Wrocław University of Technology

Refrigeration and Cryogenics

Bogusław Białko, Bartosz Zajączkowski

REFRIGERATION MACHINES, SYSTEM APPLICATIONS:
vapor compression refrigeration systems, heat pumps, air conditioning systems, absorption refrigeration

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Contents

Nomenclature

Q – heat, J

W – work, J

h – enthalpy, kJ/kgK

p – pressure, MPa

S – entropy, kJ/K

T – temperature, K

v – specific volume, m³/kg

ε – energetic efficiency of refrigerating,

η – efficiency, –

$\dot{m}$ – mass flow, kg/s

ξ – mass fraction, –

l – specific work, kJ/kg

c_p – specific heat, kJ/kgK

Indices

0 – evaporation

C – Carnot

k – condensation

L – Lorenz

m – average

HP – heat pump

R – refrigerating

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1 Vapor compression refrigeration

1.1 Introduction

1.1.1 Principle of operation

Refrigeration is the process of removing heat from one location (some closed compartment), and placing it in another location, where its presence is either required or makes no difference, i.e. the refrigerator has to pump up the heat up the temperature scale, from the temperature inside space to the temperature of environment (typically assumed between 20 and 30°C). However, as Clausius formulation of the Second Law of Thermodynamics states, the heat will not flow in the direction of increasing temperature. Therefore to achieve this, additional energy has to be introduced into the system and in a vapor compression system this means the work done by the compressor.

The process of pumping out heat from the refrigerator is similar to pumping water from a ground level up to the highest floor of a building. The only way to do this is to use a pump, which converts electric energy into usable mechanical work. As a result, the potential energy of water increases. This is also true in refrigeration systems. The compressor converts electricity into work, which is then used to increase refrigerant's internal energy.

1.1.2 Types of devices

Vapor compression refrigeration cycle is used in two main types of devices: refrigerators (including air conditioners) and heat pumps. Both devices are essentially the same, but they differ in their objectives. Refrigerators are designed to maintain low temperature in an enclosed space, by removing excess heat from it. Heat pumps are designed to maintain high temperature, using heat from an available low temperature source e.g. cold outside air in winter, well water or waste heat of some industrial process. This difference is schematically presented in Figure 1.1.
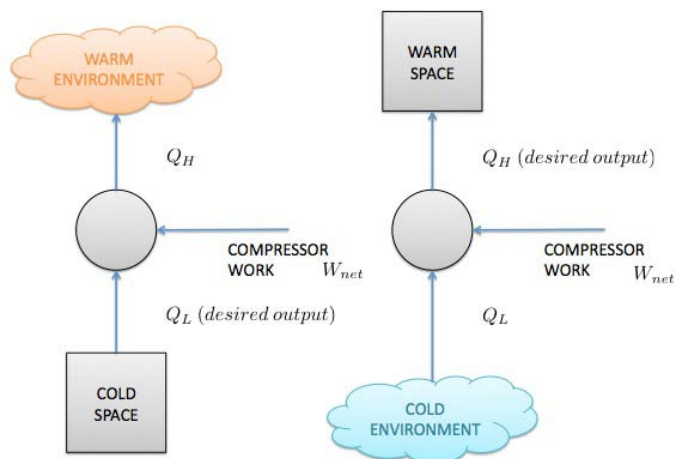
Figure 1.1. Operation principle of a refrigerator (left) and a heat pump (right)

1.1.3 Coefficient of Performance

The performance of a refrigerator or heat pump is expressed in terms of the Coefficient of Performance (COP). Typically, the COP is defined as a ratio between desired output and required input.

$$COP = \frac{desired\ output}{required\ input} \tag{1.1}$$

Required input means work done by the compressor, denoted as W_{net}. Desired output for a refrigerator differs from the heat pump. Therefore, two distinct Coefficients of Performance are defined, separately for each type of device.

$$COP_R = \frac{Q_L}{W_{net}} \tag{1.2}$$

$$COP_{HP} = \frac{Q_H}{W_{net}} \tag{1.3}$$

Where Q_L is the amount of heat absorbed in the evaporator at low pressure and Q_H is the amount of heat released in the condenser at high pressure. Further in this text, the condenser section is sometimes called the high-side, while the evaporator section is referred to as the low-side. Some authors, while discussing the performance of a heat pump, might use the name Coefficient of Amplification (COA) instead of COP_{HP} .

The values of both performance coefficients COP_R and COP_{HP} can be greater than 1. Comparison of equations (1.2) and (1.3) also reveals that:

$$COP_{HP} = COP_R + 1 \tag{1.4}$$

This means that in the absence of losses the efficiency of a heat pump is always equal to or greater than 1. If there is no energy from the low temperature source available, then the output will be obtained purely from compressor's work, i.e. all mechanical work done by the compressor will be converted into heat.

1.2 Vapor compression refrigeration cycle

1.2.1 Carnot cycle

The Carnot cycle is the theoretical base for all heat cycles. It was proposed by Nicolas Leonard Sadi Carnot (see Figure 1.2) in 1824 and expanded by Benoit Paul Emile Clapeyron in the 1830s and 40s.

The Carnot cycle defines the maximum possible efficiency for given temperature limits. It is totally reversible, which is a feature impossible to attain in actual cycles. Therefore it is called the ideal cycle and is used to compare the efficiency of other theoretical cycles. The best way to visualize the Carnot cycle is to use the temperature-entropy diagram (T-s) (see Figure 1.3).

The Carnot cycle consists of two isothermal and two isentropic conversions: a) 1-2 isentropic compression to higher temperature b) 2-3 isothermal heat addition at higher temperature c) 3-4 isentropic expansion to lower temperature d) 4-1 isothermal heat rejection at lower temperature.

Figure 1.2. Nicolas Leonard Sadi Carnot (1796 - 1832). French physicist and military engineer who, in his "Reflections on the Motive Power of Fire" (published 1824), gave the first successful theoretical account of heat engines, laying the foundations of the second law of thermodynamics. He is often described as the "Father of thermodynamics". Photograph: wikipedia.org

Figure 1.3. Carnot cycle on the T-s diagram

The efficiency of the Carnot cycle (executed between high temperature T_H and low temperature T_L) is calculated as follows:

$$\eta_{Carnot} = 1 - \frac{T_L}{T_H} \tag{1.5}$$

1.2.2 Reversed Carnot cycle

Reversing Carnot cycle means to change the direction of heat and work interactions. The result is a cycle operating counterclockwise. This reversed Carnot cycle is also called the Carnot refrigeration cycle. It is presented on the T-s diagram in Figure 1.4.

The Reversed Carnot cycle also consists of two isothermal and two isentropic conversions:
a) 1-2 isentropic compression to higher temperature
b) 2-3 isothermal heat rejection at higher temperature
c) 3-4 isentropic expansion to lower temperature
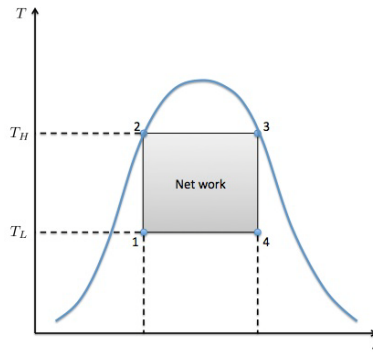d) 4-1 isothermal heat addition at lower temperature The Coefficient of Performance of Carnot refrigerator $COP_{R,Carnot}$, expressed in terms of limiting temperatures is as follows:

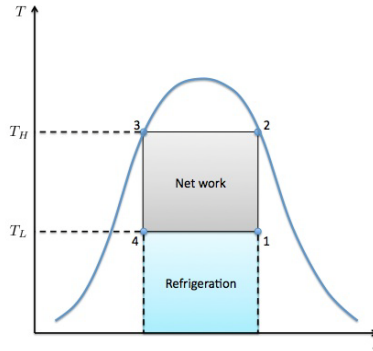$$COP_{R,Carnot} = \frac{1}{\dfrac{T_H}{T_L} - 1} \tag{1.6}$$

Figure 1.4. Reversed Carnot cycle on the T-s diagram.

Correspondingly for Carnot heat pump:

$$COP_{HP,Carnot} = \frac{1}{1 - \dfrac{T_L}{T_H}} \tag{1.7}$$

The reversed Carnot cycle is the most efficient refrigeration cycle. Unfortunately it is impossible to reach its perfection or even to build a device which will at least get close to it. It is fairly easy to approximate isothermal processes, since constant pressure in the saturation region fixes the temperature. However, isentropic compression and expansion are more difficult to attain, e.g. approximation of the isentropic compression would require a compressor capable of handling two-phase refrigerant. This can be solved by execution of the cycle outside the saturation region, but then maintaining of isothermal processes would become a complex task.

The conclusion is that the Carnot refrigeration cycle cannot be approximated in actual refrigeration devices. Thus a different ideal cycle has to be introduced.

1.2.3 The ideal vapor compression refrigeration cycle

Practical solution to the problems described in the previous subsection is to completely vaporize the refrigerant before compression, and to introduce a throttling device e.g. expansion valve or capillary tube. The result would be the ideal vapor compression refrigeration cycle, which is the most widely used cycle for refrigerators, air conditioning systems and heat pumps. Schematic representation of this cycle is shown in Figure 1.5.

The ideal vapor compression refrigeration cycle consists four following processes:
a) 1-2 isentropic compression to higher temperature
b) 2-3 isobaric heat rejection at higher temperature
c) 3-4 adiabatic expansion through a throttling to lower temperature
d) 4-1 isothermal heat addition at lower temperature

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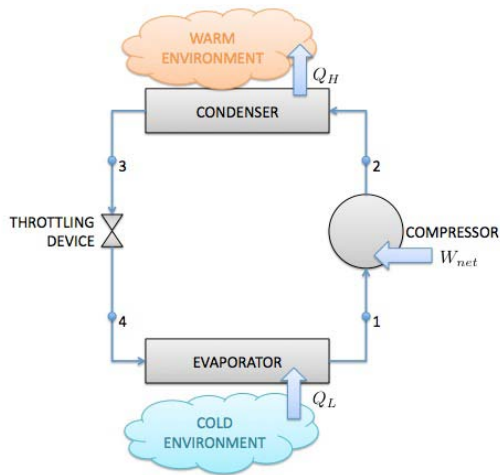
Figure 1.5. Scheme of a simple vapor compression cycle.

The refrigerant enters the compressor as saturated vapor (state 1) and is isentropically compressed (reaching superheated state 2). Then, it enters the condenser, liquefies while rejecting heat to the surroundings and leaves as saturated liquid (state 3). The saturated liquid expands adiabatically through a throttling device. Consequent to the expansion the evaporator pressure is attained and the temperature drops below the ambient (state 4). This enables isothermal heat addition at lower temperature, thus the refrigerant enters the evaporator as wet vapor and leaves it completely dry (state 1), completing the cycle. See Figure 1.6 for the visual representation of the ideal vapor compression refrigeration cycle on the T-s diagram.
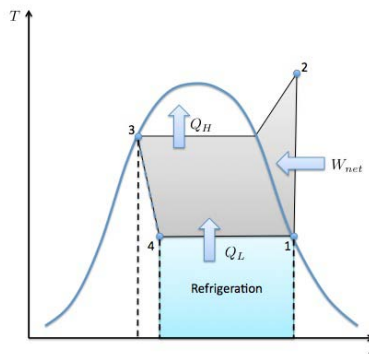
Figure 1.6. The ideal vapor compression refrigeration cycle on the T-s diagram.

The area under the curve 4-1 represents the heat absorbed by the refrigerant in the evaporator. The area under the curve 2-3 represents the heat released by the refrigerant in the condenser. In practice, during analysis of the cycle it is not convenient to compare areas, therefore another diagram is frequently used — the pressure-enthalpy diagram (P-h) — see Figure 1.7. The scale on the

pressure axis is usually logarithmic, because of the significant difference between pressures in the evaporator and in the condenser.

On P-h diagram, three out of four processes appear as straight lines (evaporation, condensation and throttling). Also, the lengths of the lines representing heat transfer in the evaporator (4-1) and the condenser (2-3) are proportional to actual transferred amount of heat and are expressed as an enthalpy differences.
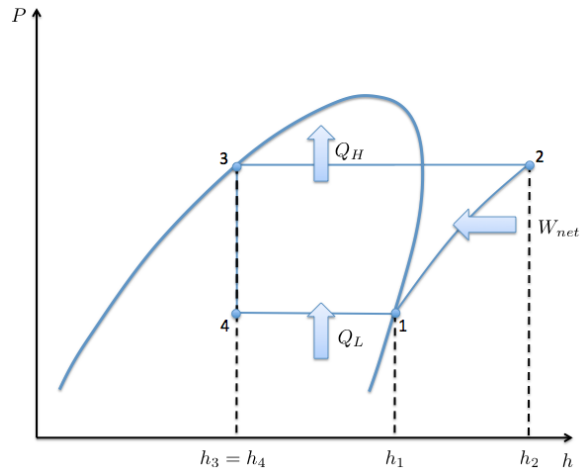
Figure 1.7. The ideal vapor compression refrigeration cycle on the P-h diagram.

The COP of devices operating on the ideal vapor compression refrigeration cycle is calculated using equation (1.2), thus:

$$COP_R = \frac{Q_L}{W_{net}} = \frac{\dot{m}_R \cdot \left(h_1 - h_4 \right)}{\dot{m}_R \cdot \left(h_2 - h_1 \right)}$$

(1.8)

Where m_R is the mass flow of refrigerant and h_{1-4} are the enthalpies at corresponding states (1, 2, 3 or 4) in Figure 1.7. The enthalpy values can be directly obtained from the P-h diagram. Equation (1.8) is valid for the ideal case, under assumption that changes of kinetic and potential energy of the refrigerant are negligible when compared to the heat transfer and the work input.

Proper operation of the vapor compression cycle requires the temperature in the evaporator to be lower than the required cooling temperature. Similarly, the temperature in the condenser has to be higher than the environment temperature. These limitations are presented on the T-s diagram in Figure 1.8.
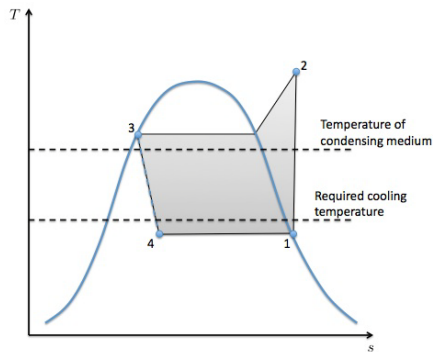
Figure 1.8. Temperature and pressure limitations in a vapor compression refrigeration cycle.

1.2.4   Practical vapor compression refrigeration cycle

Practical realization of the vapor compression refrigeration cycle differs from the ideal one because of irreversibility of the occurring processes, caused by the viscous friction and heat transfer to/from the surroundings. The schematic presentation of the practical vapor compression refrigeration cycle is presented in Figure 1.9 and on T-s diagram in Figure 1.10.
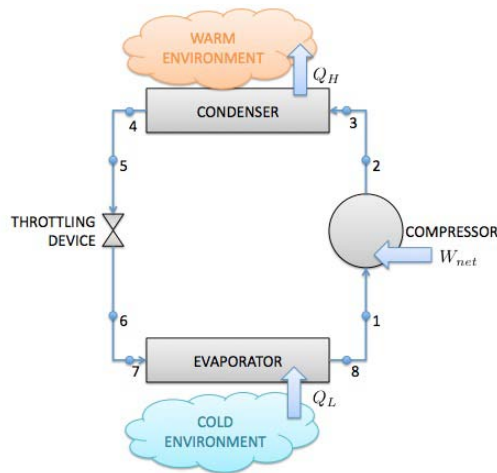
Figure 1.9. Scheme of a practical vapor compression cycle.

In reality, the most basic processes like vaporization and liquefaction aren't isothermal (because of losses). Moreover, they could occur not only in heat exchangers but also in tubing, e.g. evaporation in the line 6-7. To avoid undesired pressure drop between the throttling device and the evaporator, the line connecting them is typically very short. The line connecting the evaporator to the compressor 8-1 is usually long — the evaporator is installed at the top and of the refrigeration device, while the compressor at its bottom — which leads to a significant pressure drop and heat gain. In the ideal cycle, the compression process 1-2 is assumed isentropic. In the practical one,

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compression involves frictional effects and heat transfers which influence the entropy of the refrigerant, which either increase (1-2 in Figure 1.10) or decrease (1-2' in Figure 1.10).
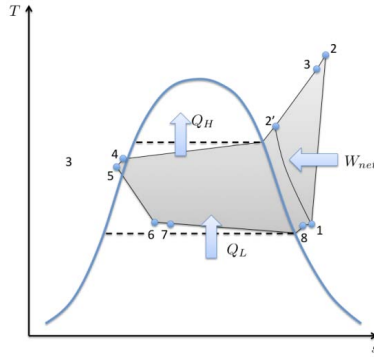
Figure 1.10. Practical vapor compression refrigeration cycle on the T-s diagram.

Accurate control over the state of refrigerant is also difficult. It is unlikely that upon entering the compressor the refrigerant will be precisely in the state of saturated vapor. Since the wet vapor would reduce compressor's efficiency (or even cause its malfunction), a typical refrigeration system is designed in a way, that vapor at the compressor's inlet is superheated. That is, it ensures that the refrigerant is completely vaporized. However, superheating, heat gain in the connecting line and pressure drops in both the evaporator and the connecting line contribute to an increase in refrigerant's specific volume, thus to an increase in required compressor's power input.

In the ideal case it is typically assumed, that the refrigerant leaves condenser as saturated liquid. In reality it is hardly possible to execute condensation so precisely. There is also a pressure drop in the condenser 3-4 and in the line connecting the condenser and the throttling valve 4-5.

Because of the described irreversibility, practical cycles have a lower COP than the reversed Carnot cycle and than the ideal vapor compression refrigeration cycle, when operating between the same temperature limits.

Vapor compression refrigeration system with an after cooler

It is undesirable to expand the refrigerant before it condensed completely. Typical practical solution to this problem is after cooling of the condensed liquid. An additional heat exchanger installed between the condenser and throttling device (see Figure 1.11) lowers the temperature of the liquefied refrigerant below its temperature of condensation 3-3'.
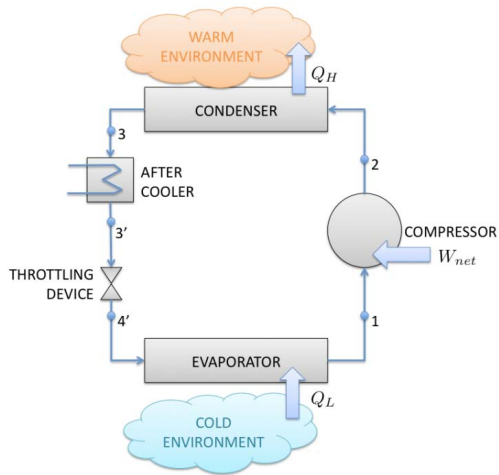
Figure 1.11. Scheme of a simple vapor compression cycle with an after cooler.

The after cooling offers an advantage of increased refrigeration capacity (see Figure 1.12).
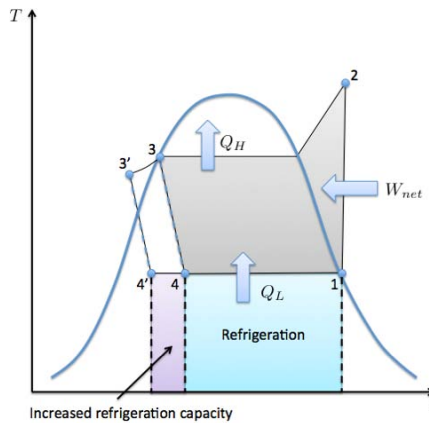
Figure 1.12. A simple vapor compression cycle with an after cooler on T-s diagram.

The Coefficient of Performance of a cycle with after cooling is calculated as follows:

$$COP_{R,AC} = \frac{Q_L + Q_{AC}}{W_{net}} = \frac{\dot{m}_R \cdot \left[\left(h_1 - h_4 \right) + \left(h_4 - h'_4 \right) \right]}{\dot{m}_R \cdot \left(h_2 - h_1 \right)} \tag{1.9}$$

Where Q_{AC} is the increased refrigeration capacity due to after cooling.

Vapor compression refrigeration system with recuperation

In cases where after cooling is not possible or not economical, a typical solution would be introduction of an internal heat exchanger - recuperator. Its purpose is to absorb heat from the liquefied refrigerant leaving the condenser (similar to the after cooler) and use it to superheat the vaporized refrigerant entering the compressor - see Figure 1.13 for the details.
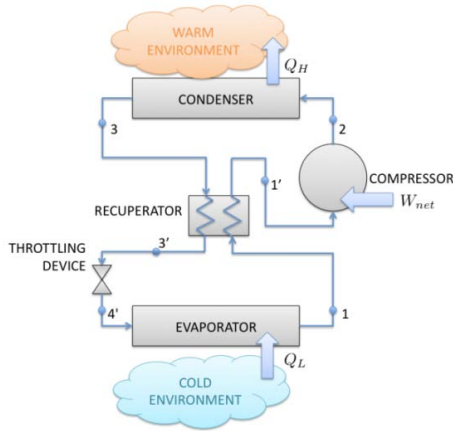
Figure 1.13. Scheme of a simple vapor compression cycle with a recuperator.

Liquefied refrigerant enters the evaporator with a lower enthalpy. Therefore, it is able to absorb greater amount of low temperature heat from the refrigerated space.
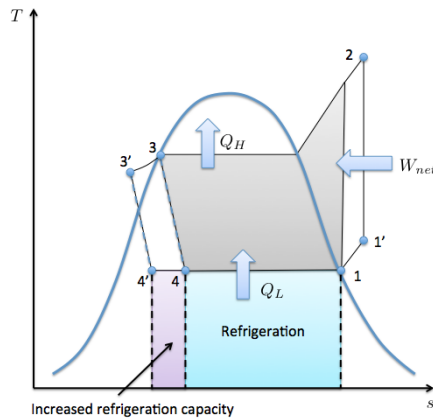
Figure 1.14. A simple vapor compression cycle with a recuperator on T-s diagram.

Close look at occurring processes gives the following list:
a) 1'-2 adiabatic compression to higher temperature
b) 2-3 isobaric heat rejection at higher temperature

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c) 3-3' sub-cooling of the liquefied refrigerant

d) 3'-4' isenthalpic expansion through a throttling to lower temperature

e) 4'-1 isothermal heat addition at lower temperature

f) 1-1' superheating of the refrigerant with the heat from 3-3'

Recuperation increases cooling capacity in a similar way as the after cooling. However, superheating 1-1' leads to increased work requirements. In consequence, internal heat transfer will not always increase efficiency. The Coefficient Of Performance of a system with recuperation is:

$$COP_{R,REC} = \frac{Q_L + Q_{AC}}{W_{net} + W_{net,REC}}$$

(1.10)

Where $W_{net,REC}$ is the increase in required work input. The value of $COP_{R,REC}$ will increase only if:

$$\frac{Q_{AC}}{Q_L} > \frac{W_{net,REC}}{W_{net}}$$

(1.11)

This condition is not obvious and it depends on the operating temperatures, the refrigerant used, etc.

1.3 Multi-pressure vapor compression refrigeration systems

The simple vapor compression refrigeration system discussed in the previous paragraph remains the most widely used and it is suitable for most of the applications. However, some applications (e.g. in industry) require temperature/pressure operating range, which may be too large for a single vapor compression refrigeration cycle. In such cases, the most regarded advantages of the basic system such as simplicity, reliability and low cost are no longer considered as the most important factors. For large industrial applications the major concern is the efficiency of the refrigeration unit, not its simplicity. One of the possible solutions to such situations is to perform the refrigeration process in stages, i.e. to use a multi-pressure refrigeration system.

A multi-pressure system is a vapor compression refrigeration device that has two or more low-side pressures. Typically we encounter:

a) cascade systems with a number of isolated refrigeration cycles operating in series;

b) compound systems operating a single loop with several compressors;

c) compound systems operating a single loop with several evaporators.

The most complicated applications may require combinations of the above solutions.

1.3.1 Cascade vapor compression refrigeration system

Cascade refrigeration system is a multi-pressure system, which employs more than one refrigerant circuit operating a separate two-pressure loop (see Figure 1.15). Each circuit could use a different refrigerant, e.g. R-22 in loop A and R-13 in loop B.
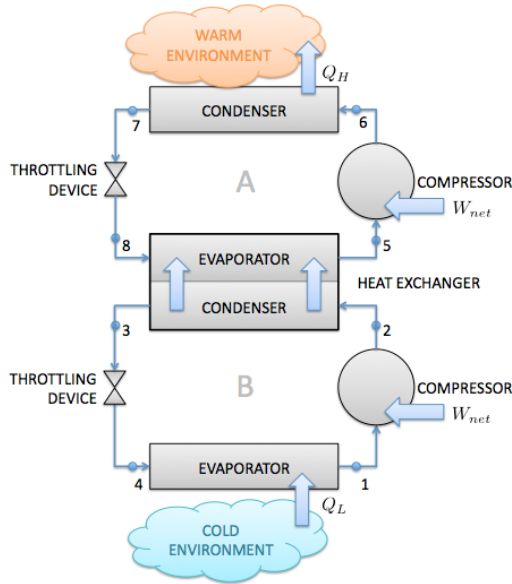
Figure 1.15. Scheme of a two-stage cascade vapor compression refrigeration system.

In the cascade, one of the cycles (A in Figure 1.15) operates at the top of the other one (B in Figure 1.15). The loops are connected through the heat exchanger.

It allows heat transfer from the condensing refrigerant B to the evaporating refrigerant A. Assuming that kinetic and potential energies of the refrigerants are negligible and that there are no heat losses to the environment (i.e. the heat is ex- changed only between the refrigerants), the energy balance of the heat exchanger can be written as:

$$\dot{m}_A \cdot \left(h_5 - h_8 \right) = \dot{m}_B \cdot \left(h_2 - h_3 \right) \tag{1.12}$$

Where m_A and m_B are the mass flows, and h is the enthalpy at given state.

The Coefficient of Performance of the cascade system has to take the work input of additional compressor(s) under consideration. Therefore it is calculated as:

$$COP_{R,cascade} = \frac{Q_L}{W_{net}} = \frac{\dot{m}_A \cdot \left(h_1 - h_4 \right)}{\dot{m}_B \cdot \left(h_2 - h_1 \right) + \dot{m}_A \cdot \left(h_6 - h_5 \right)} \tag{1.13}$$

The T-s diagram in Figure 1.16 shows that cascading decreased the required work input and increased the amount of heat absorbed from the refrigerating space. Thus, the efficiency of a cascade system is higher than the efficiency of a single stage system operating within the same temperature/pressure range. This is the reason of using three or four steps cascades in large industrial refrigeration units.
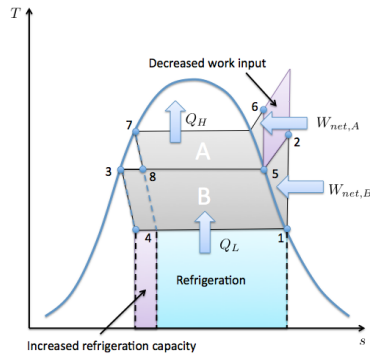
Figure 1.16. Cascade vapor compression refrigeration system on the T-s diagram.

1.3.2 Multi-pressure vapor compression refrigeration system with a flash chamber

A multi-pressure vapor compression refrigeration system with a flash chamber is also an example of a device introducing additional compressors due to the large pressure difference between the low-side and the high-side. However, the fluid used throughout the cascade is the same, thus the heat exchanger is in this case replaced by a flash chamber - see Figure 1.17.
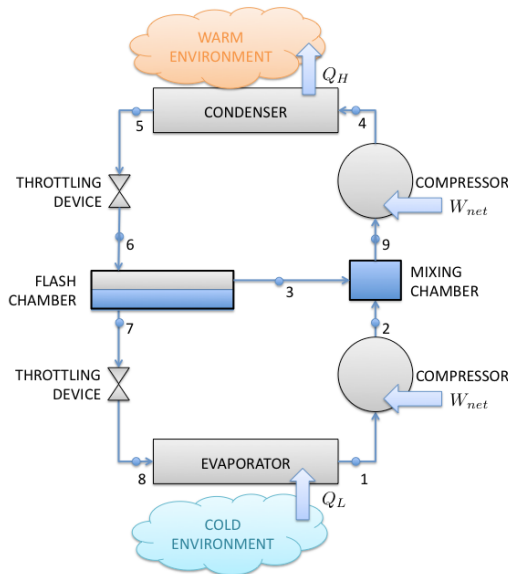
Figure 1.17. Scheme of a multi-pressure vapor compression refrigeration system with a flash chamber.

The pressure inside the flash chamber is the same as the interstage pressure in the compressor. Liquefied refrigerant expands to the flash chamber 5-6, where a new equilibrium is formed. The saturated liquid 7 is then expanded in the second throttling device 7-8. Some of the liquid evaporates

and this saturated vapor 3 is allowed to freely mix with the vapor from the low-pressure compressor 2, giving the state 9.

In order to calculate the Coefficient of Performance of the multi-pressure system with a flash chamber, it is necessary to determine the liquid fraction of the refrigerant in the chamber (at state 6). This can be calculated as follows:

$$x_6 = \frac{(h_6 - h_7)}{(h_3 - h_7)} \tag{1.14}$$

Where h_3 and h_7 are correspondingly the enthalpy of the saturated vapor and the enthalpy of the saturated liquid, at the flash chamber pressure.

The amount of heat removed from the refrigerated space is calculated as:

$$q_L = (1 - x_6)(h_1 - h_8) \tag{1.15}$$

The work input of two compressors:

$$w_{net} = w_{net,1} + w_{net,2} = (1 - x_6)(h_2 - h_1) + (h_4 - h_9) \tag{1.16}$$

The Coefficient of Performance can be calculated from equation (1.2). Substitution of equations (1.15) and (1.16) into (1.2) gives:

$$COP_{R, flash-chamber} = \frac{q_L}{w_{net}} = \frac{(1 - x_6)(h_1 - h_8)}{(1 - x_6)(h_2 - h_1) + (h_4 - h_9)} \tag{1.17}$$
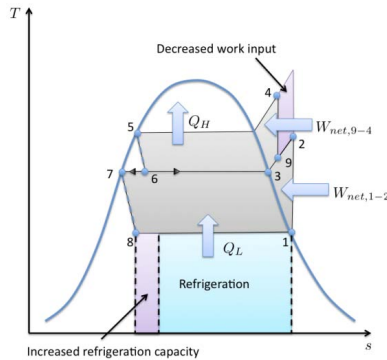
Figure 1.18. A multi-pressure vapor compression refrigeration system with a flash chamber on the T-s diagram.

To solve the above equation one has to calculate the work input of the high-pressure compressor. Thus, one has to determine h_9 - the enthalpy at state 9 and h_4 - the enthalpy at state 4. Obtaining this value requires writing down the energy balance of the mixing chamber (see Figure 1.17), which leads to equation:

$$h_9 = x_6 \cdot h_3 + (1 - x_6) \cdot h_2 \tag{1.18}$$

The enthalpy at point 4 can be found under assumption that the compression is isentropic, i.e. the entropy at state 4 is equal to the entropy at state 9.

1.3.3 Multi-purpose vapor compression refrigeration system

In some applications the system is required to operate at a few different evaporation temperature levels. This can be done with the previously discussed cascade system, however in is very ineffective in small units, e.g. domestic refrigerators with a separated freezer and cooling chamber. In such cases, so called multi-purpose vapor compression refrigeration systems are implemented. Typically such a device consists only one compressor and a number of evaporators (usually two) separated with additional throttling device(s). A schematic representation of such a system is shown in Figure 1.19.
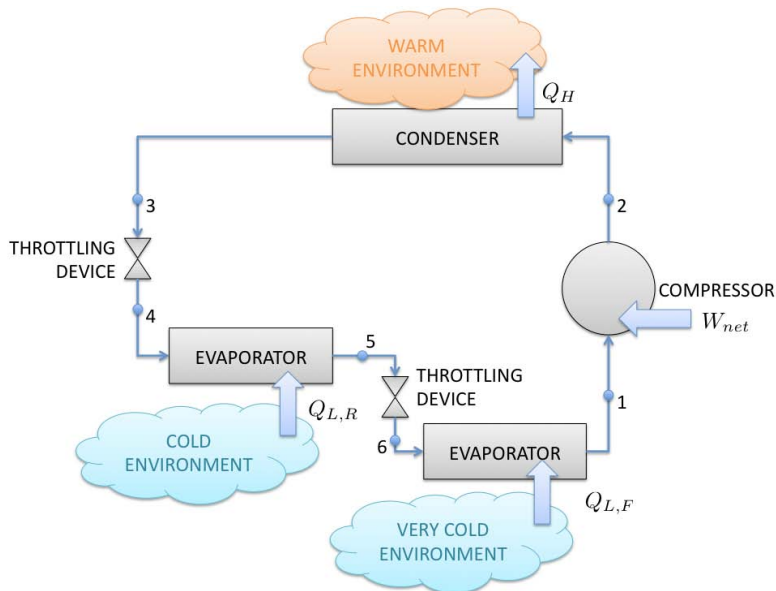
Figure 1.19. Scheme of a multi-purpose vapor compression refrigeration system with two evaporators.

The most common example of a multi-purpose vapor compression refrigeration system is an ordinary refrigerator-freezer unit. Most of food have a high water content, thus is has to be stored in the temperature just above 0°C to prevent dehydration and freezing. On the other hand, frozen goods are kept at much lower temperature, typically about -18°C. A single loop device with one evaporator and one throttling device is not able to operate at both levels. The cheapest and the simplest solution to this problem is splitting the evaporator in two and introduction of a second throttling valve.

2 Absorption refrigeration

2.1 Introduction

2.1.1 What is absorption refrigeration?

Absorption refrigeration is another method of refrigeration, which instead of mechanical work of the compressor (as described in the previous chapter) utilizes locally available heat sources. Absorption is a chemical process in which molecules of the refrigerant enter a bulk phase of a transport medium.

Absorption is not to be confused with adsorption, which means binding of refrigerant's molecules on the surface of a highly porous solid medium (adsorbent) — not within its volume.

The absorption refrigeration systems are much more complex than vapor compression systems, thus they occupy more space and are more expensive. What is the most important they are much less efficient. Absorption is economically attractive only if there is a source of inexpensive thermal energy available — the unit cost of thermal energy is low relative to electricity (and is predicted to remain low in the future).

The heat source could be either natural or artificial. The natural energy sources include renewables such as solar energy or geothermal energy. The artificial heat source is typically waste heat of some industrial process (e.g. in power plants or production facilities) or exhaust gases from the engines. The most attractive sources are at the temperature $100 - 200^{\circ}$C.

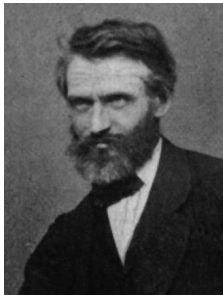### 2.1.2 The invention of absorption refrigeration system

Figure 2.1. Ferdinand Philippe Edouard Carr´e (1824-1900). French engineer who developed and patented the first ammonia-water absorption refrigeration system. Photograph: wikipedia.org

French scientist Edmond Carre invented absorption cooling in 1850. His prototype was using water and sulphuric acid. Older brother of Edmond — Ferdinand Carre (see Figure 2.1) joined the effort, and in 1858 developed a more advanced system using water as the absorbent and ammonia as the refrigerant. In following years, Ferdinand patented his newly invented machine: in 1859 in France and year later in the United States. During the Universal London Exhibition in 1862 he demonstrated a working unit capable of producing about 200kg of ice per hour. Continuing his work, in 1876 Carre installed an absorption refrigeration system on a food carrier (transport ship named "Paraguay"), making it capable of carrying frozen meat on intercontinental routes. Carre's refrigeration system remained popular through the early 1900s, when cheaper and more efficient vapor compression refrigeration devices replaced it.

2.2 Principle of operation

The most widely used absorption refrigeration device is a modern version of Carre's ammonia-water system ($NH_3 - H_2O$), where ammonia serves as the refrigerant and water as the transport medium.

The condenser and evaporator in Figure 2.2 (on left hand side) correspond to those in vapor compression refrigeration systems (e.g. in Figure 1.5). The evaporated

refrigerant is drawn off from the evaporator not by the compressor, but by the absorber. In the absorber, weak solution (of less than 25% of ammonia) sustains pressure slightly lower than the equilibrium pressure, therefore it is able to absorb incoming vapor. Resulting rich solution is then pumped (with mechanical pump) to a generator, where heat from a high temperature source is submitted. Vaporized rich solution from the generator is directed to a rectifier, where the pure ammonia is collected and sent to the condenser, while remaining weakened solution returns to the generator and further, through an expansion valve, back to the absorber.

Right hand side elements of the absorption refrigeration system (in Figure 2.2) — absorber, mechanical pump and generator and distillation (rectification) column — perform exactly the same functions as the compressor. For this reason, these elements are sometimes called together as a thermic compressor.
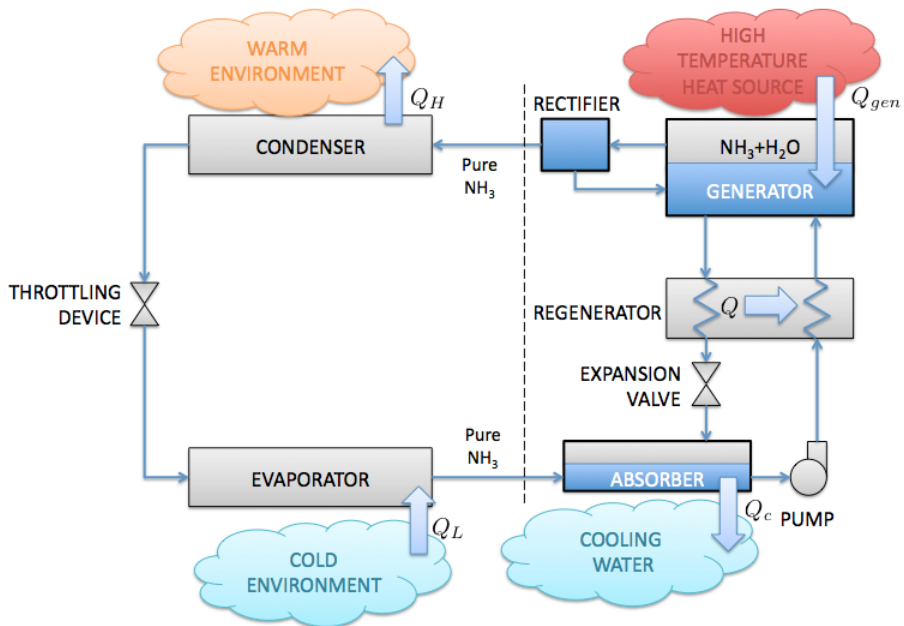
Figure 2.2. Scheme of an ammonia-water absorption refrigeration system.

The mechanical pump is not a great problem in a large industrial installation, but excludes the use of Carre in e.g. household refrigerators.

The idea of eliminating the pump closing the pressure gap between the condenser and the evaporator by introducing inert gas in the latter first occurred to Geppert in 1899. Now, simply introducing inert gas in the low pressure part of the Carre's machine, would not work: With inert gas in the way, the vapor would have to pass to the absorber by diffusion, which is not possible, because diffusion and heat conductance in gases work only over short distances like a few millimeters. Therefore, the solution was to create a closed loop for the gas between the evaporator and absorber. But this did not resolve the problem of circulating of the gas.

This was resolved in an elegant way by Platen and Munters. They proposed that the unit is charged with water, ammonia and hydrogen. There are no pressure differences in the unit except those caused by liquid columns. The condenser is situated above the evaporator, so the ammonia condensate flows by gravity from one to the other. The difference in height is not due to any flow resistance but to the fact that the liquid in the left leg of the tube is colder and thus denser. The liquid ammonia enters the evaporator tube and flows in a small stream at its bottom. The tube contains weak gas (hydrogen with, say 5% by volume of ammonia vapor) and the liquid evaporates into the gas. By this, the density of the gas mixture increases (ammonia vapor is 8.5 times heavier than hydrogen) and tends to flow downwards to the absorber coil, situated further down. Weak solution enters at the top of the absorber coil and flows downwards at the bottom of the tube. The solution absorbs ammonia vapor from the gas, which gets lighter and tends to flow upwards to the evaporator. Here we have the gas loop between the evaporator and the absorber and a natural circulation of the gas due to density differences.

2.2.1 Coefficient of Performance

When compared with a vapor compression refrigeration device, an absorption system has one major advantage — liquid is compressed instead of vapor. In a steady state flow, the pump work is proportional to the specific volume. As a result, the work input of the absorption refrigeration is tiny when compared to the heat supplied to the generator (about one percent)

2.3 Binary mixtures

2.3.1 Properties of refrigerant-absorbent mixtures

The solution used in absorption refrigeration systems may be considered as a homogeneous binary mixture of refrigerant and absorbent. Depending upon the boiling point difference between refrigerant and absorbent and the operating temperatures, one may encounter a pure refrigerant vapor or a mixture of refrigerant and absorbent vapor in the generator of the absorption system.

Unlike pure substances, the thermodynamic state of a binary mixture (in liquid or vapor phase) cannot be fixed by pressure and temperature alone. According to Gibbs' phase rule, one more parameter in addition to temperature and pressure is required to completely fix the thermodynamic state. Generally, the composition of the mixture is taken as the third independent parameter. The composition of a mixture can be expressed either in mass fraction or in mole fraction. The mass fraction of components 1 and 2 in a binary mixture are given by:

$$\xi_1 = \frac{m_1}{m_1 + m_2} \tag{2.1}$$

$$\xi_2 = \frac{m_2}{m_1 + m_2} \tag{2.2}$$

where m_1 and m_2 are the mass of components 1 and 2, respectively. The mole fraction of components 1 and 2 in a binary mixture are given by:

$$x_1 = \frac{n_1}{n_1 + n_2} \tag{2.3}$$

$$x_2 = \frac{n_2}{n_1 + n_2} \qquad (2.4)$$

where n_1 and n_2 is the number of moles of component 1 and 2 respectively.

An important property of a mixture is its miscibility. A mixture is said to be completely miscible if a homogeneous mixture can be formed through any arbitrary range of concentration values. The temperature at which the mixtures are mixed influences their miscibility. Some mixtures are miscible under certain conditions and immiscible at other conditions. The refrigerant-absorbent mixtures used in absorption refrigeration systems must be completely miscible under all conditions, both in liquid and vapor phases.

2.3.2 Ideal homogeneous binary mixtures

A binary mixture of components 1 and 2 is called as an ideal mixture, when it satisfies the following conditions:

a) Condition 1: The volume of the mixture is equal to the sum of the volumes of its constituents, i.e. upon mixing there is neither contraction nor expansion. Thus the specific volume of the mixture v is given by:

$$V = \xi_1 \cdot v_1 + \xi_2 \cdot v_2 \qquad (2.5)$$

where ξ_1 and ξ_2 are the mass fractions of components 1 and 2. For a binary

Mixture, ξ_1 and ξ_2 are related by:

$$\xi_1 + \xi_2 = 1 \Rightarrow \xi_2 = 1 - \xi_1 \qquad (2.6)$$

b) Condition 2: Neither heat is generated nor absorbed upon mixing, i.e., the heat of solution is zero. Then the specific enthalpy of the mixture, h is given by:

$$h = \xi_1 \cdot h_1 + \xi_2 \cdot h_2 = \xi_1 \cdot h_1 + \left(1 - \xi_1\right) \cdot h_2 \qquad (2.7)$$

c) Condition 3: The mixture obeys Raoult's law in liquid phase, i.e., the vapour pressure exerted by components 1 and 2 ($P_{v,1}$ and $P_{v,2}$) at temperature T are given by:

$$P_{V,1} = x_1 \cdot P_{1,sat} \qquad (2.8)$$

$$P_{V,2} = x_2 \cdot P_{2,sat} \qquad (2.9)$$

where x_1 and x_2 are the mole fractions of components 1 and 2 in solution, and $P_{1,sat}$ and $P_{2,sat}$ are the saturation pressures of pure components 1 and 2 at temperature T. The mole fractions x_1 and x_2 are related by:

$$x_1 + x_2 = 1 \Rightarrow x_2 = 1 - x_1 \qquad (2.10)$$

d) Condition 4: The mixture obeys Dalton's law in vapor phase; i.e. the vapor pressure exerted by components 1 and 2 ($P_{v,1}$ and $P_{v,2}$) in vapor phase at a temperature T are given by:

$$P_{V,1} = y_1 \cdot P_{total} \tag{2.11}$$

$$P_{V,2} = y_2 \cdot P_{total} \tag{2.12}$$

where y_1 and y_2 are the vapor phase mole fractions of components 1 and 2 and P_{total} is the total pressure exerted at temperature T. The vapor phase mole fractions y_1 and y_2 are related by:

$$y_1 + y_2 = 1 \Rightarrow y_2 = 1 - y_1 \tag{2.13}$$

and the total pressure P_{total} is given by:

$$P_{V,1} + P_{V,2} = P_{total} \tag{2.14}$$

If one of the components, say component 2 is non-volatile compared to component 1 (e.g. component 1 is water and component 2 is lithium bromide salt), then $y_1 \simeq 1$ and $y_2 \simeq 0$, $P_{v,2} \simeq 0$, then from Raoult's and to Dalton's laws:

$$P_{total} = P_{V,1} = x_1 \cdot P_{1,sat} \tag{2.15}$$

2.3.3 Real mixtures

Real mixtures deviate from ideal mixtures since:

1) A real solution either contracts or expands upon mixing, i.e.,

$$V = \xi_1 \cdot v_1 + \xi_2 \cdot v_2 \tag{2.16}$$

2) Either heat is evolved (exothermic) or heat is absorbed upon mixing;

$$h = \xi_1 \cdot h_1 + \left(1 - \xi_1\right) \cdot h_2 + \Delta h_{mix} \tag{2.17}$$

where Δh_{mix} is the heat of mixing, which is taken as negative when heat is evolved and positive when heat is absorbed.

The above two differences between ideal and real mixtures can be attributed to the deviation of real mixtures from Raoult's law. Real mixtures approach ideal mixtures as the mole fraction of the component contributing to vapor pressure approaches unity, e.g. for very dilute solutions.

Figure 2.3 shows the equilibrium pressure variation with liquid phase mole fraction (x) of ideal and real binary mixtures with positive (+v_e) and negative deviations (–v_e) from Raoult's law at a constant temperature. It can be seen that when the deviation from Raoult's law is positive (+v_e), the equilibrium vapor pressure will be higher than that predicted by Raoult's law, consequently at a given pressure and composition, the equilibrium temperature of solution will be lower than that predicted by Raoult's law. The converse is true for solutions with $_e$ deviation from Raoult's law, i.e., the equilibrium temperature at a given pressure and composition will be higher than that predicted by Raoult's law for solution with negative deviation.
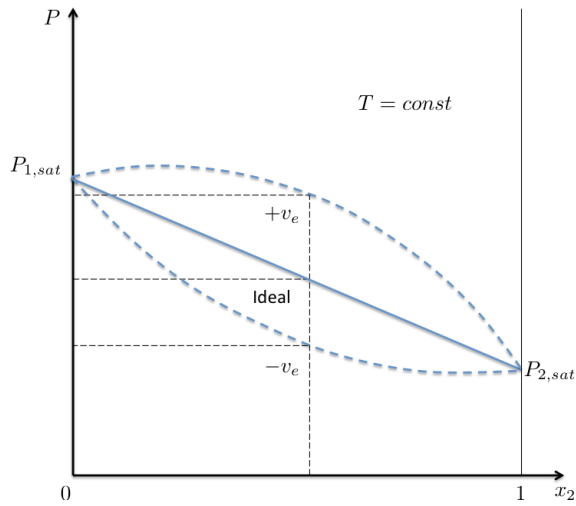
Figure 2.3. Pressure-concentration behavior of ideal and real mixtures at a constant temperature.

This behavior can also be shown on a specific enthalpy-composition diagram as shown in Figure 2.4 for a solution with negative deviation from Raoult's law. Refrigerant-absorbent mixtures used in vapor absorption refrigeration systems exhibit a negative deviation from Raoult's law, i.e., the process of absorption is exothermic with a negative heat of mixing.
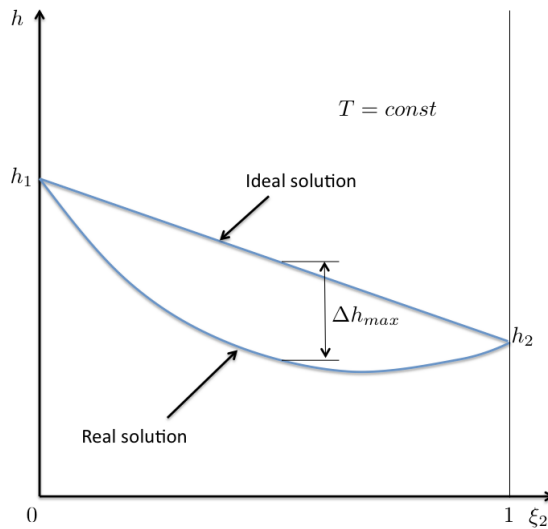

Figure 2.4. Enthalpy-concentration behavior of an ideal mixture and a real mixture with negative deviation from Raoult's law.

2.4   h-x diagram of ammonia-water binary mixture

Every refrigeration cycle requires a tool on which characteristic points of state, as well as thermodynamic and phase changes can be represented. Another diagram (h x) is used for binary mixtures absorption refrigeration cycles. In fact, it is a 3D lgP– h – x diagram, but in the past it was impossible to present it in an easy way in 3D. That is why it was prepared with reduction of one axis of coordinates.

The h– x diagram is a basic tool for analyzing thermodynamic processes in absorption refrigeration. It is analogous to the logP– h diagram in compressor refrig eration. To understand the construction of the h - x diagram and connection with the logP h diagram, it will be easier to construct a 3D logP –h–x diagram. The axes on this diagram as presented in Figure 2.5 are:
• logP — pressure in logarithmic scale (upwards),
• h — enthalpy (to the left),
• x — concentration of NH_3 in NH_3-H_2O mixture (to the right).

2.4.1   Identification with Absorption 3D

Further in the text, several characteristic lines visible on 3D logP – h – x diagram will be presented in detail. The list is as follows:
• saturation lines,
• isobars,
• isotherms,

2.4.2   Saturation lines

There are saturation lines on the diagram for a full range of useful pressures for absorption cycles and for pure components of concentration of NH_3 x = 0 and x = 1.

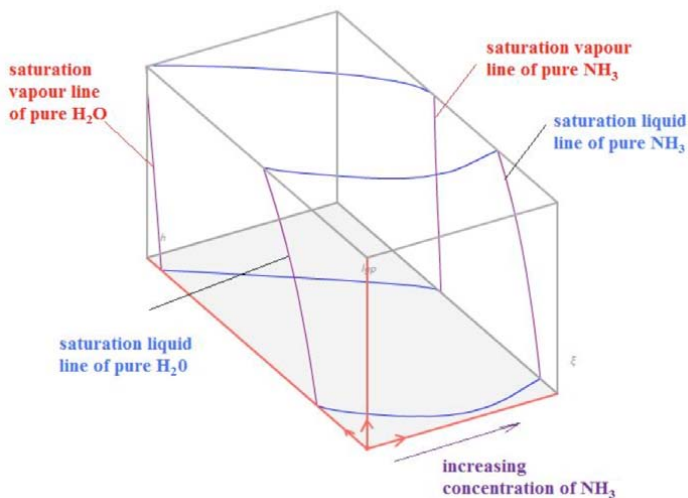Figure 2.5. Saturated liquid and saturated vapor lines on 3D logP – h – x diagram.

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After rotation of the diagram, it is possible to observe 2D logP h diagram. Presented drawings show how to obtain logP −h diagrams from the h−ξ diagram, for pure components of the mixture.
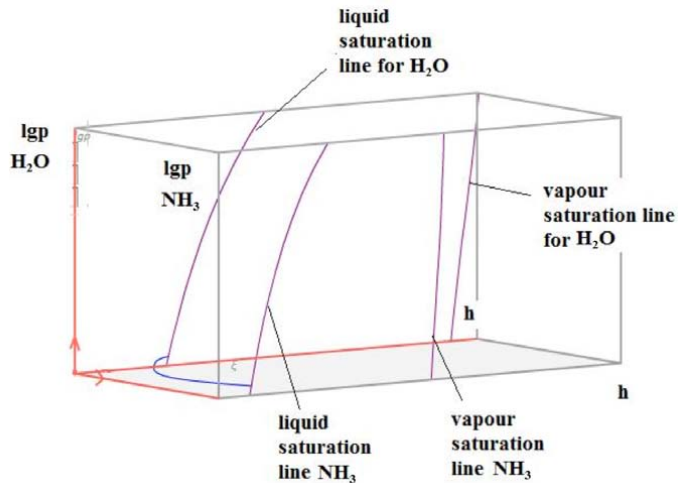

Figure 2.6. Side view of rotated logp – h – x diagram showing familiar 2D shape of saturation lines.

If a concentration $0 < x < 1$ will be selected, it is possible to obtain an individual logp- h diagram for any ratio of mixed components.

2.4.3 Identification on a classical diagram

Let's consider the problem. Assuming that the pressure of saturated NH_3 - H_2O binary mixture is 0.4bar. The temperature of the mixture is 500°C. Is the solution of saturated liquid and saturated vapor needed?

Step 1 Find the isotherm and isobar intersection on the liquid saturation line (see Figure 2.7).

Step 2 On the intersection of saturated liquid isobar 4ata and isotherm 500°C, it is possible to get a solution of the liquid mixture x_c. It is $x_c = 0.394$. Then, thanks to the lines of constant solution of opposite phase $x_p = 0.984$ (see Figure 2.7).

Step 3 Find enthalpy of the saturated liquid (see Figure 2.7).

Step 4 Find the intersection of the line of constant solution of saturated liquid with the support line of constant pressure 4ata.

Step 5 Draw a line from the found intersection with the support line of constant pressure 4ata to the intersection with the line of constant vapor pressure 4ata.

Step 6 Extend the line from the found intersection to the intersection with axis h and x and determine the enthalpy of the saturated vapor and the solution of the saturated vapor.

Step 7 Draw an isotherm line of the equilibrium from the found point of the saturated liquid to the point of the saturated vapor.

Step 8 As it is impossible to precisely determine the point of intersection of the line of the equilibrium with the line of constant concentration on the saturated vapor pressure contour, the lines of opposite phase are drawn in the liquid area, which helps achieve a precise result.
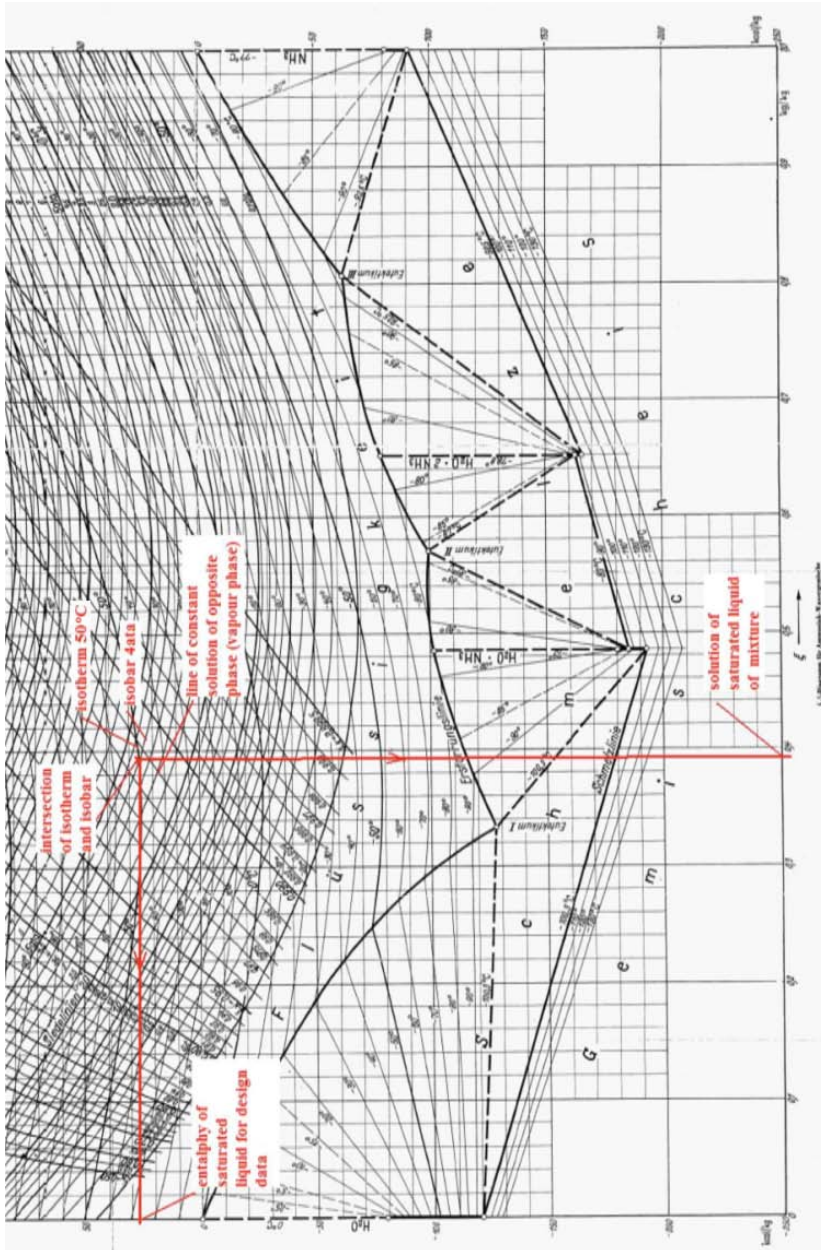
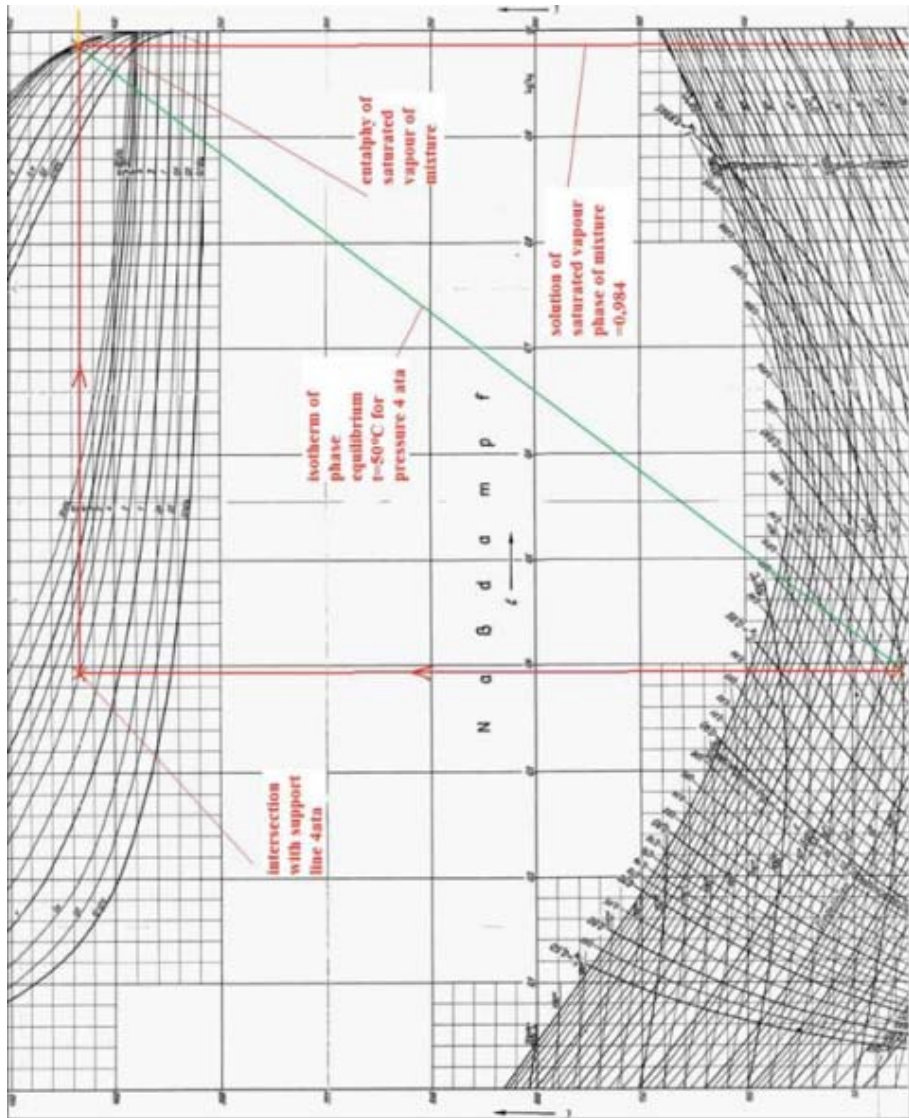Figure 2.7. Identification on a classical diagram – steps 1-3.

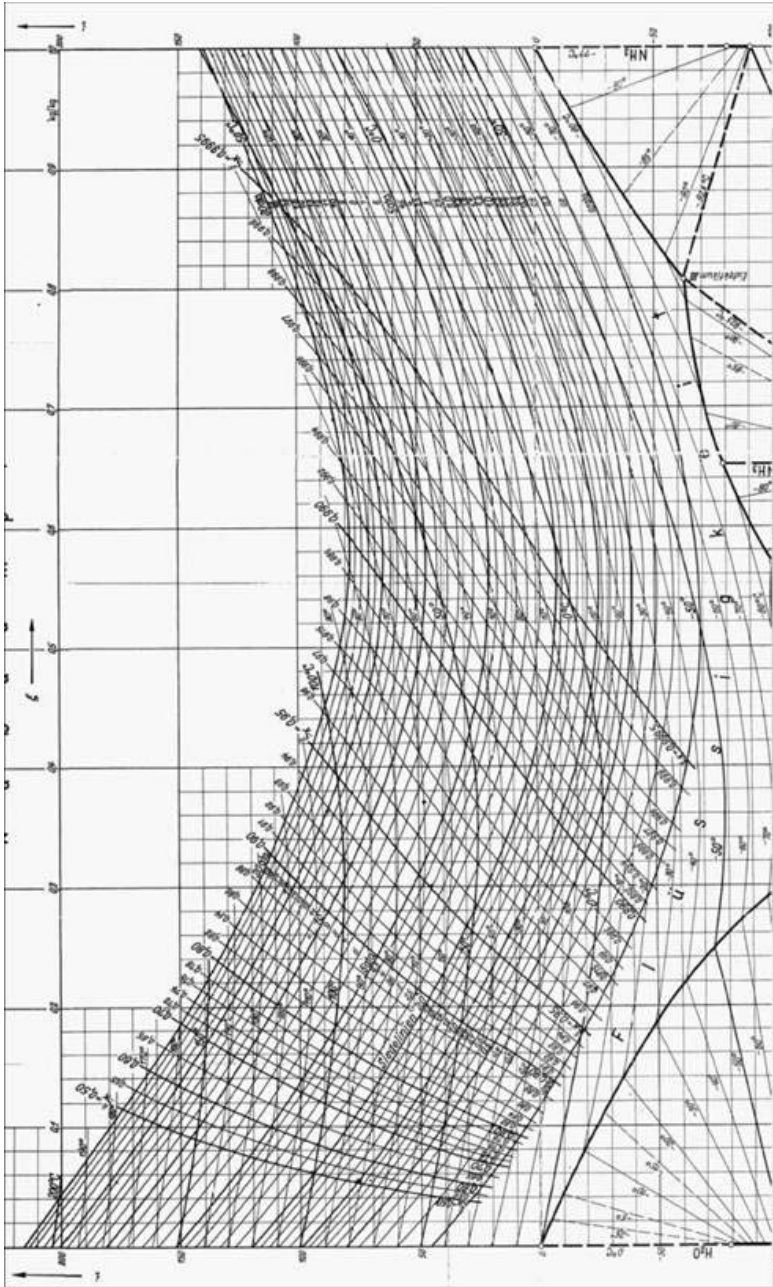Figure 2.8. Identification on a classical diagram – steps 4-8.

Figure 2.9. Parameters of the subcooled liquid on classical h - x diagram for NH_3 – H_2O binary mixture.

Figure 2.10. h – x diagram for NH$_3$ - H$_2$O binary mixture.

2.5   Applications and perspectives

Absorption refrigeration systems played a significant part in the entire history of all thermodynamics and refrigeration technology. They were replaced by compressor refrigeration systems because of the evolution of electrification and for many years were removed from the top of refrigeration technology.

Absorption systems needed much more knowledge and experience in designing and service than compressor systems, because in one absorption plant there are in fact two systems — one is realizing the cooling effect and the other is responsible for driving the force of the entire system. Every failure of working conditions of one of this systems caused wrong operation of the entire installation. Besides, to achieve an acceptable COP, a stable and high temperature source of heat is required.

In countries where consumption of electric energy was huge and climate conditions forced to apply large air conditioning plants, absorption refrigeration systems progressed. An absorption cycle was the only alternative to satisfy the needs of the cooling effect. In the last 25 years in Asian countries, absorption refrigeration grows and now more than 90% of absorption machines are produced and located there.

Parallel to absorption machines, the utilization of waste heat developed, and now quite often tri-generation plants are built, where production heat which comes from production of electric energy is used as a heat source for powering the absorption cooling machines. In this kind of technical solutions, absorption refrigeration cycles can effectively compete against compressor refrigeration systems, because the COP referenced to chemical energy stored in fuel is higher. This leads to better utilization of fuel energy, cost saving and lower pollution of the environment.

The following can be used as a heat source for powering the absorption cycle:

a) (exhaust) fumes,

b) hot process water,

c) exhaust steam.

Absorption plants are more and more popular where hot water or brine for powering comes from solar collectors. These solutions can significantly reduce the amount of eclectic power needed to achieve the cooling effect.

3 Heat pumps

3.1 Introduction

The 170-year old history of refrigeration engineering brings many methods of artificial cooling. It is always connected with heat transfer from a high temperature heat source to a low temperature heat source. Basics of this were taken from Sadi Carnot's work: *Reflections on the Motive Power of Heat and on Machines Fitted to Develop That Power*. Paris: Bachelier. (French), 1824.

Machines that make possible heat transport by the reversible Carnot cycle are:

– refrigerators – cooling down below the ambient temperature;

– heat pumps – heating up over the ambient temperature.
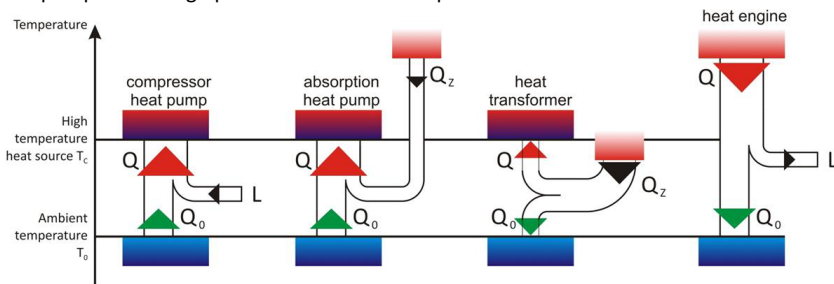

Figure 3.1. The heat transfer from a heat source.

Around 1920-1921 the name "HEAT PUMP" was used for the first time independently by two scientists: G. Flugel in Germany – "Warmepumpe" and F. Kraus in USA – "Heat Pump".

A heat pump extracts heat from the source at lower temperature and rejects heat to the source at higher temperature. The working fluid (refrigerant) changes from liquid to gas (evaporates) as the heat is absorbed from the heat source (H). Later in the cycle, the working fluid condenses to liquid as the heat is released to where it is needed (A).
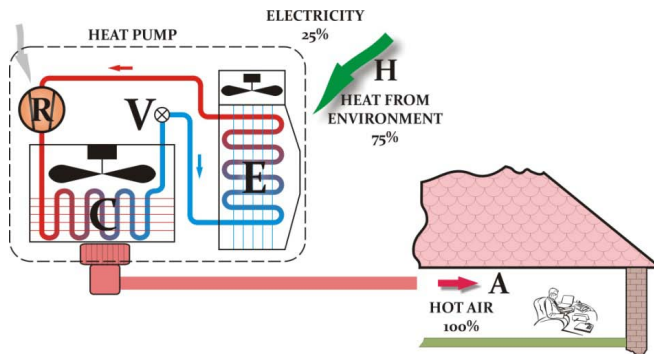

Figure 3.2. Basics of the heat pump operation.

The first heat pump (in laboratory scale) was built by Narine in 1778. It was a very simple heat device which allowed to take heat from one place and release it at another place (Fig. 3.3)
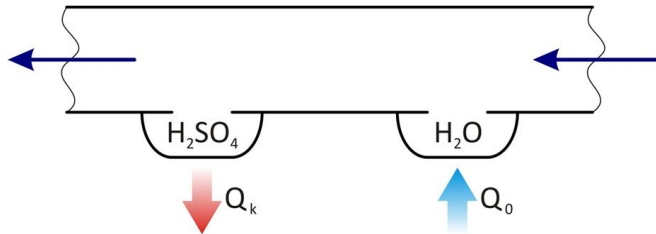

Figure 3.3. First heat pump build in laboratory scale.

In 1852 other heat pump application named "heat multiplier" was built by William Thomson (Lord Kelvin).
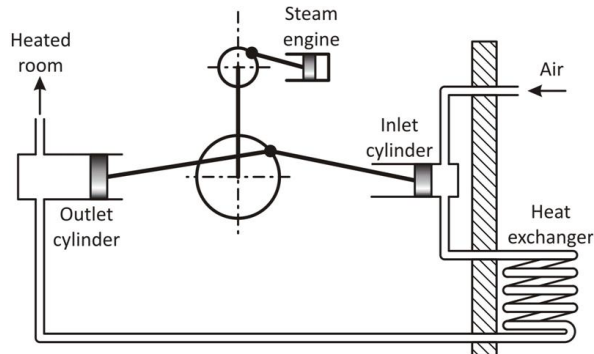

Figure 3.4. Heat multiplier by William Thomson (1852).

In the history of heat pumps there were many innovations, especially in late 30-ties and 40-ties of twentieth century. At that time Swiss and English appliances were well-known on the heat pump market.
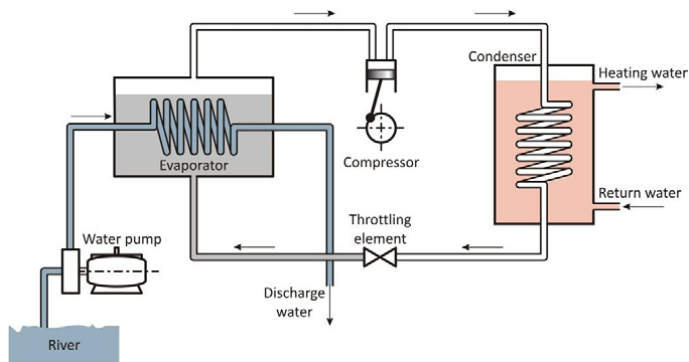

Figure 3.5. Norwich heat pump for an office heating system built in 1945.

According to the size heat pumps can be classified as:

– Small heat pumps: heating and air-conditioning of small rooms; preparing of hot tap water.

– Medium size heat pumps: year round heating and air-conditioning of hotels, offices and in the industry.

– Large size heat pumps: heating and air-condition of buildings, trade or leisure complexes.

Other heat pump applications are presented in Table 3.1

Table 3.1. Heat pump application

Application and size	Household 1 to 70 kW	Craft, agriculture 2 to 120 kW	Industry 0,1 to 10 MW
Application	Room heating	Hot water production, evaporation	Drying
Heat distribution	Hot air, hot water	Immediate contact, radiation	-
Heat source	Air, surface water	Ground water, ground, Sun	Waste heat
Heat pumping process	Refrigerant compressing, absorption device	Air refrigerator, vapor compression	Thermoelectric
Main product	Heat, simultaneous heating and cooling	Cooling during summer and heating during winter	Cooling by waste heat

3.2 Heat pump classification

Heat pump systems can be classified into groups dependent on such categories as:

Principle of Operation:

– compressor heat pumps:

- single-factor:
  - o an electrically powered heat pump;
  - o a gas powered heat pump;
  - o an internal-combustion engine powered heat pump;
  - o a gas-cooling heat pump;
  - o high-temperature (two-stage) heat pump;
  - o heat pump powered by other source.
- mixture-factor:
  - o zeotropic;
  - o azeotropic.

– sorption heat pumps:

- heat boost absorption heat pump;
- heat reduce absorption heat pump;
- resorption heat pump.

– compressor-sorption heat pumps;

– thermo-compressor heat pumps;

– ejector heat pumps;

– thermo-electric heat pumps;

– rotary heat pumps;

– magnetic with gas circuit heat pumps;

– heat pumps with open circuit of air;
– heat pumps with the electro-diffusion effect;
– heat pumps utilizing the Ranque effect.

Heat source:

– monovalent heat pumps;
– bivalent heat pumps:

- serial;
- parallel;

– multivalent heat pumps;
– primary heat pumps – use natural a heat source present in the environment;
– secondary heat pumps – reuse waste heat as a heat source;
– tertiary heat pumps – they are in series with a primary or secondary heat pump.

With this kind of classification is connected heat pump nomenclature. Whole used names are presented in Table 3.2.

Table 3.2. Heat pumps classification – heating of buildings.

Heat source	Coolant	Heat carrier	Heat pump name	Heat devices name (HPD – Heat Pump Device)
Water	Hot water	-	Water to water	HPD water/water
Water	Hot air	-	Water to air	HPD water/air
Air	Hot water	-	Air to water	HPD air/water
Air	Hot air	-	Air to air	HPD air/air
Air	Hot water	Brine	Brine to water	HPD air/water
Air	Hot air	Brine	Brine to air	HPD ground/air
Ground	Hot water	-	Ground to water	HPD ground/water
Ground	Hot air	-	Ground to air	HPD ground/air
Ground	Hot water	Brine	Brine to water	HPD ground/water
Ground	Hot air	Brine	Brine to air	HPD ground/air

3.3 Heat energy

Based on the First Law of Thermodynamics it can be written:

$$E_a + B_a = const \tag{3.1}$$

and according to Figure 3.6 a heat Q can be written as:

$$Q = E_a + B_a \tag{3.2}$$

As the heat of the transformation process from point 1 to point 2 can be written as follows:

$$Q_{1-2} = \int_1^2 TdS \tag{3.3}$$

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and

$$Q_{1-2} = B_{Q_{1-2}} + E_{Q_{1-2}},$$ (3.4)

Because B$_{Q1-2}$ can be written that:

$$B_{Q_{1-2}} = T_{am} \int_{1}^{2} dS$$ (3.5)

and

$$E_{Q_{1-2}} = \int_{1}^{2} \left(T - T_{am}\right) dS = \int_{1}^{2} \left(1 - \frac{T_{am}}{T}\right) dQ$$ (3.6)

so heat energy can be determined as:

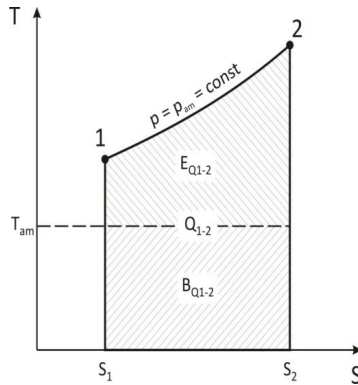$$E_{Q_{1-2}} = Q_{1-2} - T_{am} \int_{1}^{2} \frac{dQ}{T}$$ (3.7)

Figure 3.6. Isobaric transformation from point 1 to point 2 on T-s diagram.
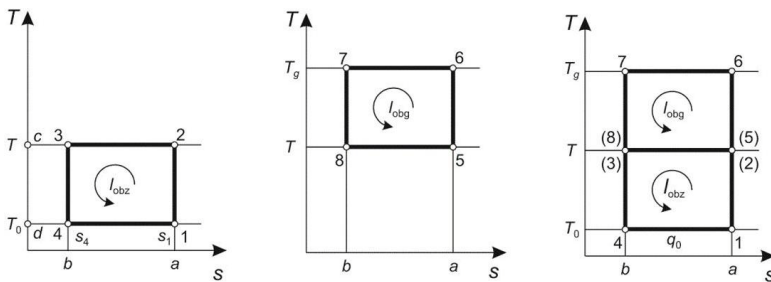

Figure 3.7. Reversible Carnot cycle realized on constant temperature levels.

Reversible Carnot cycle realized on constant temperature levels is the base for the heat pump efficiency consideration, as it is presented in Fig. 3.7.

Processes which are occurring in this cycle are described in Chapter 1. For the efficiency estimation of this cycle Linde cycle can helpful (Fig. 3.8).
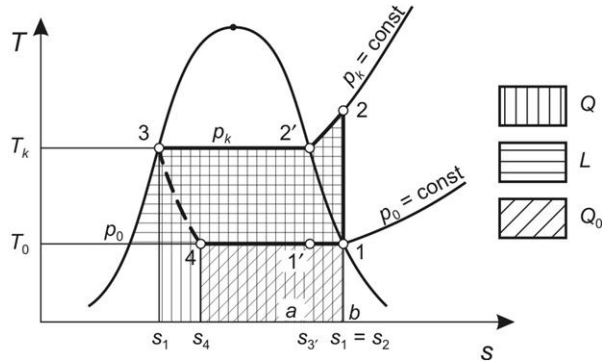

Figure 3.8. Linde cycle and determined heat of condensation, heat of evaporation, and work of the cycle.

Based on Figure 3.8 efficiency of the heat pump can be determined:

$$\eta_{HP} = \frac{T_k \Delta s}{(T_k - T_{am}) \Delta s} \tag{3.8}$$

Equation (3.8) describes the heat pump efficiency and shows that η_{HP} depends only on the temperature of condensation and the temperature of environment (ambient). Temperature of condensation strongly impacts heating efficiency of the Carnot cycle. It is presented in Figure 3.9.
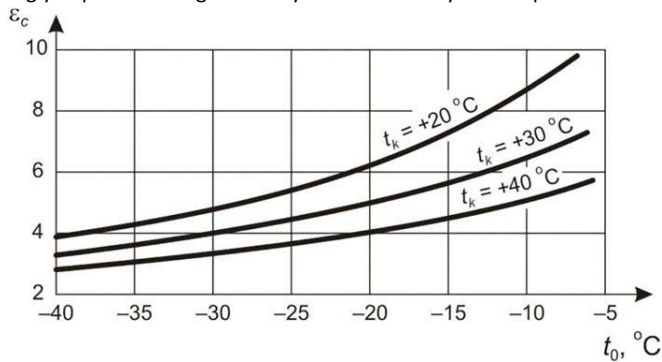

Figure 3.9. Impact of the temperature of condensation on the heating efficiency of Carnot cycle.

The Second Law of Thermodynamic indicates that a thermal machine working between two heat sources (at constant temperatures) achieves greatest efficiency if it follows the reversible Carnot cycle. In refrigeration engineering, this represents a reversible cycle (so-called ravines comparative) for a compressor cycle, with unary or azeotropic matter, which changes its state of aggregation

(evaporation and condensation) at constant temperature. In this cycle, it is accepted that the differences between the temperatures of both heat sources and the intermediary factor are near zero (Fig. 3.10 – points 1-2-3-4).
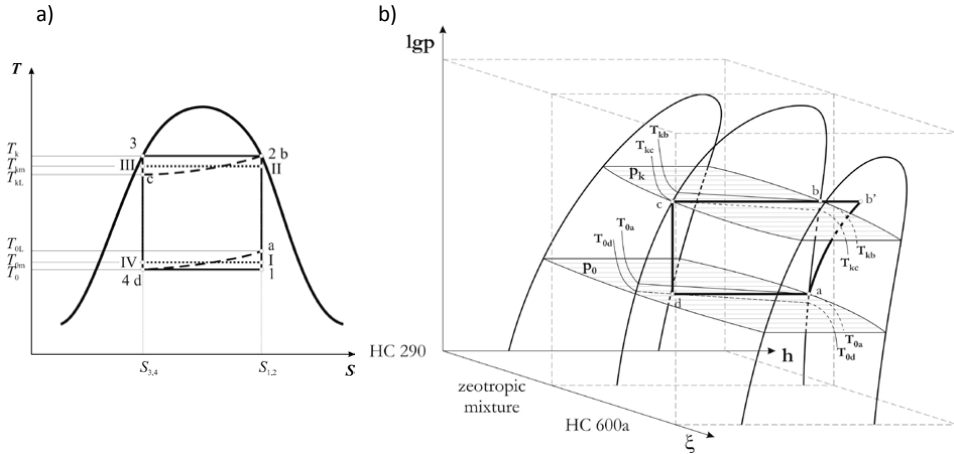

Figure 3.10. Heat pump cycles: a) statement of the Carnot cycle (1-2-3-4), the Lorenz cycle (a-b-c-d) and the supplementary Carnot cycle based on the average temperatures of the Lorenz cycle (I-II-III-IV) in T – S coordinates; b) the Linde cycle in lgp-h-ξ coordinates.

Using a zeotrope creates completely new thermodynamic and technical possibilities connected with temperature glide during phase reverse. Constructors and users of heat pumps filled with mixtures of natural factors are by necessity turning more attention towards different views on the transformations occurring inside the installation. For example, it appears advisable to resign from the Carnot cycle as a comparative cycle, and to accept the theoretical Lorenz cycle for analysis of the theoretical Lorenz cycle.

The Lorenz cycle is a reversible, ravines, compressor cycle, that uses zeotropic mixtures of two or more substances. It is characteristic that, at variable temperatures, the mixture changes its state of aggregation during evaporation and condensation. The assumption is made that heat exchange processes take place during politropic processes (at p = const and ≠tconst), while tensing and expanding of a given factor takes place isentropically.

The energetic efficiency of refrigerating can be represented in dependences by averaging the temperatures of the two heat sources, where the lower one:

$$T_{0m} = \frac{\int_d^a T_{0i}dS}{S_a - S_d} = \frac{T_a - T_d}{\ln\dfrac{T_a}{T_d}} \approx \frac{T_a + T_d}{2} \tag{3.9}$$

and upper one

$$T_{km} = \frac{\int_b^c T_{ki}dS}{S_b - S_c} = \frac{T_b - T_c}{\ln\dfrac{T_b}{T_c}} \approx \frac{T_b + T_c}{2} \tag{3.10}$$

The Lorenz and Carnot cycles energetic efficiency of refrigerating (in dependences on average temperatures) adopts the following form:

$$\varepsilon_L = \frac{T_{km}}{T_{km} - T_{0m}} \tag{3.11}$$

so:

$$\varepsilon_C = \frac{T_k}{T_k - T_0} < \varepsilon_L \tag{3.12}$$

Considering equations (3.11) and (3.12), we obtain the result that the cycle has a greater energetic efficiency of heating when the average difference between the temperatures T_{km} and T_{0m} is lower. Small differences in temperatures occur wherever temperature glide occurs during phase aggregation (e.g. with usage of zeotropic mixtures).

Thermodynamic aspects show the full legitimacy of using refrigerant mixtures which changes phase following variation in temperature, and only economic regards can be the reason for the rejection of these mixtures.

However, the question remains of the quantitative qualification of the influence that temperature glide has on the processes of the Lorenz cycle energetic efficiency of heating. It proves that the magnitude of the temperature glide is dependent, in the case of zeotropic mixtures, on the kind of components, and above all on the molar composition of the mixtures.
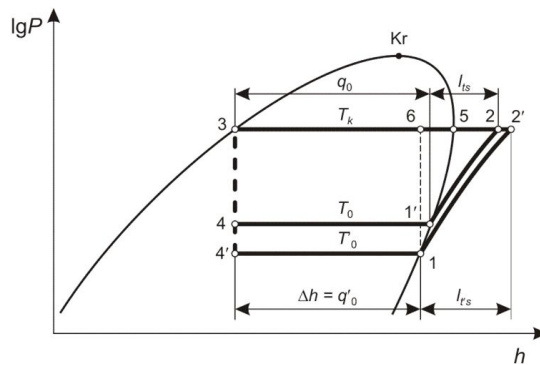

Figure 3.11. The influence of evaporation temperature on the COP.

Other important factor that is important for heat pump efficiency consideration is the influence of evaporation temperature on the COP. It is showed in Fig. 3.11.

According to this figure it can be written:

$$q_0 = h_1 - h_4 \tag{3.13}$$

and

$$q_0^{'} = h_{1'} - h_{4'} \tag{3.14}$$

Because:

$$h_4 = h_{4'} \text{ and } h_{1'} < h_1 - q_0^{'} < q_0 \tag{3.15}$$

and from the other hand:

$$l_{ts} = h_2 - h_1 \tag{3.16}$$

and

$$l_{ts}^{'} = h_{2'} - h_{1'} \tag{3.17}$$

Because:

$$h_1 > h_{1'}, h_{2'} > h_2 \tag{3.18}$$

so:

$$\varepsilon_L = \frac{q_0}{l_t} > \frac{q_0^{'}}{l_0^{'}} = \varepsilon_L^{'} \tag{3.19}$$

On this basis we can say that the coefficient of performance is lower when evaporation temperature goes down.

The same situation occurs when condensation temperature goes up. It is shown in Fig. 3.12.
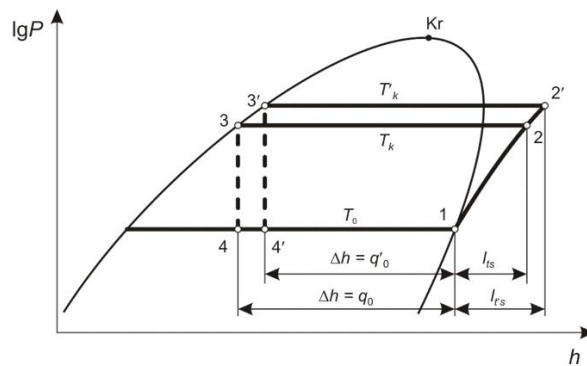

Figure 3.12. The influence of condensation temperature on COP.

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3.4 Heat sources

The ideal heat source is characterized as follows:

– unlimited heat capacity (renewable);

– the highest and most constant temperature;

– without any pollutant and corrosive substances;

– the cheapest possible exploitation;

– availability;

– the source coherency.

Of course it is impossible to find this kind of heat source.

Depending on the place origin heat sources can be divided into two main groups:

– natural;

– artificial

These both types of heat sources can be described by their location, heat exchanger construction materials, and annual, monthly and daily temperature changes. Generally a heat source can be considered as in Table 3.3.

Table 3.3. Heat sources.

Secondary heat pump	Primary heat pump
Waste energy sources	Renewable energy sources
Air Waste gases Combustion gases: • from gas combustion, • from solid fuel combustion, • from systems Condensate Vapor Water	Fresh air Ground Water: • ground water • surface water • deep-well water • urban water Solar radiation Geothermal energy Wind energy Ambient energy

A low temperature heat source determines the type of heat pump and it covers the application area. It should have a specific qualitative and quantitative characteristic. As qualitative factors we can assume:

– availability

– corrosivity

and as quantitative:

– temperature and its changing in time

– exergy reserves and their changing in time

– investment and exploitation costs

As the low temperature heat sources we can describe:

– air;

– ground;

– solar radiation;

– subterranean water;

– a drilled well;

– river;

– lake;

– seas and oceans.

Air

Strengths: availability, low investment costs, useful even at -15°C.

Weaknesses: low coherency, low film coefficient of heat transfer a, large heat exchanging areas, low temperatures and efficiencies, noise from fans, frost, temperature changes.

Water

Water is an attractive heat source, has good physical properties and good thermodynamic properties. Has high values of heat exchanging coefficients, good transportation properties, requires compact heat exchangers and brings monovalent heat pump construction possibility.
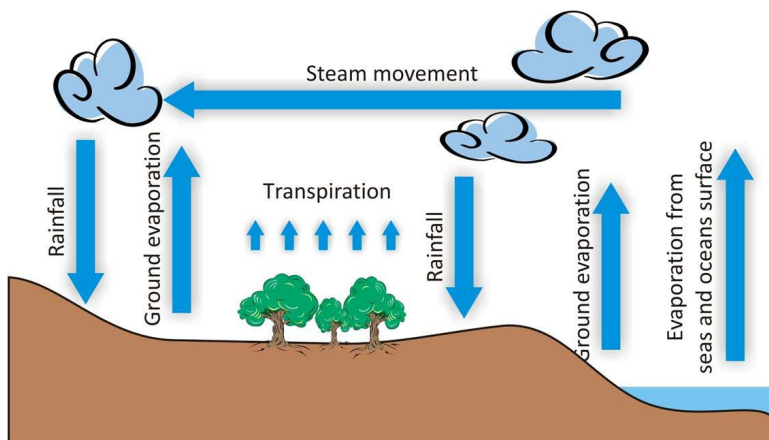
Water environmental circuit is shown in Fig. 3.13.

Figure 3.13. Water environmental circuit.

For the economy, it is, of course, important that the piping length between the house and the collector can be kept short. This type of heat source is very rarely used. Typical data are very close to the ones for the soil heat pump.

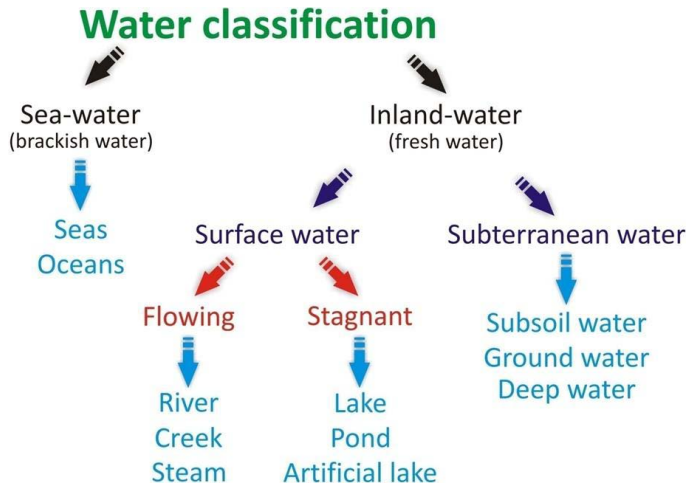Typical water-heat source can be divided as in Fig. 3.14.

Figure 3.14. Classification of water used as the heat source.

Geothermal heat

Geothermal heat pumps are highly efficient, renewable energy technology for space heating and cooling. This technology relies on the fact that, at depth, the Earth has a relatively constant temperature, warmer than the air in winter and cooler than the air in summer. A geothermal heat pump can transfer heat stored in the Earth into a building during the winter, and transfer heat out of the building during the summer. Special geologic conditions, such as hot springs, are not needed for successful application of geothermal heat pumps. Ground-source heat pumps are receiving increasing interest because of their potential to reduce primary energy consumption and thus reduce emissions of greenhouse gases.
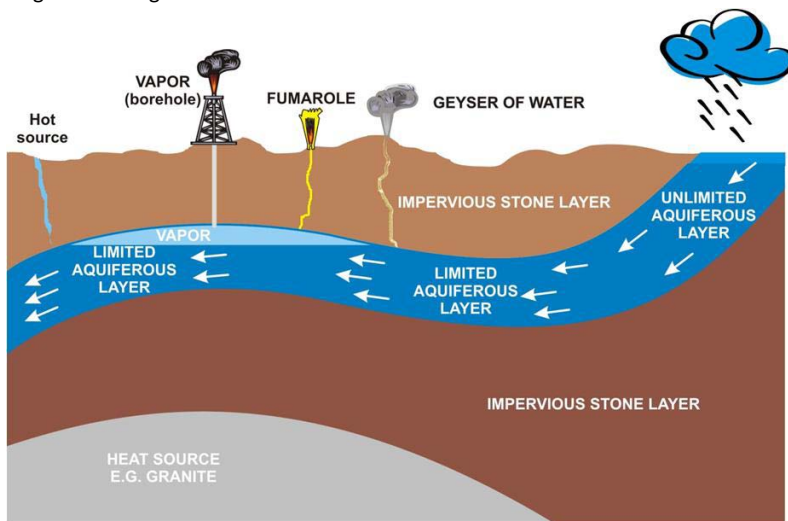

Figure 3.15. Geothermal water as a heat source.

One of the main geothermal heat sources is geothermal water. Its circuit is shown in Fig. 3.15.

Ground

Ground as a heat source has very small exploitation costs, constant temperature, can accumulate the heat in the ground. Has also possibility of ground heat exchanger use as evaporator or condenser. Moreover it can be used as a heat source for a monovalent heat pump and is very useful in bivalent variant (with solar heat). But has high investments costs and requires a large area for installation.
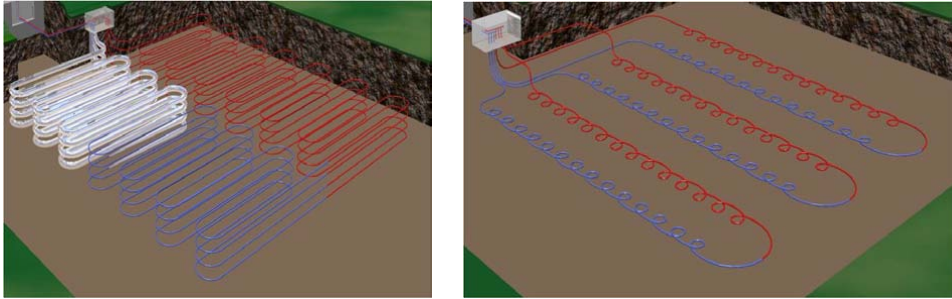

Figure 3.16. Typical installation of ground heat exchangers.

Solar radiation

Solar technology presents a novel system whereby the performance of the heat pump can be significantly enhanced by taking heat from a natural source – solar energy. The applications for a solar heat pump system include water heating, heat storage and drying. Solar radiation meets some barriers in the atmosphere (Fig. 3.17).
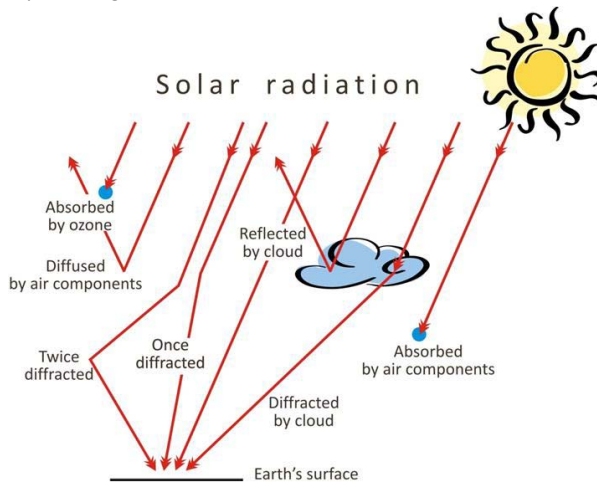

Figure 3.17. Interaction between solar radiation and the atmosphere.

The most convenient way of solar heat exploitation is solar collectors. But the efficiency of the heat transformation strongly depends on collector's construction and used materials. There are some losses in the collector's cover (around 10%), in the absorber's surface (6%), in the absorber (21%),

and through the isolation (3%). This is the reason why this kind of heat source is not so popular to use as a heat source for heat pumps in central and north European countries (Fig. 3.18).
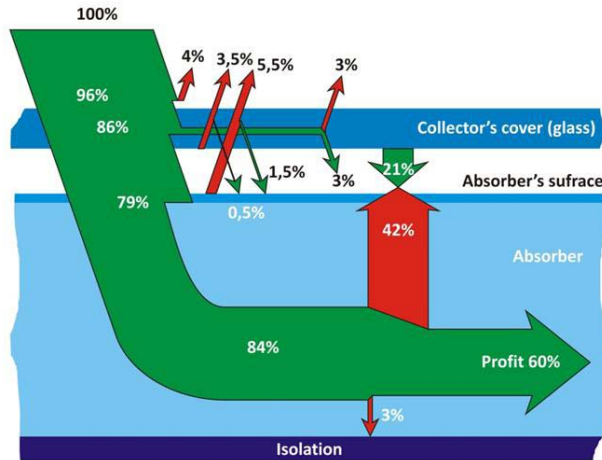

Figure 3.18. Loss in a solar collector.

Artificial heat source

This kind of heat source can be considered as a high-potential and low-potential source (P = 500kW). It may have continuous and discontinuous heat emission and as a heat carrier can be used vapor, water or condensate. Hot air or exhaust fumes from gas solid fuel combustion can be used as well (P = 12 – 700kW). Exhaust fumes from solid fuel combustion in smelting systems are more potential and may achieve P = 140 – 8000kW.

3.5   Heat pump with gas cycle

In the Fig. 3.19. there is shown a gas cycle realized by a heat pump.
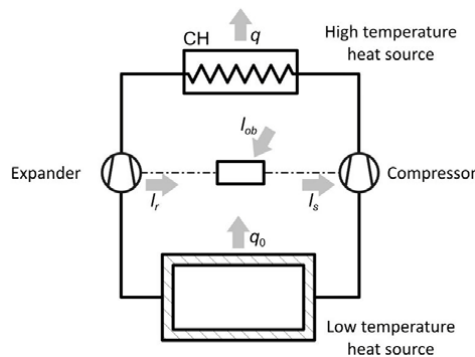

Figure 3.19. Gas cycle heat pump

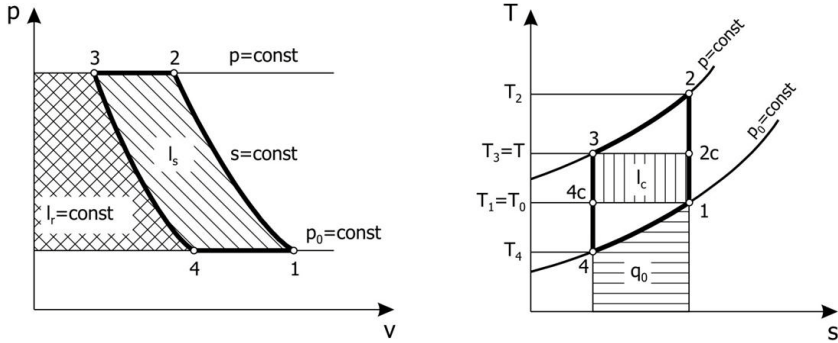Theoretical cycle of this heat pump is shown in Fig. 3.20.

Figure 3.20. Theoretical cycle of the heat pump with a gas cycle in p-v and T-s diagrams.

Heat taken from the condenser can be written as:

$$q_k = h_2 - h_3 = c_p\left(T_2 - T_3\right) \tag{3.20}$$

Specific compressing work:

$$l_s = h_2 - h_1 = c_p(T_2 - T_1) \tag{3.21}$$

and adiabatic decompressing:

$$l_r = h_3 - h_4 = c_p(T_3 - T_4) \tag{3.22}$$

Theoretical heat balance:

$$q_0 + l_s = q + l_r \tag{3.23}$$

So theoretical coefficient of efficiency:

$$\varepsilon_t = \frac{q_k}{l_{ob}} = \frac{q_k}{l_s - l_r} = \frac{c_p(T_2 - T_3)}{c_p(T_2 - T_1) - c_p(T_3 - T_4)} \tag{3.24}$$

3.6 Absorption heat pumps

Absorption heat pumps operate according to the scheme presented in Fig. 3.21.
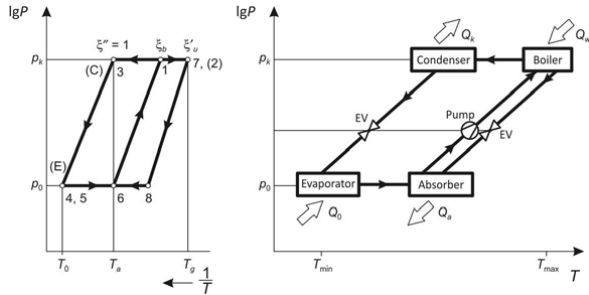
Figure 3.21. Theoretical cycle of an absorption heat pump.

Adequate theoretical reversible Carnot cycle of an absorption heat pump is shown in Fig. 3.22.
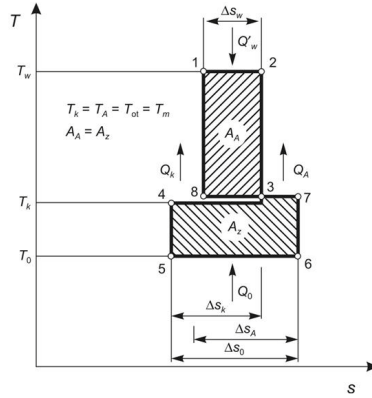

Figure 3.22. Theoretical reversible Carnot cycle of an absorption heat pump.

When we consider the heat balance:

$$Q_0 + Q_w - Q_k - Q_A = 0 \qquad (3.25)$$

we can describe all the elements. Evaporation heat:

$$Q_0 = T_0 \left(S_6 - S_5 \right) = T_0 \Delta S_0 \qquad (3.26)$$

absorption heat:

$$Q_A = T_m \left(S_7 - S_8 \right) = T_m \Delta S_A \qquad (3.27)$$

condensation heat:

$$Q_k = T_m \left(S_3 - S_4 \right) = T_m \Delta S_k \qquad (3.28)$$

boiling heat:

$$Q_w = T_Z \left(S_2 - S_1 \right) = T_w \Delta S_w \qquad (3.29)$$

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From the heat balance results:

$$\Delta S_k + \Delta S_A = \Delta S_w + \Delta S_0 \tag{3.30}$$

so:

$$\Delta S_A = \Delta S_w + \Delta S_0 - \Delta S_k \tag{3.31}$$

After replacement in equations (3.25) to (3.29) and transformation we can write as follow:

$$\Delta S_w \left(T_w - T_m \right) = \Delta S_0 \left(T_m - T_0 \right) \tag{3.32}$$

So efficiency of an absorption heat pump can be written as:

$$\varepsilon_{HP} = \frac{Q_A + Q_k}{Q_w} = \frac{T_m \left(\Delta S_z + \Delta S_0 \right)}{T_w \Delta S_w} = \frac{T_m}{T_z} \frac{T_w - T_0}{T_m - T_0} =$$
$$= \frac{T_w - T_0}{T_w} \frac{T_m}{T_m - T_0} \tag{3.33}$$

In general there are some methods of the COP improvement in heat pump engineering. We can list them as building a multi-stage cycle, cascade cycle, Voorhees cycle, using economizer and special cycles application.

3.7 Ejector heat pump

The system consists of a boiler, an ejector nozzle, an evaporator, a condenser, a pump and an expansion valve. The ejector unit consists of a primary nozzle and mixing tube – diffuser. When heat is supplied to the boiler, the working fluid (e.g. ammonia) evaporates and flows to the ejector, where it expands through the primary nozzle, thereby entraining low-pressure refrigerant vapor from the evaporator, producing a refrigeration effect. The ejector exhaust is discharged into the condenser. Form here some of the liquid refrigerant is returned to the generator using a pump, while the remainder is expanded through an expansion valve to the evaporator.
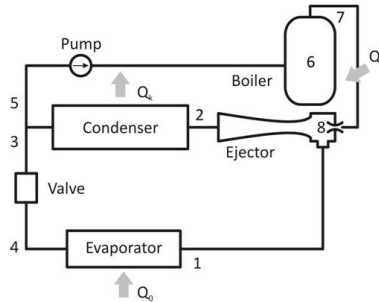

Figure 3.23. Ejector heat pump.

The minimum evaporator temperature is limited to 0°C, when water is used as the refrigerant. The use of alternative working fluids, such as ammonia, R 134a, R32 and Care 30, would allow lower operating temperatures to be achieved.
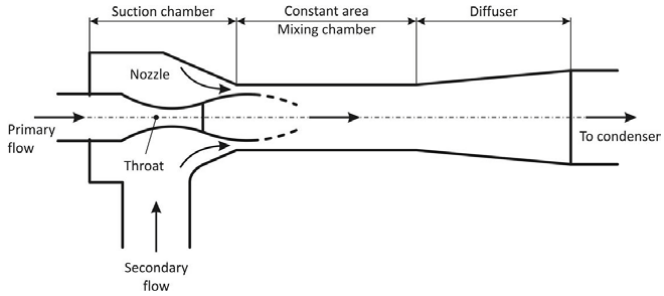
Figure 3.24. Ejector scheme.

3.8 Thermoelectric heat-pump

A semiconductor thermoelectric heat-pump is based on the Peltier effect. We can notice there the absence of moving components and a working fluid. It is used in military, aerospace, instrument and industrial or commercial products.
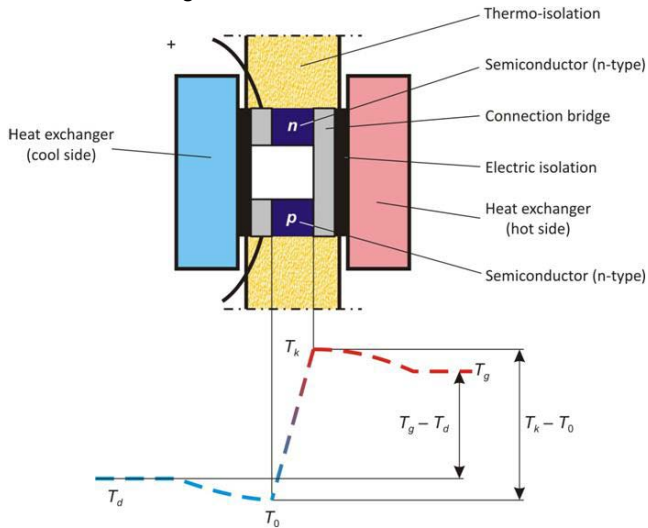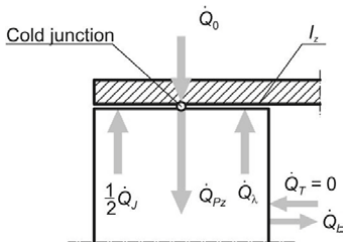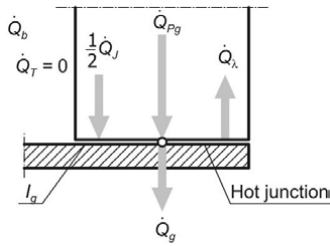
Basic components are shown in Fig. 3.25.

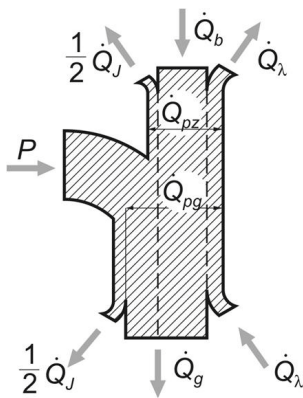Figure 3.25. Basic components and operational mode of a thermoelectric heat pump.

Junctions of the thermoelement we can consider as:

$$-\dot{Q}_{Pz} + \frac{1}{2}\dot{Q}_J + \dot{Q}_\lambda + \dot{Q}_0 = 0 \qquad (3.34)$$

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$$\dot{Q}_{Pg} + \frac{1}{2}\dot{Q}_J - \dot{Q}_\lambda - \dot{Q}_g = 0 \tag{3.35}$$

Peltier heat:

$$\dot{Q}_{Pz} = \left(\alpha_p - \alpha_n\right)IT_z \tag{3.36}$$

$$\dot{Q}_{Pg} = \left(\alpha_p - \alpha_n\right)IT_g \tag{3.37}$$

Joule heat:

$$\dot{Q}_J = I^2 R \tag{3.38}$$

$$R = L\left(\frac{1}{\sigma_p f_p} + \frac{1}{\sigma_n f_n}\right) \tag{3.39}$$

3.9 Heat pump in the heating system

The heat pump based heating system has many benefits because of its working parameters. As the advantages of low temperature heating we can list:

– lower heat losses when the heating water flows from the furnace to the radiators and back to the furnace;

– possibility of using heat pumps with the high coefficient of performance (COP);

– under-floor heating option;

– highly efficient condensing light heating oil furnaces.

The drawbacks of low temperature heating can be listed as:

– radiators and pipes must be larger, which results in higher investment costs;

– higher quantity of water in the central heating system.

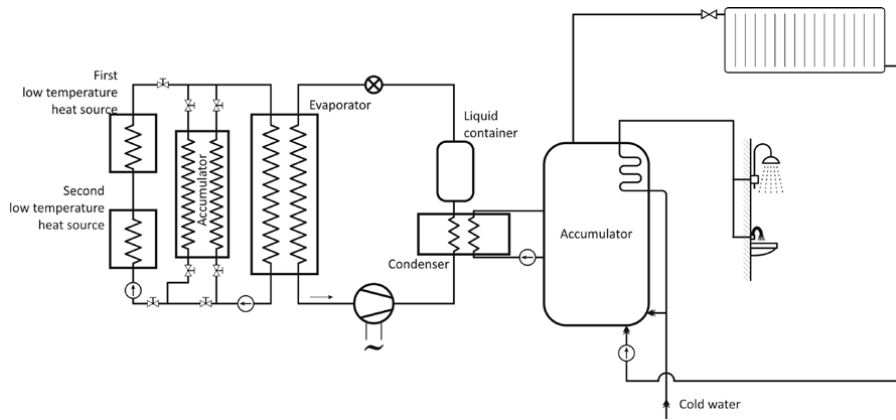One of the possible heating system is shown in Fig. 3.26.

Figure 3.26. Heat pump in the heating system.

3.10 Heat pump development trends

Nowadays there are many directions developed in industry offices and scientific centers focused on heat pump engineering. Some of them are highly specialized and used in only specific demands when the others are commonly used in a wide scale of application. Table 3.4 presents modern trends in this area.

Table 3.4. Heat pump development trends.

Year	Type of study		Type of system		Application			Result(s)
	Theoretical (simulation)	Experimental	Heating	Cooling	Water heating	A/C	Other	
2010	√		√		√			SAHP system – means achievable COP of 6.46
2009		√	√		√			Improved COP about 6% with the use of new refrigerant mixture
2008		√		√		√		HP with ejector – COP approaching as high a value as 6
2008		√		√		√		HP with dehumidifier – COP approaches 5 to 6

4   Air conditioning systems

4.1   Introduction

Air conditioning covers following matters:

– temperature;

– moisture in the air (humidity);

– supply of outside air for ventilation;

– filtration of airborne particles;

– air movement in the occupied space.

The most important air conditioning processes are:

- Heating – the process of adding thermal energy to the conditioned space for the purposes of raising or maintaining the temperature of the space.
- Cooling – the process of removing thermal energy from the conditioned space for the purposes of lowering or maintaining the temperature of the space.
- Humidifying – the process of adding water vapor to the air in the conditioned space for the purposes of raising or maintaining the moisture content of the air.
- Dehumidifying – the process of removing water vapor from the air in the conditioned space for the purposes of lowering or maintaining the moisture content of the air;
- Cleaning – the process of removing particulates and biological contaminants from the air delivered to the conditioned space for the purposes of improving or maintaining the air quality.
- Ventilating – the process of exchanging air between the outdoors and the conditioned space for the purposes of diluting the gaseous contaminants in the air and improving or maintaining air quality, composition, and freshness.
- Air movement – the process of circulating and mixing air through conditioned spaces in the building for the purposes of achieving the proper ventilation and facilitating the thermal energy transfer.
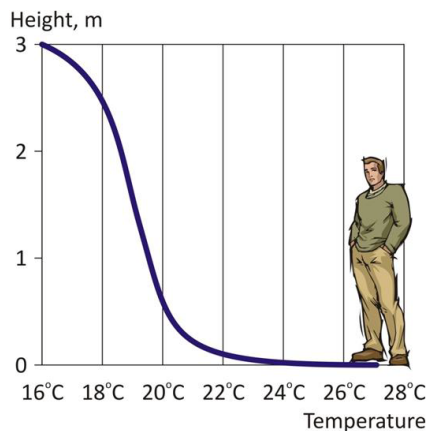

Figure 4.1. The best temperature distribution depends on height.

Thermal comfort is connected with air-conditioning engineering as well. It depends on activity level, clothing, expectation, air temperature, radiant temperature, humidity, and air speed. In Fig. 4.1 we can see thermal zones convenient for human race.

Heat balance of human body is also important in air conditioning projects. It is shown in Fig. 4.2.
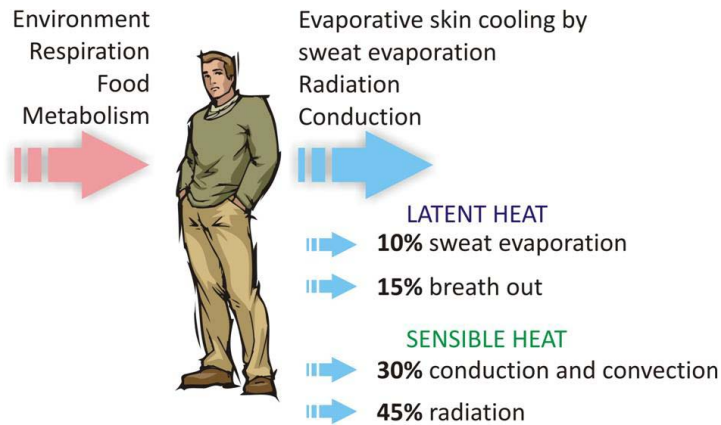

Figure 4.2. Heat balance of a human body.

During air flow our skin feels different types of heat interaction. According to air velocity it can be described as:

$$t_{od} = 33 + \left(0,478 + 0,237 \cdot \sqrt{v} - 0,0124 \cdot v\right) \cdot \left(t_p - 33\right) \tag{4.1}$$

where:
t_{od} – sensible temperature, ^{o}C;
t_p – air temperature, ^{o}C;
v – wind velocity, km/h

4.2 Air conditioning systems classification

The main classification of air conditioning systems can be described as:
- individual room air conditioning systems or simple individual systems;
- evaporative cooling air conditioning systems;
- desiccant-based air conditioning systems or simple desiccant systems;
- thermal storage air conditioning systems or simple thermal storage systems;
- clean room air conditioning systems or simple clean room systems;
- space conditioning air conditioning systems or simple space systems;
- unitary packaged air conditioning systems or simple packaged systems;
- central hydronic air conditioning systems or simple central systems.

4.3   Psychrometric chart

In air conditioning engineering a psychrometric chart is the very useful tool for many calculations and estimations. It is shown in Fig. 4.3.
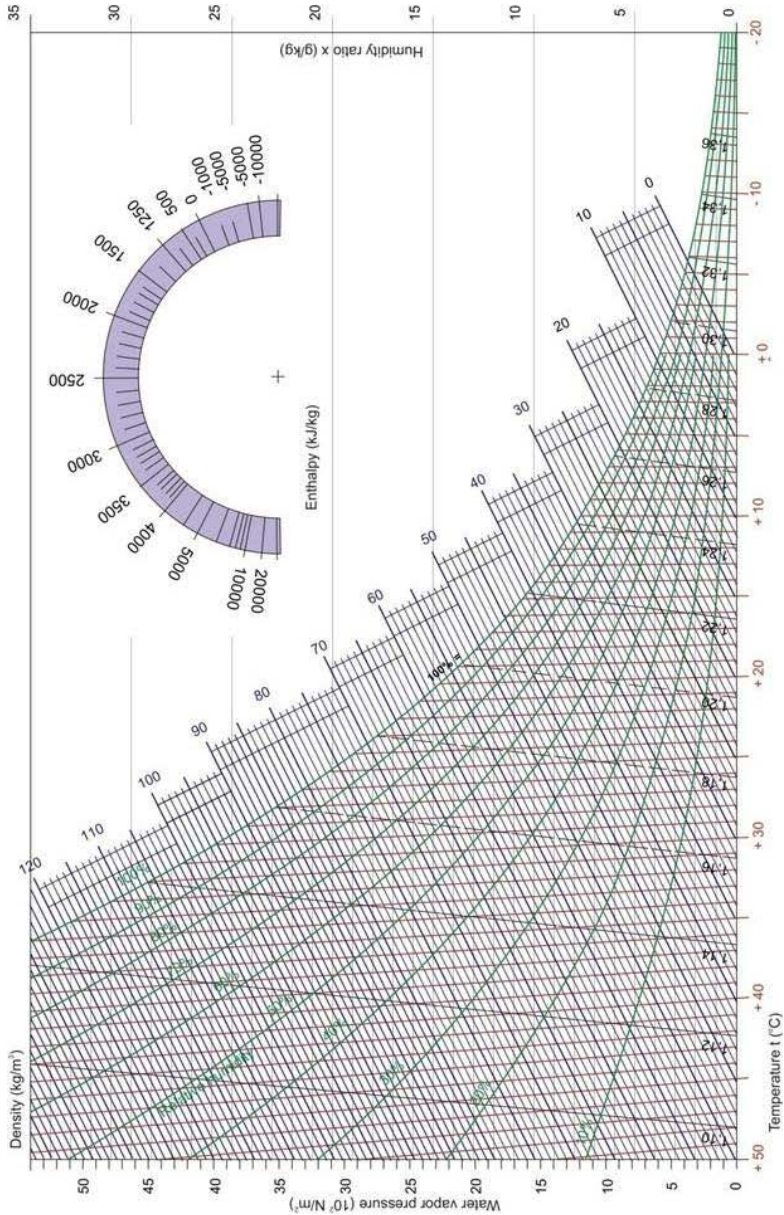

Figure 4.3. Psychrometric chart.

Basic thermodynamic changes on a psychrometric chart are shown in Fig. 4.4 to Fig 4.8.

Heating ($t_2 > t_1$ and $x_2 = x_1$)
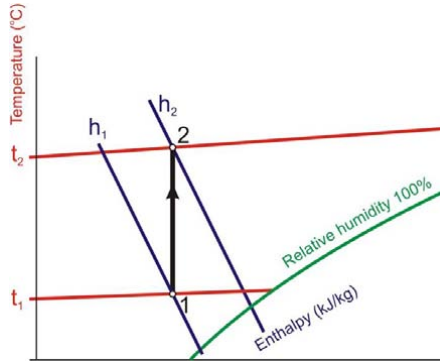
Figure 4.4. Heating realized on a psychrometric chart.

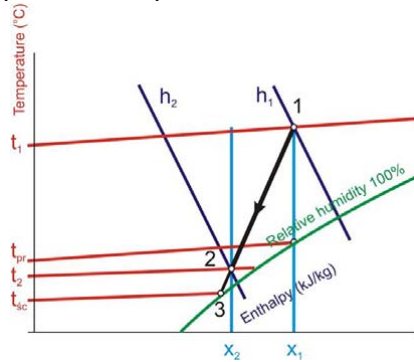**Cooling and dehumidifying ($t_2 < t_1$ and $x_2 < x_1$)**

Figure 4.5. Cooling and dehumidifying realized on a psychrometric chart.

Humidifying by water ($x_2 > x_1$ and $t_2 < t_1$)

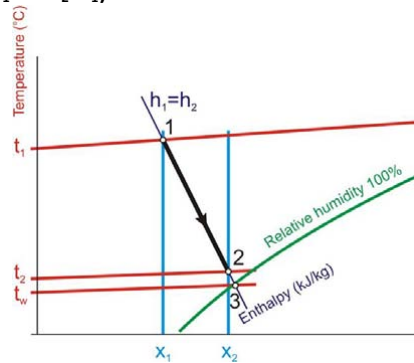Figure 4.6. Humidifying by water realized on a psychrometric chart.

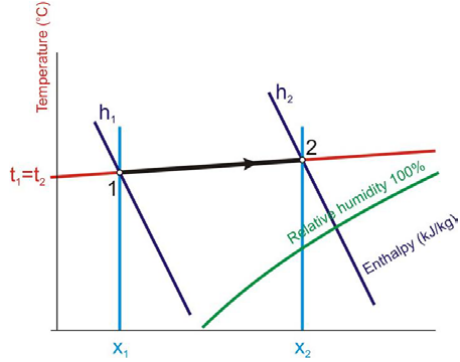**Humidifying by water vapor ($x_2 > x_1$ and $t_2 = t_1$)**

Figure 4.7. Humidifying by water vapor realized on a psychrometric chart.

Air mixing ($t_1 < t_3 < t_2$ and $x_1 < x_m < x_2$)
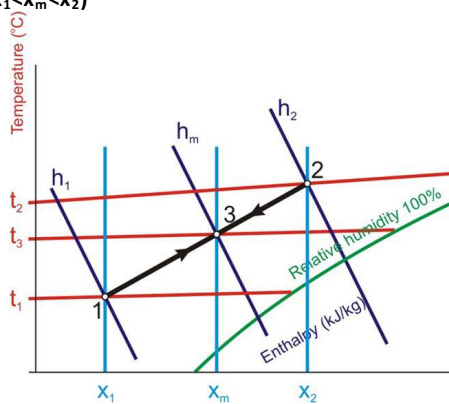
Figure 4.8. Air mixing realized on a psychrometric chart.

Mixing of the two air streams is described as follows:

$$m_1 \cdot t_1 + m_2 \cdot t_2 = m_3 \cdot t_3$$
$$m_1 + m_2 = m_3$$

(4.2)

When dry air and water vapor are mixed then:
– the temperature is constant, but the quantity of water vapor is increasing;
– the temperature is dropping, but the quantity of water vapor is constant;
   Specific amount of energy in the mixture at a specific temperature and pressure influences:
– the temperature of the air;
– the proportion of water vapor in the air;
– sensible heat transferred to the air;
– more water vapor, which increases the latent heat of the mixture.

4.4   Indoor environment

Indoor air quality depends mainly on the minimum ventilation control; the removal of bacteria, particulates, irritating vapors, and toxic gases; preventing dampness; and proper maintenance of the Heating, Ventilating, Air Conditioning & Refrigerating (HVAC&R) system. Among various minimum ventilation controls, the dedicated ventilation system with CO_2-based demand-controlled ventilation guarantees the minimum outdoor ventilation rate and is energy-efficient.

The dedicated ventilation system, the mixed plenum pressure control, and the outdoor air injection fan are good minimum ventilation control systems with satisfactory performance. The supply and return fan tracking systems, and the direct measurement of outdoor air and are considered as poor minimum ventilation control systems. As the volume flow rate of the outdoor ventilation air is usually less than one-fourth that of the total supply air, it is difficult to measure it correctly by measuring the volume flow rate of supply and return air. This may result in an insufficient outdoor ventilation intake, especially during part-load operation. Indoor environment depends also on air leakage paths (Fig. 4.9).
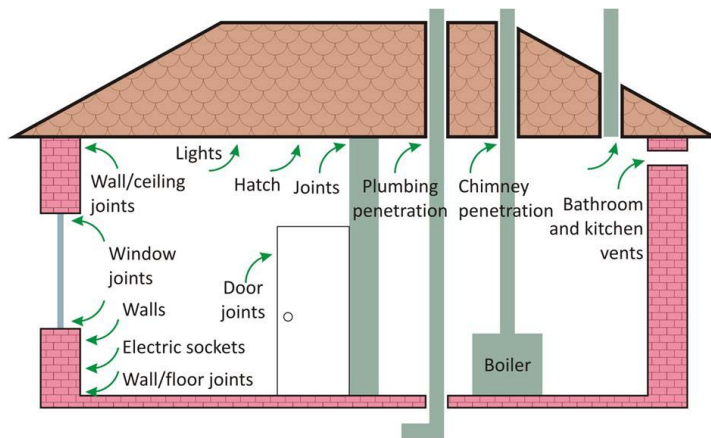
Figure 4.9. Typical air leakage paths.

4.5   Ventilation and air conditioning systems

When the air movement is induced either by wind or by the effect of temperature difference, ventilation is named natural. When air movement results from power drive applied to a fan, it is described as mechanical ventilation. So ventilation systems can be divided as:

- Natural ventilation:
  – single-sided ventilation;
  – cross-ventilation;
  – stack ventilation.
- Combined natural and mechanical (hybrid) ventilation:
  – hybrid ventilation in domestic and small buildings;
  – hybrid ventilation in commercial and large buildings.
- Ventilation for fire control.

Basic mechanical ventilation system equipped optionally in heating and cooling devices is presented in Fig. 4.10.
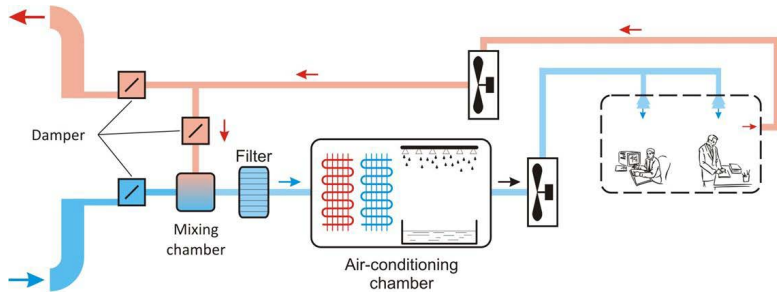

Figure 4.10. Basic mechanical ventilation system.

Air distribution systems are strongly connected with ventilation systems. They are presented in Fig. 4.11.
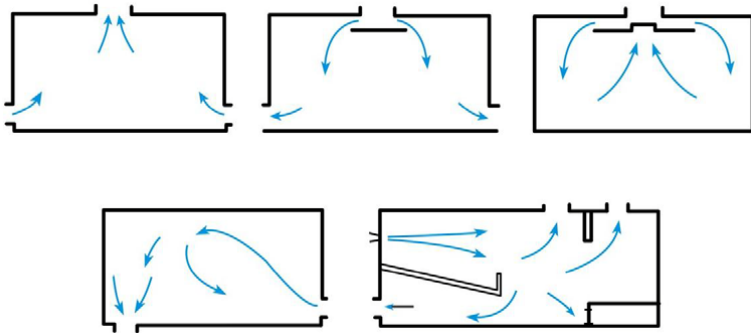

Figure 4.11. Air distribution systems.

Direct evaporative cooler

A direct evaporative cooler evaporates moisture into the air, reducing the temperature at approximately constant enthalpy. In a hot dry climate this process may often be enough to provide comfortable conditions for people. In medium to wet climates, the increase in moisture content is frequently not acceptable for sedentary human comfort but is considered acceptable for high effort work places and is ideal for some operations, such as greenhouses.

Direct evaporative coolers for residences in desert regions typically require 70% less energy than direct expansion air conditioners. The main advantages of evaporative cooling include substantial energy and cost savings, reduced peak power demand and reduced size of mechanical refrigeration equipment, easily integrated into built-up systems.

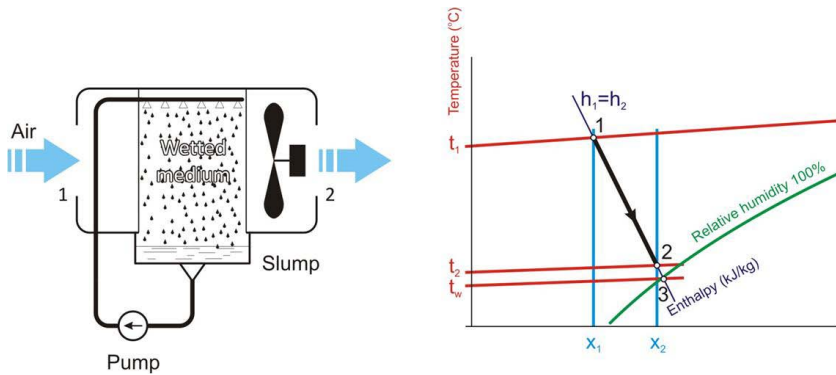A direct evaporative cooler is shown in Fig. 4.12.

Figure 4.12.Direct evaporative cooler and its representation on a psychrometric chart.

Indirect evaporative cooler

An indirect evaporative cooler uses evaporation to cool a surface, such as a coil, that is then used to cool the incoming air. The indirect evaporative cooler, which reduces both temperature and enthalpy, can be very effective in all but the most extreme conditions.

In this indirect evaporative-intake cooler, water flows down the outside of the air intake passages. As it flows down, outside air is drawn up over the water, causing evaporation and cooling. The cooled water cools the intake air passages and hence the intake air. Depending on the local climate, this kind of unit can reduce the peak mechanical refrigeration by 30% to 70% with a very low water and energy requirement from the indirect cooler. The performance may be improved even further if the relief air from the building is used as the air that passes over the evaporative surface (Fig. 4.13).
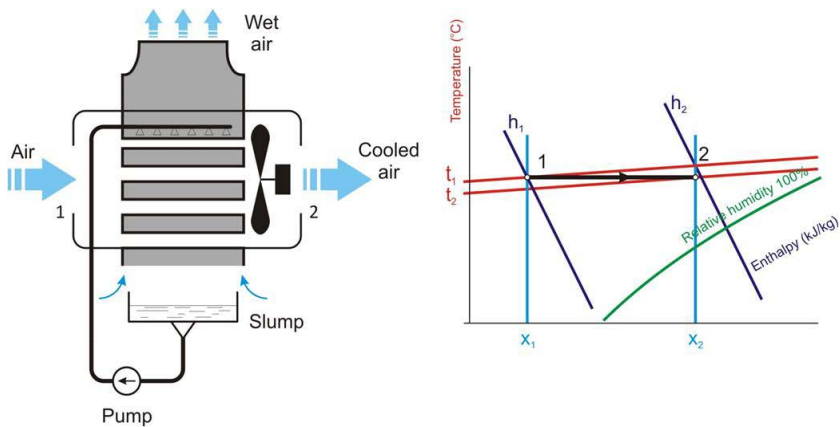

Figure 4.13. Indirect evaporative cooler.

4.6   Air duct

An optimal air duct system transports the required amount of conditioned, recirculated, or exhaust air to the specific space. An ideal design should bring an optimal duct layout, duct size, and

total pressure loss of the duct system. Air duct system design often requires comprehensive analysis and computer-aided calculating programs. Different air duct systems have different transport functions and thus have their own characteristics. It is difficult to combine the influences of cost, system balance, and noise together with duct characteristics into one or two representative indices.

Air ducts can be described as:

- Supply duct. Conditioned air is supplied to the conditioned space.
- Return duct. Space air is returned to the fan room where the air-handling unit is installed or to the packaged unit.
- Outdoor air duct. Outdoor air is transported to the air-handling unit, to the fan room, or to the space directly.
- Exhaust duct. Space air or contaminated air is exhausted from the space, equipment, fan room, or localized area.

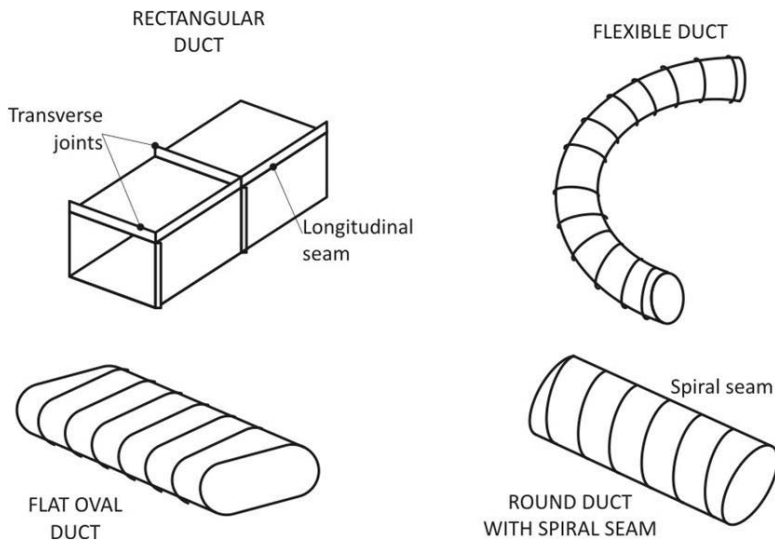Basic types of air duct shows Fig. 4.14.

Figure 4.14. Basic types of air duct.

The magnitude of the local loss coefficient of an elbow can be caused by turning the angle of the elbow, ratio of centerline radius to diameter, a three-gore, five-gore, or pleated seven-gore 90° elbow, installation of splitter vanes, and shape of cross-sectional area of the duct (Fig. 4.15).

Before an air duct system is designed, the supply volume flow rate for each conditioned space, room, or zone should be calculated, and the locations of the supply outlets and return inlets should also be settled according to the requirements of space air diffusion. For an air duct system, the supply volume flow rate of cold supply air in summer is usually greater than the warm volume flow rate needed in winter. If an air duct system conditions the space with cold air supply in summer, it often also conditions the space with warm air supply in winter.
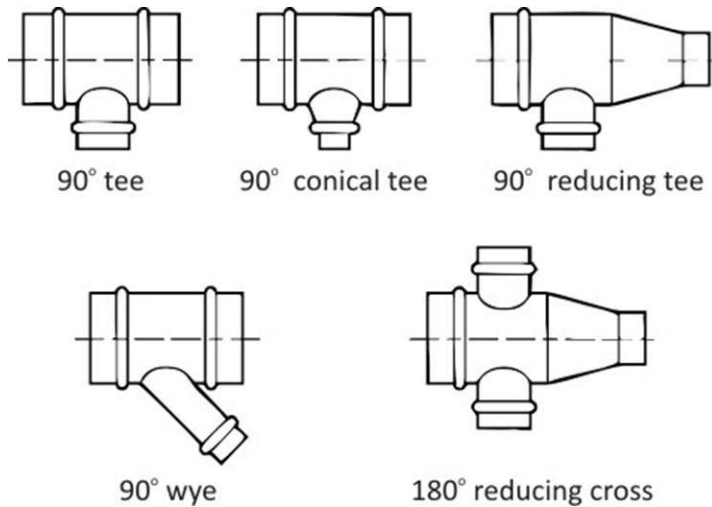
Figure 4.15. Round tees, wyes and cross used in air ducts.

Computer-aided duct design and sizing programs are widely used for more precise calculation and optimum sizing of large and more complicated duct systems. Manual air duct design and sizing are often limited to small and simple duct systems. The design procedure for an air duct system is as follows:

- The designer should verify local customs, local codes, local union agreements, and material availability constraints before proceeding with a duct design.
- The designer proposes a preliminary duct layout to connect the supply outlets and return inlets with the fan(s) and other system components through the main ducts and branch takeoffs. The shape of the air duct is selected. Space available under the beam often determines the shape of the duct and affects the layout in high-rise buildings.
- The duct layout is divided into consecutive duct sections, which converge and diverge at nodes or junctions. In a duct layout, a node or junction is represented by a cross-sectional plane perpendicular to airflow. The volume flow rate of any of the cross sections perpendicular to airflow in a duct section remains constant. A duct section may contain one or more duct segments (including duct fittings). A duct system should be divided at a node or junction where the airflow rate changes.
- The local loss coefficients of the duct fittings along the tentative critical path should be minimized, especially adjacent to fan inlets and outlets.
- Duct sizing methods should be selected according to the characteristics of the air duct system. The maximum design air velocity is determined based on the space available, noise, energy use, and initial cost of the duct system. Various duct sections along the tentative critical path are sized.
- The total pressure loss of the tentative critical path as well as the air duct system is calculated.

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- The designer sizes the branch ducts and balances the total pressure at each junction of the duct system by varying the duct and component sizes, and the configuration of the duct fittings.
- The supply volume flow rates are adjusted according to the duct heat gain at each supply outlet.
- The designer resizes the duct sections, recalculates the total pressure loss, and balances the parallel paths from each junction.
- The airborne and breakout sound level from various paths should be checked and the necessary attenuation added to meet requirements.

4.7 Heat transfer between the space air and the surroundings

- Space heat gain represents the rate at which heat enters a conditioned space from an external source or is released to the space from an internal source during a given time interval.
- Space cooling load is the rate at which heat must be removed from a conditioned space so as to maintain a constant temperature and acceptable relative humidity.
- Space heating load is the rate at which heat must be added to the conditioned space to maintain a constant temperature and sometimes specified relative humidity.
- Space heat extraction rate is the rate at which heat is actually removed from the conditioned space by the air system.
- Coil load is the rate of heat transfer at the coil.
- The heating coil load is the rate at which heat is added to the conditioned air from the hot water, steam, or electric heating elements inside the coil.
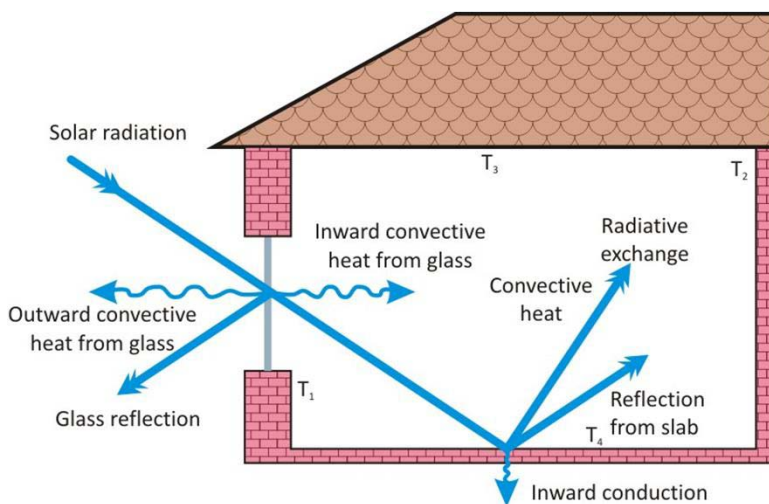- Refrigerating load is the rate at which heat is absorbed by the refrigerant at the evaporator.

Figure 4.16. Convective and radiant heat source.

Heat transfer calculation can be conducted as follows:
- Conduction heat gain through exterior walls and roofs

$$q_{e,t} = \left(\sum_{n=0} b_n T_{sol,t-n\Delta} - \sum_{n=1} d_n \frac{q_{e,t-n\Delta}}{A} - T_r \sum_{n=0} c_n \right) A \tag{4.3}$$

- Heat gain through ceilings, floors, and interior partition walls

$$q_{p,t} = UA\left(T_{ad} - T_r\right) \tag{4.4}$$

- Solar heat gain and conductive heat gain through window glass

$$q_{so,t} = A_{s,t}\left(SC\right)\left(SHGF_t\right) + A_{sh,t}\left(SC\right)\left(SHGF_{sh,t}\right) \tag{4.5}$$

$$q_{win,t} = U_{win} A_{win}\left(T_{o,t} - T_r\right) \tag{4.6}$$

- Internal heat gain

$$q_{sp,t} = N_{p,t}\left(SHG_p\right) \tag{4.7}$$

$$q_{lp,t} = N_{p,t}\left(LHG_p\right) \tag{4.8}$$

$$q_{s,l} = 3.413 W_{lamp} F_{usl} F_{al} = 3.413 W_A A_{fl} \tag{4.9}$$

- Radiative and convective heat transfer from the lighting fixture downward directly into the conditioned space

$$q_{ld} = q_{s,l} - q_{lp} = \left(1 - F_{lp}\right) q_{s,l} \tag{4.10}$$

- Heat carried away by return air from the ceiling plenum

$$q_{ret} = 60 \dot{V}_r \rho_r c_{pa}\left(T_p - T_r\right) \tag{4.11}$$

- Heat transfer from the plenum air to the conditioned space through the suspended ceiling and heat transfer from the plenum air to the conditioned space through the composite floor:

$$q_{cl} = U_{cl} A_{cl}\left(T_p - T_r\right) \tag{4.12}$$

$$q_{fl} = U_{fl} A_{fl}\left(T_p - T_r\right) \tag{4.13}$$

- Heat gain from the electric lights that enters the conditioned space

$$q_{es,l} = q_{ld} + q_{cl} + q_{fl} \tag{4.14}$$

$$q_{lp} = q_{ret} + q_{cl} + q_{fl} \tag{4.15}$$

4.8   Heat recovery systems

The recovery of internal heat loads—such as heat energy from lights, occupants, appliances, and equipment inside the buildings can be considered as a real alternative way to operative cost reduction. Waste heat can also be recovered from the flue gas of the boiler, exhaust gas and water jacket of the engine that drives the HVAC&R equipment, especially engine-driven reciprocating vapor compression systems and from the exhaust air from air conditioning systems as well (Fig. 4.17).
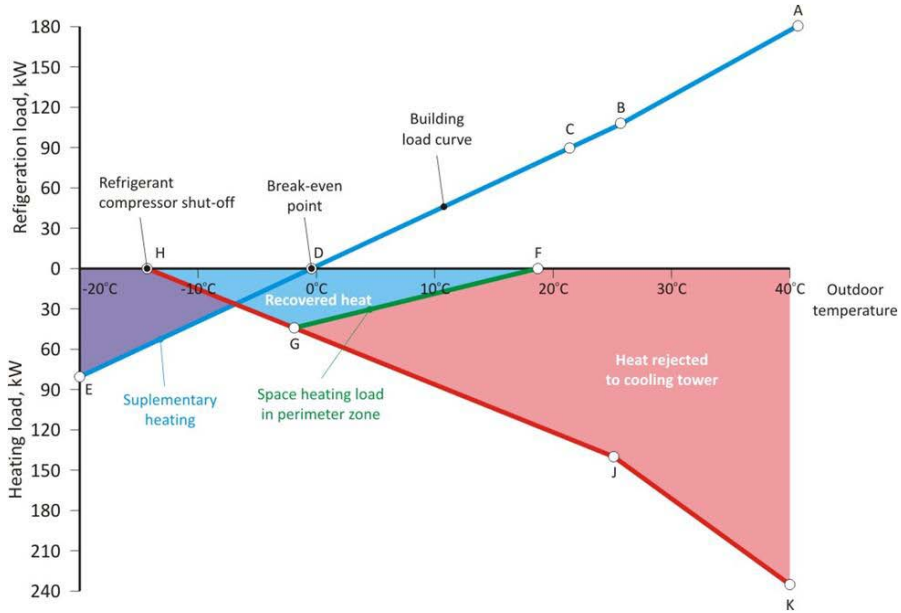

Figure 4.17. Operational modes of a heat recovery system.

It is always beneficial if the cooling effect of the exhaust air can be used to cool and dehumidify the incoming outdoor air during summer, and if the heating effect can be used to heat the cold outdoor air during winter.

Effectiveness can be written:

$$\varepsilon = \frac{\text{actual transfer}}{\text{maximum possible transfer between airstreams}} \tag{4.16}$$

so

$$\varepsilon_T = \frac{C_h\left(T_{he}-T_{hl}\right)}{C_{\min}\left(T_{he}-T_{ce}\right)} = \frac{C_c\left(T_{cl}-T_{ce}\right)}{C_{\min}\left(T_{he}-T_{ce}\right)} \tag{4.17}$$

Heating-cooling system

The sources of waste energy for the driving of water-ammonia refrigerant absorption systems should satisfy the following conditions:

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– temperature t ≥ 90°C – for fluid, t ≥ 150°C for gas;

– exergetic power Ex ≥ 30 kW;

– minimal chemical and mechanical pollutants;

– long and steady time of work τ ≥ 7000 hours/year.

– vapor and hot water – form of flows which have the pressure of 0.1 – 0.2 MPa and the temperature of 90 – 130°C;

– hot air and combustion gases from gas burning –relatively low pollutants rate. In most cases hot air occurs as a rather low exergetic medium: t = 30 – 90°C;

– combustion gases from the burning of solid fuels – high exergy, even up to 8 MW and temperature up to 600°C:

– metallurgic;

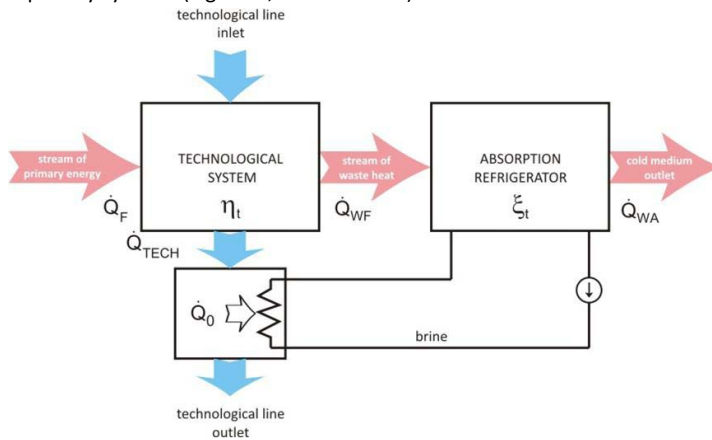– coking and pottery systems (Fig. 4.18, 4.19 and 4.20)

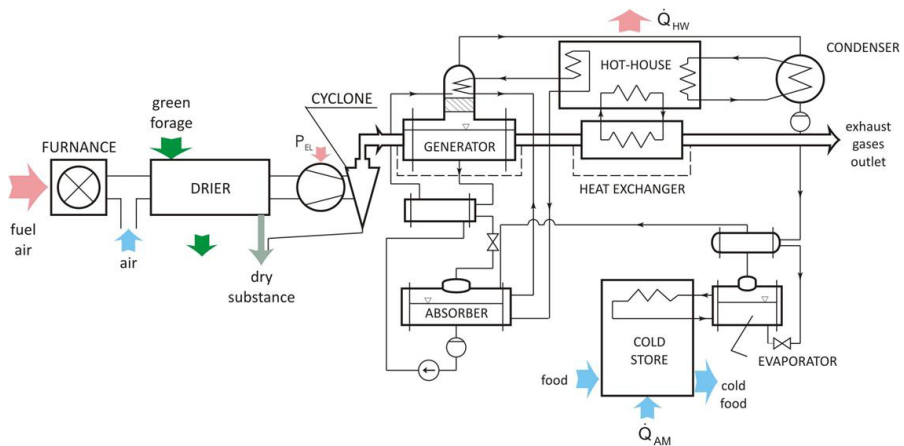Figure 4.18. Energy recovery from the technological system.

Figure 4.19. Coupled drying – cooling system.

Heat recovery effectiveness can be described as follows:

- The necessary condition for profitability of a given variation of a recovery system:

$$E > 0 \text{ and } E(I_{opt}) = \max \tag{4.18}$$

- Waste heat source exploitation (dimensionless parameters):

$$\alpha = \frac{Q_{WF} - Q_{WC}}{Q_{WF}} \tag{4.19}$$

$$\beta = \frac{\dfrac{Q_{TECH}}{\eta_A}}{Q_{WF} - Q_{WC}} \tag{4.20}$$

$$\gamma = \frac{Q_{AF}}{\dfrac{Q_A}{\eta_A}} \tag{4.21}$$

- Objective function E divided to a fuel cost (dimensionless form):

$$\varepsilon = \varepsilon(\alpha, \beta, \gamma, x_1, ... x_n) = \frac{1}{K_F} \cdot [\sum_i \Delta P_i(\alpha, \beta, \gamma, x_1, ... x_n) +$$
$$+ \sum_j \Delta k_j(\alpha, \beta, \gamma, x_1, ... x_n) - \sum_k z_k I_k(\alpha, \beta, \gamma, x_1, ... x_n)] \tag{4.22}$$

$$\varepsilon(\alpha_{opt}, \beta_{opt}, \gamma_{opt}, x_{1opt}, ... x_{nopt}) = \max \tag{4.23}$$
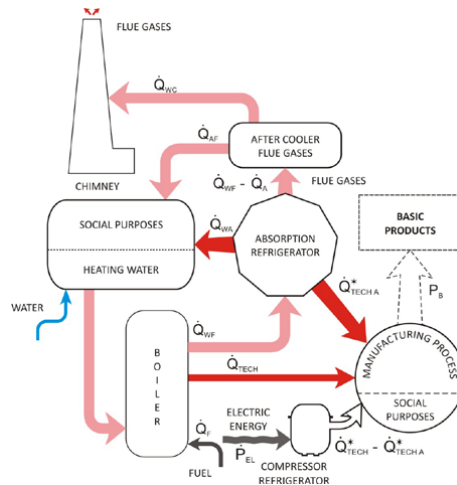

Figure 4.20. Exploitation of heat and energy for exemplary manufacturing process.

Cogeneration

Cogeneration is defined as the combined production of electrical power and useful thermal energy by the sequential use of a fuel. The production processes of the electric power and thermal energy are linked, and often are accomplished in a series or parallel fashion. Electrical power is the electricity produced by an electrical generator, which is most often powered by a prime mover such as a steam turbine, gas turbine, or reciprocating engine. Thermal energy is that product of the process which provides heating or cooling. Forms of this thermal energy include hot exhaust gases, hot water, steam, and chilled water (Fig. 4.21).
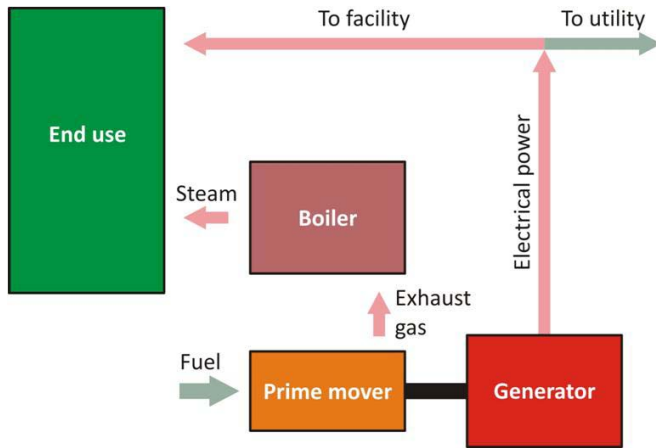
Figure 4.21. Cogeneration topping-cycle system.

The fuel inputs are:

$$\left(Fuel\right)_{conv} = \frac{\left(P+T\right)}{\left(\eta_0\right)_{conv}} \tag{4.24}$$

$$\left(Fuel\right)_{cogen} = \frac{\left(P+T\right)}{\left(\eta_0\right)_{cogen}} \tag{4.25}$$

The fuel savings is the difference between equation (4.24) and (4.25):

$$\left(Fuel\ savings\right) = \left(Fuel\right)_{conv} - \left(Fuel\right)_{cogen} \tag{4.26}$$

Cogeneration systems may involve different types of equipment and may be designed to satisfy specific needs at individual sites. On the other hand, many sites have similar needs and cogeneration systems may satisfy these needs and are more economical than custom-engineered systems. Cogeneration systems are often grouped into one of three sectors: industrial, institutional, and commercial (Fig. 4.22).
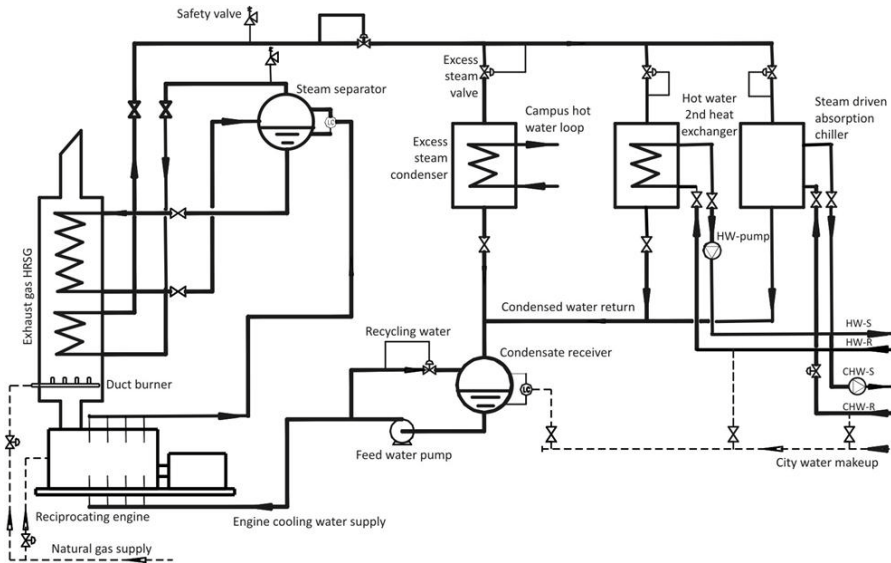
Figure 4.22. IC-engine-driven micro cogeneration system.

4.9 Air conditioning system main components

Air conditioning system, subsystems, and main components selection is performed mainly on six levels:

- air conditioning system;
- air system;
- water system;
- central plant cooling system;
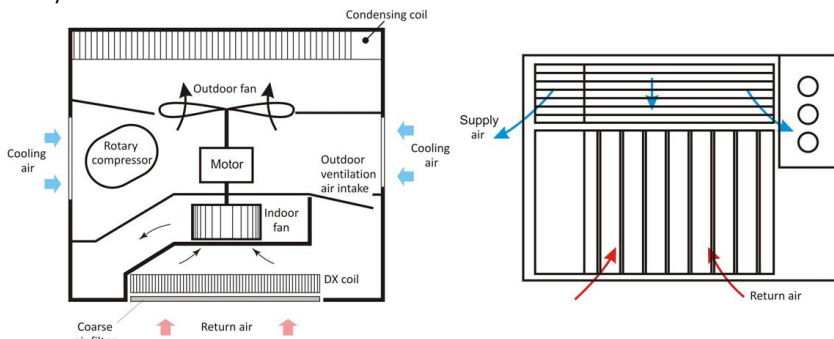- central plant heating system;
- control system.

Figure 4.22. Room air conditioning system.

Individual room air conditioning system uses a self-contained, factory-made packaged air conditioner to serve an individual room. It is ready to use after electric cable and necessary water

drainage are connected. Individual systems always use a DX coil to cool the air directly. Individual systems can be subdivided into the two following air conditioning systems:

- room air conditioning systems
- packaged terminal air conditioning systems

The differences in construction and operational characteristics between packaged and central systems can be described as:

– central system (CS) adopts chilled water as the cooling medium;

– in packaged system gas-fired furnace, electric heaters are often used to heat the air, and DX coils, often air-cooled, or sometimes evaporative condensers are used;

– in a packaged system, the controls of the heating and cooling system are often a part of the discharge air temperature control in the packaged unit. In a central system, there are separate water system controls, heating system controls, and refrigeration system controls in the central plant;

– in general, low- , medium- , and high-efficiency filters are used in packaged systems; and usually, medium- and high-efficiency filters are used in central systems;

– packaged units used in packaged systems are factory-fabricated and -assembled, whereas some components in an AHU may be custom-built in the field;

– the modulation of cooling capacity in a packaged system with scroll and reciprocating compressors is achieved by cycling of the cylinders or step controls;

– both the PS and the CS can provide the minimum ventilation air to dilute the air contaminants;

– both the PS and CS can provide the humidity control and prevent wet surfaces and mold growth in the ducts and conditioned space.

4.10 Ecological house

Ecological house is a building that sets up and operates itself. Has high-quality, personalized, and localized environmental control, including all aspects of the environment – heat, cooling, light, ventilation, and acoustics. It is also equipped with highly reliable, cost effective building services and environmental control — levels of reliability that far exceed today's at minimal costs, as automated learning systems that adapt to changing conditions are introduced into buildings. High speed multimode communication for voice, data, graphics, audio and video makes it flexible. Reconfigurable workspaces and services give efficient, robust, cost effective building operation.

Intelligent building technologies cover:

- advanced control system;
- automated diagnostic;
- intelligent environmental control system;
- flexible building system;
- smart windows;
- wireless controls.

Plug and play control concept reduces the manual labor in setting up control systems and crafting control algorithms. It ensures compatibility of control strategies with equipment characteristics and utilizes the most appropriate control strategies. It also provides a degree of standardization for control strategies and algorithms that assists with their maintenance, reduces callbacks by detecting errors at the time of installation and generally results in a higher quality product.

Integrating alternate generation technologies cover:

- fuel cells;
- microturbines;
- solar power;
- automated real-time energy purchasing capabilities;
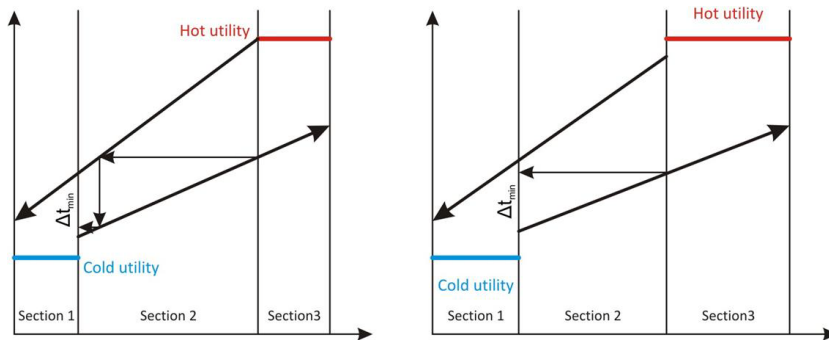- optimized dynamic building systems.

Figure 4.22. Effects of Δt_{min} on the energy in a heat exchange system.

Energy which is used in buildings should be utilized in the most wise way. The importance of energy efficiency have to be adjusted for:
- reviewing historical energy use;
- performing energy audits;
- identifying energy management opportunities;
- implementation changes to save energy;
- monitoring the energy management program, set goals, and review progress.

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For additional information and further study, following books are recommended. These are the books frequently referenced during the preparation of this document.

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