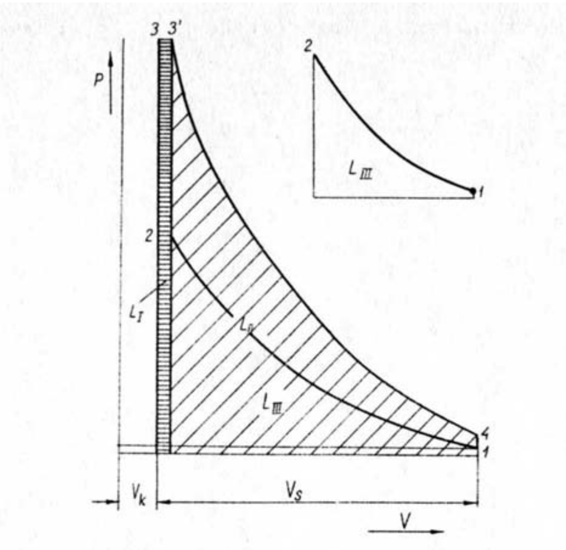


Figure. 14.5. Comparative indicator diagram

$$L_i = L_I + L_{II} - L_{III}$$

hence

$$V_s \, p_i = L_I + L_{II} - L_{III}$$

(14.17)

Work $\boldsymbol{L_I}$ equals

$$\phi\phi L_I = p_3(V_{3'} - V_3) = p_3 V_3 (\rho - 1) = \phi p_2 V_k (\rho - 1) = \phi \, \varepsilon^{n1} p_1 V_k(\rho - 1)$$

Since

$$\varepsilon = (V_s + V_k) / V_k$$

it leads to

$$L_I = \varphi\varepsilon^{n}{}_{1}{}^{-1}(V_s + V_k)p_1{}^{(\rho-1)}$$

(14.18)

Work L_{II} equals the absolute work of the polytropic decompression from point 3’ to 4

$$L_{II} = \frac{p_4 V_4}{n_2 - 1}\left(\varepsilon_r^{n_2 - 1} - 1\right)$$

After substituting p_4 and transforming it, the result is

$$L_{II} = \varphi\rho^{n_2}\frac{\varepsilon^{n_1 - n_2}}{n_2 - 1}p_1(V_s + V_k)\left(\varepsilon_r^{n_2 - 1} - 1\right)$$

(14.19)

Work L_{III} equals the absolute work of polytropic compression from point 1 to 2

$$L_{III} = \frac{p_1}{n_1 - 1}(V_s + V_k)\left(\varepsilon_r^{n_2 - 1} - 1\right)$$

(14.20)

After substituting (14.18), (14.19) and (14.20) for (14.17) and transforming it, the result is

$$p_i = \frac{\varepsilon p_1}{\varepsilon - 1}\left[\varphi(\rho - 1)\varepsilon^{n_1 - 1} + \varphi\rho^{n_2}\frac{\varepsilon^{n_1 - n_2}}{n_2 - 1}\left(\varepsilon_r^{n_2 - 1} - 1\right) - \frac{\varepsilon^{n_1 - 1}}{n_1 - 1}\right]$$

(14.21)

15. COMPRESSORS

The purpose of compression is:

a) to increase the factor pressure,
b) to store the factor by increasing its density,
c) to increase the factor temperature.

There are two types of compressors, relying on the built type and the method of compression:

a) piston
b) rotodynamic
c) rotary

In rotary compressors, the pressure goes up by decreasing the gas volume, which is done by a rotation engine inside the compressor chamber. It is a similar process as in the piston compressors, in which the piston moves in an alternating motion (Fig. 15.2).

Rotodynamic compressor

A rotodynamic compressor (Fig 15.1) comprises of

- a rotor accelerating gas
- a stator, it brings gas to a stop.

A rotor and a stator are so-called stages.

A **compression ratio** attained from one stage of multi-stage rotodynamic compressor is not big (within p_2 / p_1 = 1,25-1,20 or less).

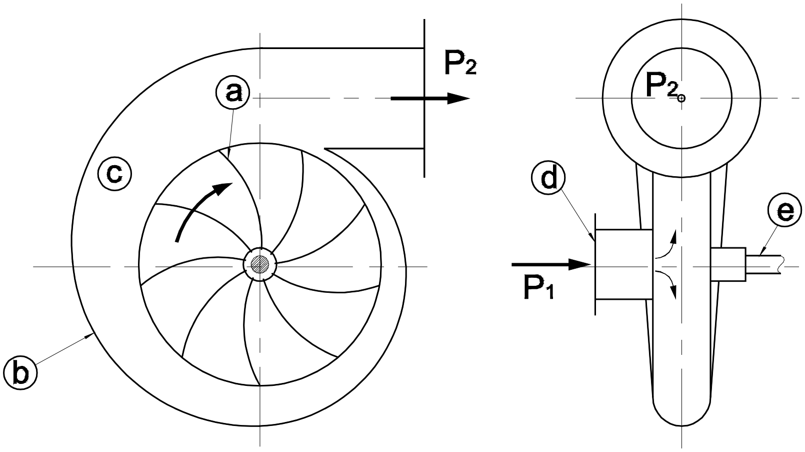


Fig. 15.1. Centrifugal single-stage compressor, a – rotor, b – stator, c – diffuser, d – inlet, e – transmission shaft

Reaching higher **compression ratio** requires more stages.

Depending on the gas flow direction in the rotor, there are **centrifugal compressors** (it employs a radial motion in relation to the axis of rotation) and **axial compressors** (the fluid flows parallel to the axis of rotation).

A single-stage rotodynamic compressor is also called a **ventilator**, if the pressure increase does not go over 100 mm H_2O.

\

Piston compressors

In piston compressors, pressure increases due to decreasing gas volume inside the cylinder caused by the alternating piston movement (Fig. 15.2).

To control the fluid flow, valves are used (usually self-closing), i.e. they close and open influenced by the pressure differences.

A **suction valve** opens, when the cylinder pressure is lower that the pressure inside a suction pipe 1, and it closes when the cylinder pressure starts increasing.

A **delivery valve** opens, when there is gas pressure surplus in the cylinder, comparing to the gas pressure in a delivery pipe 2, and it closes when the cylinder pressure starts dropping.

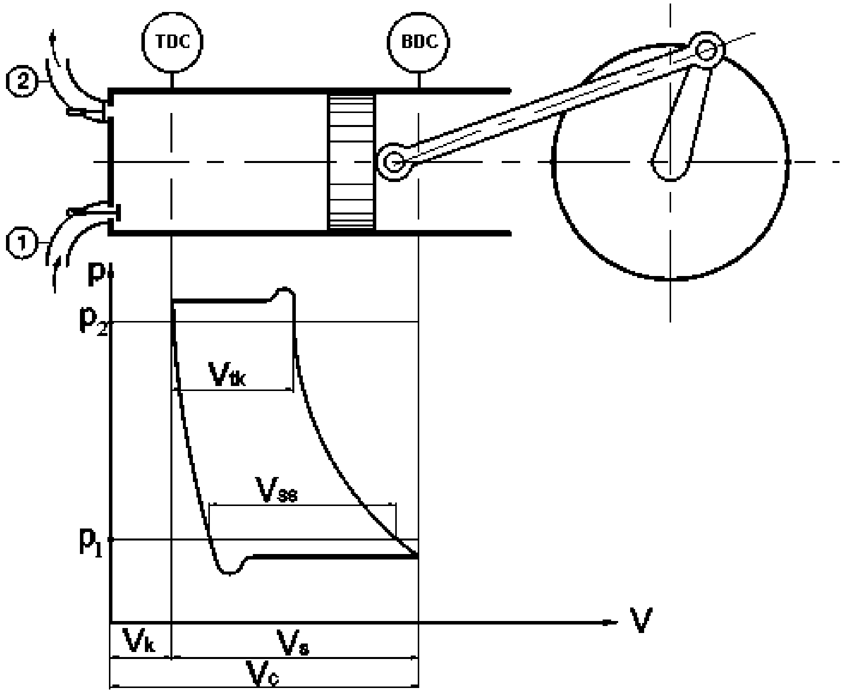


Fig. 15.2. Piston compressor diagram: V_{ss} – suction volume, V_{tk} – delivery volume,

For the constructive reasons, every piston compressor has got so-called **clearance volume V_k**. It is volume, sealed off by a piston in its TDC.

A piston, moving between TDC and BDC marks out so-called **engine displacement V_s**.

The sum of the engine displacement and clearance volume is called **complete volume V_c**.

15.1. Mechanical efficiency of a compressor η_m

$$\eta_m = \frac{L_i}{L_e} \qquad (15.1)$$

To evaluate a compressor, it is compared with a so-called **ideal** compressor.

If the compression purpose is to increase pressure of fluid storing, an ideal compressor that will do the job with the minimum work is a **reversible isothermal compressor, marked T at point 2 of the diagram,** (Fig. 15.3).

$$p\,\upsilon = R\,T$$

$$T = idem,\ R = const;\ p\,\upsilon = idem$$

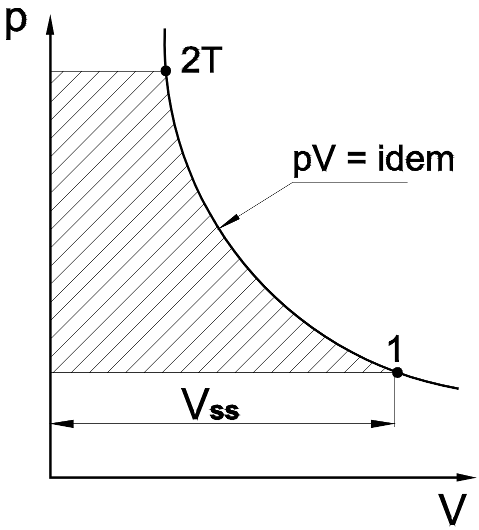


Fig. 15.3. Diagram of ideal work of an isothermal compressor

The comparison of the indicator diagrams of a real and ideal isothermal compressor is illustrated with Fig. 15.4.

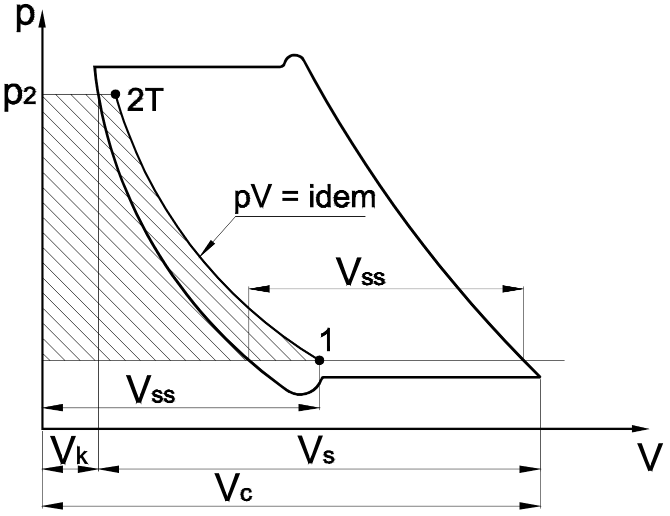


Fig. 15.4. Indicator diagram of a real compressor and an ideal isothermal compressor: V_c*- complete volume,* V_s *– engine displacement,* V_k *– clearance volume,* V_{zs} *– suction volume.*

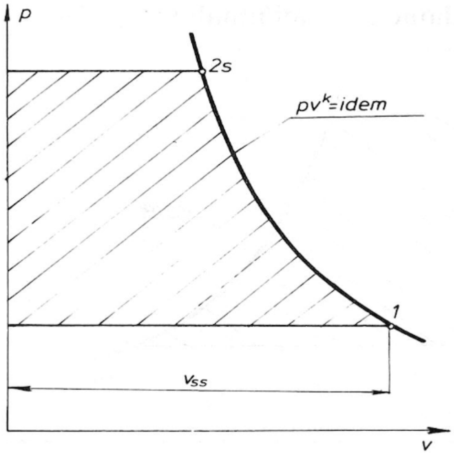
An **ideal** compressor is tight and for the gas suction it uses the total engine displacement, provided that clearance volume is $\boldsymbol{V_k = 0}$.

It uses the same gas quantity per cycle as a real compressor.

An ideal isothermal compressor requires cooling system.

A cooling system in a real compressor lowers the required work input.

If the compression purpose is to **increase temperature**, this kind of compressor does not need a cooling system. An ideal compressor for the described process is a reversible adiabatic compressor, i.e. **isentropic compressor,** marked s in point 2 of the diagram (Fig. 15.5). It is based on the process $pv^k = idem$.



Rys. 22.4. Wykres pracy idealnej sprężarki izentropowej

Fig. 15.5. Diagram of ideal work of an isentropic compressor

The work of an ideal isothermal and isentropic compressor is illustrated with Fig. 15.6.

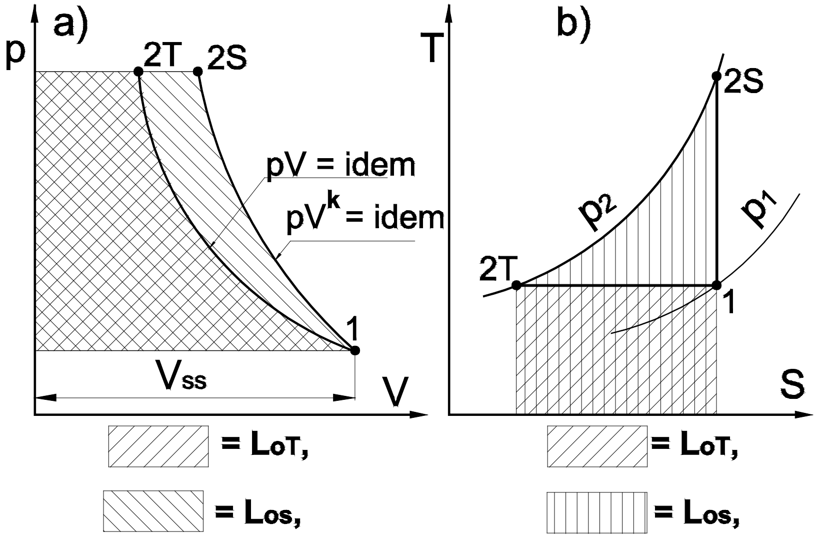


Figure15.6. Contrasted work of the ideal compressors: Isothermal and isentropic

15.2. Indicated efficiency

Indicated efficiency of a compressor is the ratio of work done by an ideal compressor to work done by a real indicated compressor. It equals:

for the cooled compressors

$$\eta_{iT} = \frac{L_{oT}}{L_i} \qquad (15.2)$$

For the not-cooled compressors

$$\eta_{is} = \frac{L_{os}}{L_i} \qquad (15.3)$$

The total compressor effectiveness is the ratio of work done by an ideal compressor to the effective work (isothermal compression):

$$\eta_{eT} = \frac{L_{oT}}{L_e} \qquad (15.4)$$

which can be transformed into

$$\eta_{eT} = \frac{L_{oT}}{L_i}\frac{L_i}{L_e} = \eta_{iT}\eta_m$$

or in case of isentropic compression:

$$\eta_{es} = \frac{L_{os}}{L_e} \qquad (15.5)$$

after the transformation

$$\eta_{es} = \frac{L_{os}}{L_i}\frac{L_i}{L_e} = \eta_{is}\eta_m$$

15.3 Compression work

An **isothermal** compressor is an ideal compressor to achieve the same compression ratio ***p₂/p₁*** of the given gas quantity applying the **minimum work**.

It employs the maximum cooling of fluids comparing to the environment temperature, i.e. the temperature during suction.

Cooling below the environment temperature employs more work (e.g. refrigerators).

Gas in the **real** compressors, even during heavy cooling of the cylinder during the compression process, does not reach the environment temperature, which means that compression **is not isothermal**.

The **maximum temperature** at the end of compression is reached only during the **adiabatic** compression (higher than that is achieved only by heating gas up during the process from the outside).

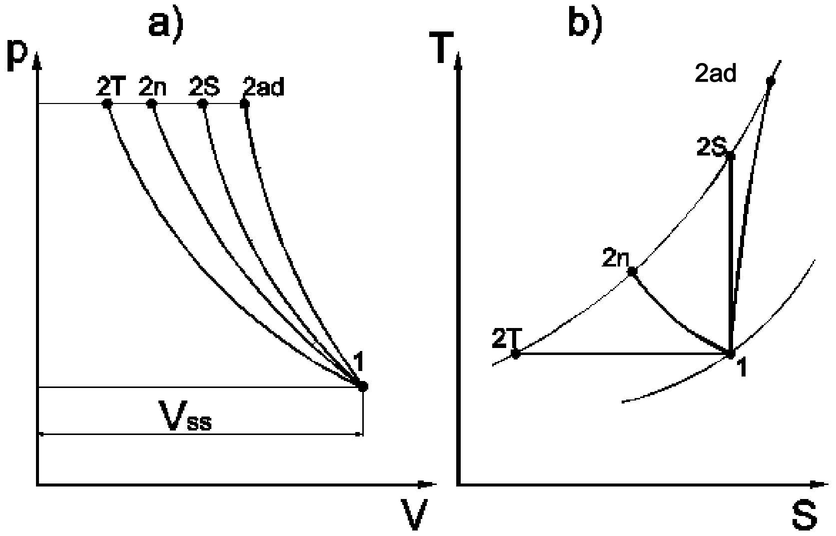


Figure. 15.7. Course of the polytropic compression: 2T – isothermal, 2n – polytropic, 2S – isentropic, 2 ad – adiabatic with friction

Figure. 15.7 presents compression course in the polytropic form without cooling, with partial cooling and with full cooling down to the environment temperature. It proves that **the higher the temperature at the end of the compression process, the greater the work put into the process**.

16. HEAT TRANSFER

Heat transfer is present, when there is a temperature difference inside some system or between a number of systems, which can interact. According to II LT, the systems transfer energy as well, while a part of the system or the system with the higher temperature transfers energy onto the lower temperature system.

Cogitations of this kind are the subject of the heat transfer studies. Quantative relations, defining the quantity of exchanged energy, obey I LT.

There is the close and substantial correlation between the heat transfer studies and thermodynamics. Since the temperature difference is absolutely vital as a condition for heat transfer, heat transfer studies analyse cases with no thermodynamic balance. Thus it is a science based on unbalanced thermodynamic relations.

There are two very important consequences of that conclusion: the first one relates to the question, whether the basic concepts and laws of thermodynamics, defined basing on a balanced thermodynamic system, could be applied to an unbalanced one.

That question could be answered positively only if it is possible to separate a part of that system and define in it the momentary values of the thermodynamic parameters, i.e. temperature, pressure and specific volume. In all the cases, which heat transfer studies focus on, this condition has been prevailed and therefore the basics of thermodynamics can be applied.

The second result of the thermodynamic unbalance is that in all the cases of heat transfer time plays a very important role, while it is not present in the basic laws of thermodynamics.

The theory of thermal conduction is a part of applied mathematics and it amounts to solving partial differential equations (PDE) with various boundary conditions.

Convection relates to fluids motion – hydrodynamics – and many of the interrelations are based on semi-empirical methods of dimensional analysis and the theory of probability.

Lastly, heat radiation is mainly based on the achievements of theoretical physics, focusing on the radiation theory.

Technical questions concerning heat transfer process are usually quite complicated and very often require many simplifications in order to solve equations for particular cases. Those simplifications have to be adequately applied to the theoretical solutions of the practical problems

In practice, heat transfer issues amount to calculating the amount of heat transferred from one system to another or within just one analysed system.

The heat transfer problem diversity is immense and all those issues touch different scientific fields.

16.1. Primary types of heat transfer

Talking about the types of heat transfer, there are three major ones:

a) thermal conduction
b) convection
c) radiation

The difference is in the mechanism for the transfer of thermal energy, but in practice those types are rare in their pure form. They usually come in certain combinations, which should be allowed for in the calculations.

Thermal conduction is phenomenon of energy transfer inside a material medium or from one medium to another by a direct physical contact, from the higher temperature to the lower one, where single particles of the analysed system do not show major movement.

This type of heat transfer is consistent with solids. For liquids and gases, pure thermal conduction, without any additional heat transfer methods involved, is very rare.

Convection takes place, when single molecules, that carry heat, change their position.

This process is characteristic of liquids and gases, while heat transfer is done mainly by fluid mixing but, to some extent, through conduction as well.

Thermal radiation – heat is transferred by quanta of electromagnetic radiation at a certain range of the wave length. Not like thermal conduction and convection, thermal radiation does not need a medium to travel in, because it propagates in vacuum as well.

Regardless the introduced classification, heat transfer divides into stable and unstable. Stable heat transfer happens when the temperature distribution in the analysed system does not change over time and the transferred heat quantity stays the same. During the unstable heat transfer, the temperature distribution and the transferred heat quantity change over time.

In practice, most of the cases concern the stable heat transfer.

16.2. Basic concepts and interrelations of heat transfer

The main purpose of solving the heat transfer problems is to calculate the quantity of heat *Q* transferred in the investigated system. Such a system is usually restricted to some surface *F*, e.g. surface of a heat exchanger, and, in practice, it is the heat quantity transferred by that surface that is calculated.

Next to a total heat quantity ***Q***, heat flux density is another often used term. Heat flux density ***q*** is a quantity described with the following equation

$$q = \lim_{\Delta F \to 0} \frac{\Delta Q'}{\Delta F} \qquad (16.1)$$

where *ΔQ'* is the heat quantity transferred by the surface ***ΔF*** per certain time unit.

If it is a stable heat transfer, than ***q*** for the given area of the surface does not change over time, in case of unstable heat transfer, ***q*** is a time function.

At the fixed heat transfer, heat flux density can be stable on the entire surface, or its value can change upon the location of the element ***ΔF*** within the surface area. In the first case it is correct to assume:

$$q = Q' / F$$

the total heat quantity transferred by the surface ***F*** within time ***τ*** can be calculated with the following equation:

$$Q = Q' \tau = q\, F\, \tau \qquad (16.2)$$

If ***q*** on the considered surface is variable, the equation for calculating ***Q*** is as follows:

$$Q = \int_F q \tau dF$$

As it results from the above, the heat flux density equals the heat quantity transferred within a unit of time through 1 m^2 of surface, with the given heat transfer and even heat transfer through the entire heating surface.

Thermal conduction

Heat conduction applies **Fourier's law**, which says that heat flux density is proportional to the temperature gradient, measured along the heat transfer flow.

The mathematical representation of it is following:

$$q = -\lambda \frac{dt}{dx} \qquad (16.3)$$

The minus sign in (16.3) results from the fact that heat flows from a hotter spot to the cooler one, which means the segment *dx*, measured along the heat transfer direction, corresponds with the negative value of the temperature increase *-dt*.

The constant for proportionality *λ* is also called a **thermal conductivity factor**, it characterises the medium's ability to conduct heat. The unit of thermal conductivity is [λ] = W/m K.

If ***F*** is the size of the surface measured perpendicularly to the heat transfer flow, the quantity of the heat conducted within the unit of time through that surface is:

$$Q' = -\lambda F \frac{dt}{dx}$$

Given that the fixed heat conduction through a ***δ*** thick flat wall at the heat conductivity ***λ***, which does not depend on temperature, as well as the wall surface temperatures are constant and equal t_w and t_z (Fig. 16.1), the heat flux density can be calculated with the following formula:

$$q = \frac{\lambda}{\delta}(t_w - t_z) \qquad (16.4)$$

which results from (16.3).

For walls of a different shape, the equations are more complex, but can be always brought down to

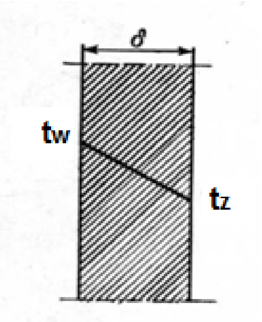


Figure. 16.1. Thermal conductivity through a flat wall

$$q = \frac{\lambda}{\delta}\varphi(t_w - t_z) \qquad (16.5)$$

where:

φ – factor depending on the shape of the conducting wall,

t_w, t_z – temperatures of the wall surface.

The total heat quantity conducted through the surface *F* of the wall is:

$$Q' = \frac{\lambda}{\delta}\varphi F(t_w - t_z) \qquad (16.6)$$

and for a flat wall $\varphi = 1$.

Using the equation (16.6) in reference to a wall in the shape other than flat, the surface of reference F, which might change along the heat flow direction and the size of which influences the factor φ, has to be carefully chosen.

Adopting heat

The heat transfer inside fluid employs convection, although very close to the wall there is a layer that uses conduction to transfer heat.

It is illustrated with Fig. 16.2, which shows the temperature distribution in the fluid close to the wall emitting heat.

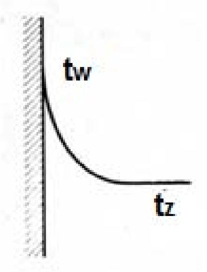


Figure. 16.2. Temperature distribution in fluid close to the wall

During the fixed heat transfer, the constant heat flux density ***q*** travels from the wall into the fluid and the more intense the heat transfer is, the smaller the temperature drop per a length unit perpendicularly to the wall. There is a significant temperature drop close to the wall, caused by the fact that in that thin layer conduction dominates the heat transfer comparing to the fluid further away from the wall, where there is also convection.

Heat transfer between the wall and the fluid is called **heat adoption**. The mathematical representation of it is as follows and it is called **Newton's law of cooling**.

$$q = \alpha(t_w - t_f), \qquad (16.7)$$

where:
t_w – wall temperature,
t_f – fluid temperature measured far from the wall, where the method of calculating t_f is not predetermined.

The quantity describing the heat transfer intensity is called a ***heat adoption factor***. The unit for that factor is $[\alpha] = W/m^2\ K$.

If heat travels from the fluid flow to the wall, then $t_w < t_f$ and in the equation (16.7) $t_f - t_w$ substitutes for $t_w - t_f$.

The value of the heat adoption factor may vary across the whole considered surface. Therefore it is important to distinguish between the local value α_{lok} of the factor and its average value α. The local value α_{lok} is defined by the equation (16.7) and since according to (16.1)

$$q = \frac{dQ'}{dF}$$

hence

$$\alpha_{lok} = \frac{dQ'}{dF}\frac{1}{t_w - t_f}$$

which means that α_{lok} refers to a very small part of the surface dF, surrounding the considered point.

The average value of the heat adoption factor relates to the local one as follows:

$$\propto = \frac{1}{F}\int_F \propto_{lok} dF \qquad (16.8)$$

If the value of $\boldsymbol{\alpha}_{\mathbf{lok}}$ is constant for the considered surface then the equality applies:

$\alpha = \alpha_{\text{lok}}$

The quantity of heat transferred within a unit of time through adoption by the surface ***F*** is:

$$Q' = \alpha F \Delta T \qquad (16.9)$$

where ΔT is the difference between the average surface temperature and the average fluid temperature.

Thermal radiation

In the theory of thermal radiation the concept of a **black body** is crucial. It is an idealized physical body that absorbs all incident electromagnetic radiation, without reflecting or emitting any of it.

Thermal radiation obeys the Stefan-Boltzmann law, which states that the total heat radiated by a black body is directly proportional to the fourth power of the black body's absolute temperature.

The mathematical representation of it is:

$$q = C_0 (\frac{\tau}{100})^4$$

where $\boldsymbol{C_0}$ is called **the Stefan-Boltzmann** constant.

The total heat radiated by the unit surface ***F*** in the unit time is:

$$Q' = C_0 F (\frac{\tau}{100})^4 \qquad (16.10)$$

Real bodies are not black bodies and at the same temperature radiate less energy than a black body.

If the ratio of heat radiated by a real body to heat radiated by a black body in the same conditions does not depend on the length of the radiation wave, that body is called **a grey body**.

The heat transfer between grey bodies is:

$$Q_{1-2} = C_0 F_1 \Phi_{1-2} [(\frac{T_1}{100})^4 - (\frac{T_2}{100})^4] \qquad (16.11)$$

where:

T_1 and T_2 — temperatures of bodies transferring heat,
F_1 — body surface at temperature T_1,
Φ_{1-2} - factor allowing for real body qualities deviation from the black body qualities and the geometric system of the two bodies.

Complex heat transfer

Heat transfer basing on pure conduction, convection or radiation is a very rare phenomenon. Those three major heat transfer types usually appear at the same time in different combinations.

Technologically important example is the case of heat transfer from one medium to another through a dividing solid wall. It is a complex process consisting in adopting heat through a wall from hot fluid, conducting it through the same wall and then acquiring it from it through much cooler fluid.

Naturally, the heat transfer between the wall and environment on either side of the wall can happen through conduction, convection or radiation.

The example, illustrated in Fig. 16.3, shows the diagram of temperature distribution in both fluids and the wall itself; it is called **thermal transmission**.

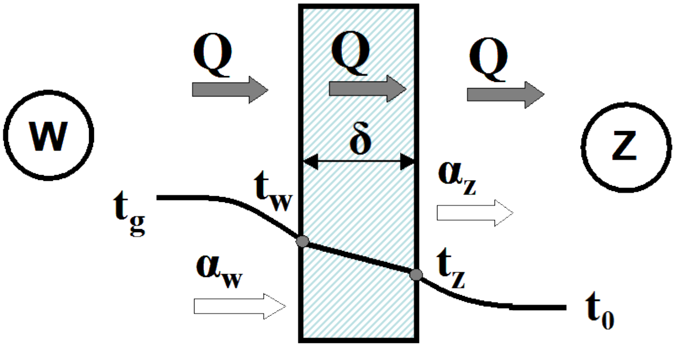


Figure 16.3. Thermal transmission through a flat wall

Assuming, that the wall surface temperatures are constant and equal t_w and t_z, as well as heat transfer is fixed, it results in an equation, describing heat transfer through adoption between the wall and two fluids, and through conduction in the wall itself:

$$Q' = \alpha_w F_w (t_g - t_w) \qquad (16.12)$$

$$Q' = (\lambda/\delta)\, \varphi\, F_o (t_w - t_z) \qquad (16.13)$$

$$Q' = \alpha_z F_z (t_z - t_0) \qquad (16.14)$$

In the equations mentioned above, $\boldsymbol{F_w}$ and $\boldsymbol{F_z}$ define the wall surfaces on the cool fluid side as well as the hot fluid side, which do not have to equal; F_0 is the surface of reference for calculating the φ factor and the heat transfer via conduction.

Usually, it is assumed that F_0 equals F_w or F_z.

If the heat transfer between the wall surface and the fluid took place through convection and radiation, the factors α_w and α_z would have to be substituted with $\alpha + \alpha_\tau$.

The equations (16.12) ÷ (16.14) allow to determine the temperature differences

$$t_g - t_w = \frac{Q'}{\alpha_w\ F_w}$$

$$t_w - t_z = \frac{Q'}{\varphi\, F_0} \frac{\delta}{\lambda}$$

$$t_z - t_0 = \frac{Q'}{\alpha_z\ F_z}$$

Adding the sides of those equations up correspondingly gives

$$t_g - t_0 = Q'\left(\frac{1}{\propto_w F_w} + \frac{1}{\varphi F_0}\frac{\sigma}{\lambda} + \frac{1}{\propto_z F_z}\right)$$

which can transform into

$$Q' = \frac{1}{\frac{1}{\propto_W F_W} + \frac{1}{\varphi F_0}\frac{\delta}{\lambda} + \frac{1}{\alpha_Z F_Z}}\left(t_g - t_0\right) \qquad (16.15)$$

The equation (16.15) can be brought down in form to the Newton equation (16.17), describing the heat adoption, which is:

$$Q' = kF_0\left(t_g - t_0\right) \qquad (16.16)$$

where

$$k = \frac{1}{\frac{F_0}{\propto_W F_W} + \frac{1\,\delta}{\varphi\lambda} + \frac{F_0}{\propto_Z F_Z}} \qquad (16.17)$$

is called the **heat transfer coefficient.**

Surface $\boldsymbol{F_0}$ is the surface of reference, allowing to determine the coefficient $\boldsymbol{k}$.

In the case of the flat wall bordering the two media $\boldsymbol{F_0 = F_w = F_z}$ $\boldsymbol{(\varphi=1)}$, the equations (16.15) and (16.17) get simplified as follows:

$$k = \frac{1}{\frac{1}{\propto_W} + \frac{\delta}{\lambda} + \frac{1}{\propto_Z}} \qquad (16.18)$$

The unit of the coefficient k is $[k]$=W/m^{2}K, which is the same as the unit of thermal transmission factor.

In general case, when $\boldsymbol{F_w \neq F_z}$, the heat transfer coefficient k usually refers to only one of the surfaces, i.e. $F_w = F_0$, or $F_z = F_0$ and then the equation (10.13) can be adequately simplified.

From (16.17) and (16.18) it results, that from the three quantities: α_w, α_z i λ/δ the biggest influence on the coefficient k has the expression of the lowest value. That is why, in practice, it is assumed that $k \approx \alpha_w$ if the value of λ/δ and α_z are significantly higher, comparing to α_w.

17. REFRIGERATING DEVICES

Lowering the object's temperature to the temperature of the environment does not require special devices.

It can be done by e.g. direct cooling with air or water at the environment temperature.

Lowering the object's temperature below the environment temperature in this way is impossible; it would contradict the second law of thermodynamics.

It requires special equipment called refrigerating devices.

The tendency is to divide the cooling process into:

- cooling – the process of lowering the object's temperature to the environment temperature,
- chilling – lowering the object's temperature below the environment temperature.

Currently, the devices used for the second process are commonly called refrigerating or cooling devices.

Small refrigerating devices are called refrigerators (coolers), after the first devices that used ice for the cooling process.

Considering the construction, the refrigerators divide into:

- compression refrigerators,
- absorption refrigerators.

Absorption refrigerators divide into cogenerations and trigenerations with so-called inert gas.

Heat pump cycle can be used for other purposes than cooling a room or an object. It can be heating a room or an object using a circular factor at a high temperature.

A device, which takes the energy for heating from a source at lower temperature (lower source) and transfers it to a heated room (upper source) using the heat pump cycle, is called a **heat pump**.

A cooler, with the environment as the lower source, is such a device.

Compression refrigerators

Compression devices employ the **heat pump cycle**.

The circular factors, used in the heat pump cycles, are gases, which during the cycle do not change the state of aggregation and the factors that during the cycle go through phase changes.

A proper circular **cooling factor** presents the following **qualities**:

- low coagulation or solidification temperature (within the pressure range used in refrigerating devices),
- high evaporation temperature (less factor required),
- low specific volume of vapour (small compressor size),
- non-toxic,
- non-corrosive and
- cheap.

The mostly used **cooling factors** include:

- ammonia (NH_3),
- chloromethane (CH_2C1_2),
- sulphur dioxide (SO_2),
- carbon dioxide (CO_2),
- propane (C_3H_8),
- ethane (C_2H_6) and
- chlorifluoric derivatives of hydrocarbons, so-called freons.

Some industrial refrigerating devices usually use ammonia, despite it being toxic and highly corrosive for copper and its alloys, but it is cheap and the highest evaporation temperature of all. All the elements that come in contact with ammonia have to be made of steel.

Devices used for **intense cooling** are filled with **ethane or propane**.

Small house refrigerators and the ones in the public places - up until 1990s - were filled with freon R12 (contemporary replacement is R134a – hydrogen instead of chlorine).

Carbon dioxide presents very good cooling properties, but due to low heat transfer coefficient for vapours CO_2 condenser has to be big.

Principle of operation for a compression refrigerator

The principle of operation for a compression refrigerator is shown in **Fig. 17.1.** Compressed circular factor increases its pressure and temperature, reaching stage 1, and then it flows into a condenser.

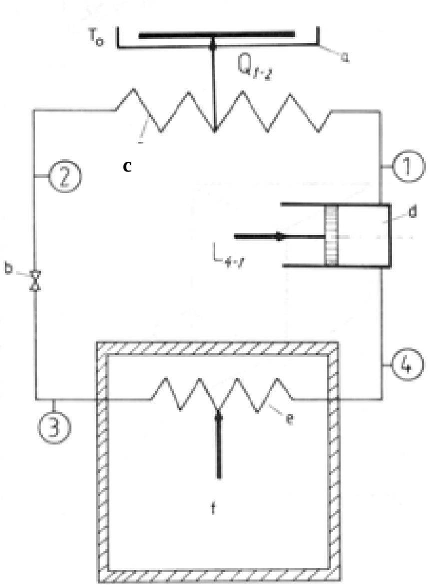


Figure. 17.1. Compression refrigerator system: A – cooling factor, b – throttle, c – condenser, d – compressor, e – evaporator, f – cooling chamber.

A **condenser** is a surface heat exchanger, inside which the vapour of the circular factor becomes condensed, due to cooling with water or air, through the dividing wall; smooth for water and ribbed for air.

Condensed circular factor in **stage 2** is expanded inside the **throttle** reaching stage 3. As a result of choking, the pressure of the circular factor drops and partially that factor evaporates. It leads to lowering the saturation temperature, which depends solely on pressure.

A **cold circular factor** flows into the **evaporator**, where, absorbing heat from the cooling chamber, it cools it down.

Evaporized circular factor reaches **stage 4** (dry saturated steam). This created vapour gets vacuumed by a compressor and compressed to stage 1. The cycle repeats itself.

A comparable circulation for the discussed cycle is the Linde cycle (Fig. **17.2 i 17.3)**, given, that

- The compression is adiabatic and isentropic,
- there is no pressure loss (choking) in the piping, in the condenser and evaporator,

- Choking occurs only inside a throttle.

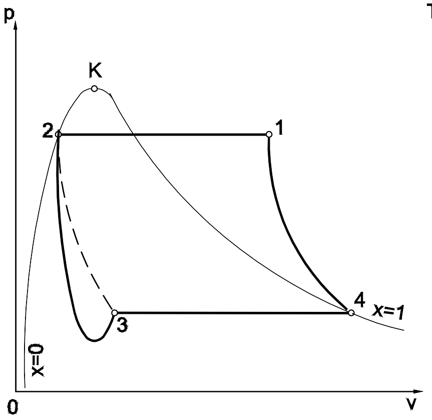


Figure. 17.2 The Linde cycle in the *p*-*v* diagram

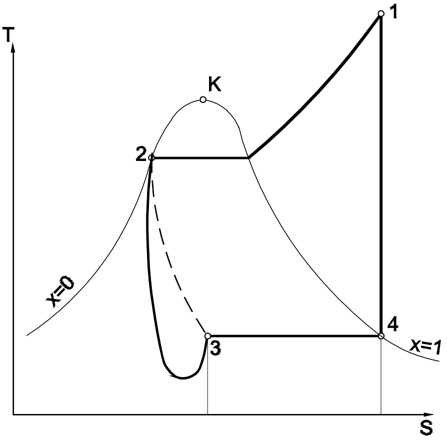


Figure. 17.3. Linde cycle in the T-S diagram.

The phase balance line: x = 0 – vapour with the dryness degree 0 (liquids); K – the critical point;

x=1 – dry saturated vapour (gas)

The **line 1-2** shows the **isobaric condensation of vapour**, forced into the condenser by a compressor.

Approximate flow of the process **2-3 (adiabatic throttle)** is the continuous line, the dashed line is the **isenthalpic** flow.

The **2 and 3** stages are on the same isenthalpy, which is typical of isenthalpic throttle.

The 3-4 process occurs in an evaporator – **isobaric evaporation** of moist and cold steam coming into the evaporator. When the process is completed, the vapour is sucked in and compressed in an **adiabatic** and frictionless way back to **stage 1**.

Cooling efficiency *ε* of a refrigerating cycle is a ratio of the heat introduced into the cycle Q^{+}_{ob}, so-called cooling effect (cooling productivity), to the circulation work with the minus sign (so that the cooling efficiency is a positive number):

$$\varepsilon = \frac{Q_{ob}^{+}}{-L_{ob}} \qquad (17.1)$$

For the Linde cycle it is:

$$\varepsilon_L = \frac{q_{3-4}}{-l_{4-1}} \qquad (17.2)$$

The individual heat of the isobaric process 3-4, according to the second form of I LT is:

$$q_{3\text{-}4} = i_4 - i_3 \qquad (17.3)$$

and for the isentropic process 4-1

$$l_{4\text{-}1} = i_4 - i_1 \qquad (17.4)$$

$$l_{4\text{-}1} = l_L$$

The process heat $\boldsymbol{q_{3-4}}$ and work $\boldsymbol{l_{4\text{-}1}}$ can be presented in the *T*-*s* diagram with corresponding areas (Fig. 17.4).

Substituting (17.3) and (17.4) into (17.2) results in:

$$\varepsilon_L = \frac{i_3 - i_4}{i_1 - i_4} \qquad (17.5)$$

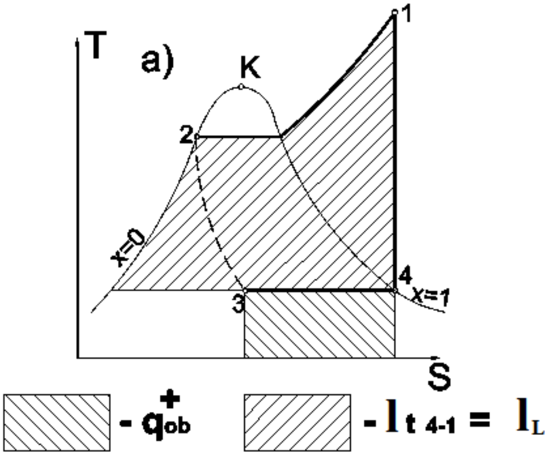


Fig. 17.4. Cooling effect and work in the T-s diagram

The efficiency of a real cooling **device is smaller** than the one calculated with the equation (17.5) because of the flow resistance inside the pipelines.

During the adiabatic compression there is some friction in the factor and the compressor mechanisms. Chocking the factor in the throttle is not adiabatic.

The cooling effect cannot be used up entirely due to piping insulation flaws at the stretch from the throttle to the cooling chamber and from the chamber to the compressor.

The actual efficiency ε_e equation is:

$$\varepsilon_e = \frac{Q'_e}{L'_e} \qquad (17.6)$$

where

Q'_e – useful cooling effect,
L'_e – effective compressor power.

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