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5. HEAT PUMPS

5.1. Introduction

Geothermal energy is the heat from the Earth that can be recovered and exploited by man. Natural geothermal energy examples are hot springs, volcanoes and other thermal sources. Research showed that the Earth's temperature was increasing with depth, under a gradient of 2-3°C/100m. The total heat flux from the Earth's interior amounts to ca 80 (mW/m^2). It is a great resource, non-polluting, almost infinite source of clean and renewable energy. The heat originates from the Earth's core temperature (4,000°C at the depth of 4,000 km).

The total heat amount of the Earth is about 12.6×10^{24} MJ, and the total world energy demand is 6×10^{13} MJ. However, only a part of it can be used at present. Utilization of this energy has been limited to areas in which geological conditions allow (Kreider et al., 2010).

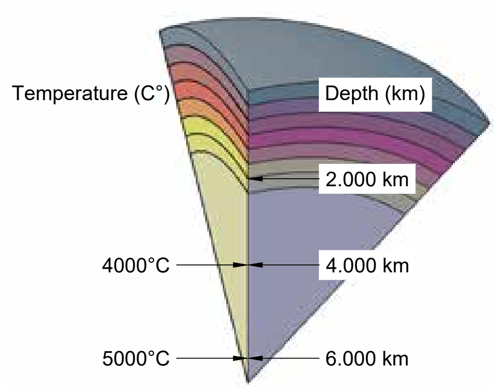

Fig. 5.1. The temperature inside the Earth (Source: own elaboration)

There are three common methods of heat generation. The first is radioactive decay reaction, i.e. the heat from the process reaches the Earth's surface. The second is the transmission of thermal energy from the depth of the Earth, transporting it through several layers to reach the surface. There are several areas where direct channels bring molten rock and steam to the surface. These direct channels are known as high temperature geothermal heat sources and can be used for electricity generation. The last of the heat generation methods is solar radiation. The earth's crust absorbs approximately 47% of solar radiation, making it a very low-cost energy source. By some estimates, this low-cost energy geothermal source produces 500 times more energy than humanity uses in a year. The temperatures inside the Earth can be seen in Fig. 5.1.

Geothermal energy is classified according to the temperature: low (<150°C) or high temperature (>150°C). For heating, low temperature heat sources are used. High temperature geothermal energy is utilised for electricity generation. There are four types of geothermal power plants: dry steam power plants, flash power plants, binary power plants and combined flash-binary power plants.

Low temperature heat sources are commonly used for heating. The most useful are three possibilities. The first, direct heating, is one of the oldest and most common ways of geothermal energy use for building heating, e.g. in aquaculture, horticulture, balneological industry and swimming pools. The system is cost-effective, the main costs are generated by well-drilling and installation of transmission pumps, pipelines and other down-hole equipment (Kreider et al., 2010).

The second one is absorption machines technology. These machines are well known and used for heating and cooling. The absorption cycle is a process that uses heat energy instead of compressor work. In the cycle, a mixture of two fluids is used: a refrigerant, which evaporates and condenses, and a secondary fluid as an absorbent. For a cycle above 0°C, lithium bromide is used as the refrigerant and water as the absorbent, whereas for applications below 0°C an ammonia and water mix is used. Geothermal fluids run these machines. The geothermal effluent is usually expelled back into the reservoir from which it was taken to avoid the possible negative effect of the discharge on the environment.

The third possibility is a heat pump, used to transfer low-temperature heat stored in the earth to buildings.

The annual high temperature variability of the ground at the depth of 1.5 m, is nearly unnoticeable at depths of 18 m and more (Fig. 5.2).

The ground-source heat pump (GSHP) is able to use the low grade geothermal energy to heat and cool homes, schools, governmental and commercial buildings. The heat pump uses a relatively small amount of electricity to transport four times more heat energy to heat consumers. The first heat pump principle was described by Lord Kelvin in 1852. Robert Webber from Indianapolis modified the GHSP in 1945. It was first applied in the USA in 1960 and in Europe it has been used since 1970 (Oughton & Wilson, 2015).

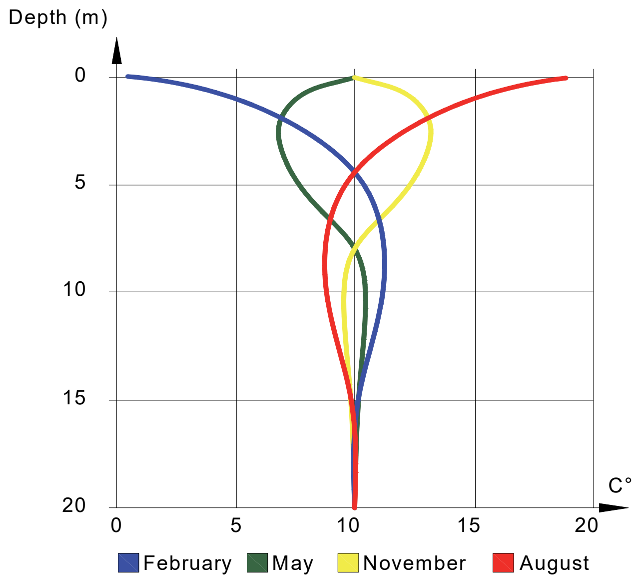

Fig. 5.2. Earth's surface temperature in moderate climate (Source: own elaboration)

5.2. Thermodynamic principles of the cyclic process

Heat pumps are devices which transfer heat from one place to another. Heat pumps can carry heat from low temperature surroundings to a higher temperature building. Heat pump theory is based on the second law of thermodynamics, which says that heat passes from higher temperature body to lower temperature body. This means that if the heat transfer collector is kept at a lower temperature than its surroundings, it will absorb heat from the environment.

An anti-clockwise cycle transports heat from a low temperature level to a high temperature level. A compression heat pump system operates based on the principle of the thermodynamic cycle. As a mechanical device, the heat pump can provide energy for space heating or cooling.

The fundamental principle of the heat pump circuit is the dependence of the boiling temperature of a liquid on the pressure. As we know, in the anti-clockwise cycle heat absorption must occur at a lower temperature and heat emission at a higher temperature. To allow this, the pressure must be reduced so that the evaporation temperature is lower than the heat source temperature and then raised up to a level at which the boiling temperature is above the temperature of the surroundings (Eicker, 2014).

5.3. Functioning and components of compression heat pump system

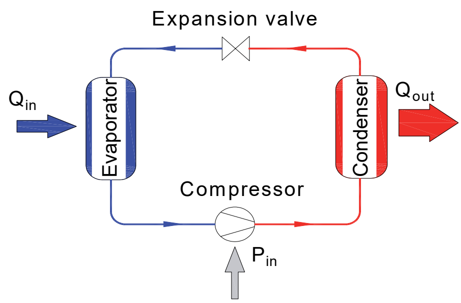

Fig. 5.3. The energy flow in a heat pump system (Source: own elaboration)

As it was mentioned above, the task of a compression heat pump is to transport heat energy from a low temperature level to a high temperature level (Fig. 5.3). The fluid used to heat the pump cycle is called a refrigerant. The refrigerant is a liquid or gaseous substance that circulates through the heat pump in turns absorbing, transporting and discharging heat. This can be achieved using the anti-clockwise cycle (Eicker, 2014).

The system must satisfy the first law of thermodynamics, which says that all energy supplied must be equal to the energy discharged, and this relationship can be expressed by the following Eq. (5.1):

$$Q_C = Q_0 + P_{in} \tag{5.1}$$

where:
Q_c – total output of the condenser (kW),
Q_0 – total output of the evaporator (kW),
P_{in} – total compressor output (kW).

Calculations for a heat pump system are performed using the p-h diagram. The temperatures and pressures measured in a system can be plotted in the diagram and used to determine corresponding enthalpy, results are presented graphically. The theoretical calculation is extremely difficult, depending on very complicated relationships between thermal and calorific state variables for actual substances, which can be derived from Gibbs fundamental equations (Oughton & Wilson, 2015). The p-h diagram is an important tool, it means that calculation is not necessary. The specific enthalpy h is on the horizontal axis in the diagram. The pressure p is on the vertical axis. The cycle can be divided into 4 stages, which can be seen in the p-h diagram (Fig. 5.4).

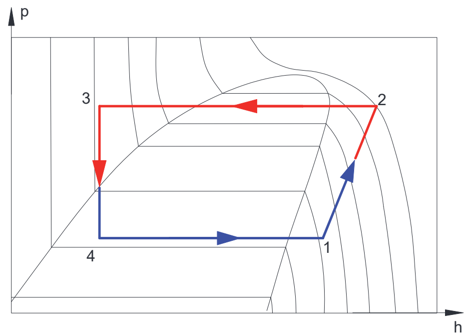

Fig. 5.4. Heat pump process in a *p-h* diagram, where: 1-2 – isentropic compression, 2-3 – isobaric condensation, 3-4 – isenthalpic regulation, 4-1 – isobaric evaporation (Source: own elaboration)

5.3.1. First stage – Isentropic compression

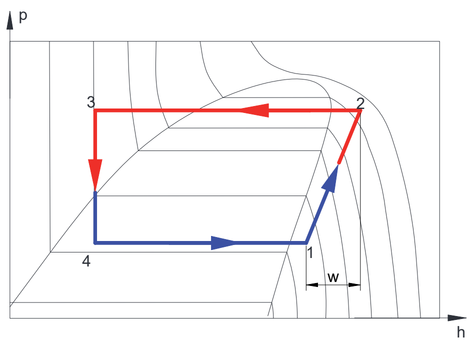

Fig. 5.5. Heat pump compression in a *p-h* diagram (Source: own elaboration)

At the first stage (Fig. 5.5), the refrigerant is compressed by a compressor from the evaporation pressure to the condensation pressure. In ideal conditions, compression in the isentropic energy exchange does not take place during this process. The enthalpy increase is equivalent to the amount of work added. The compressor output can be expressed by Eq. (5.2):

$$P_{in} = (h_2 - h_1) \cdot m_R \tag{5.2}$$

The specific compressor work is calculated by Eq. (5.3):

$$w = P_{in}/m_R = h_2 - h_1 \tag{5.3}$$

where:
P_{in} – compressor output (kW),
h_1 – refrigerant enthalpy before compression (kJ/kg),
h_2 – refrigerant enthalpy after compression (kJ/kg),
m_R – refrigerant mass flow rate (kg/s),
w – specific compressor work (kJ/kg).

The compressor compresses the refrigerant and discards it at high pressure. During compression, the internal energy and temperature of the gas increases.

The isentropic compression cannot be achieved in a real process. This is because the internal friction of the fluid is never reduced to zero, and also because of the inevitable transfer of heat through the cylinder wall.

5.3.2. Second stage – Isobaric condensation

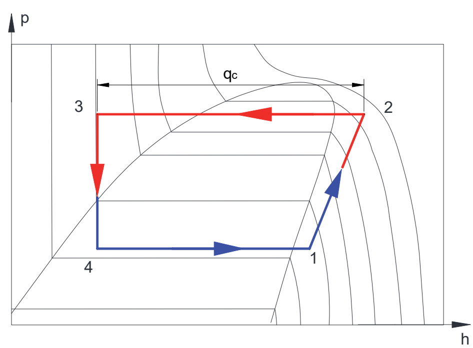

Fig. 5.6. Heat pump condensation in a *p-h* diagram (Source: own elaboration)

In isobaric condensation, the heat energy (Fig. 5.6) is discharged to the environment by condensation of the refrigerant. The total output of the condenser can be determined from the p-h diagram and expressed by Eq. (5.4):

$$Q_c = (h_3 - h_2) \cdot m_R \tag{5.4}$$

The specific condenser output rate is calculated by Eq. (5.5):

$$q_c = Q_c/m_R = (h_3 - h_2) \tag{5.5}$$

where:
Q_C – condenser output (kW),
h_2 – refrigerant enthalpy before condenser (kJ/kg),
h_3 – refrigerant enthalpy after condenser (kJ/kg),

m_R – refrigerant mass flow rate (kg/s),
q_c – specific condenser output (kJ/kg).

The compressed, superheated refrigerant vapour supplies the condenser with the temperature t_2 and the condensation pressure p_c. The refrigerant then passes through three stages:

1. Heat dissipation zone – the discharged heat flow is absorbed by the surroundings or by a cooling fluid. In this zone the refrigerant is cooled until it reaches the dew point; the zone accounts for 5-15% of the total heat transfer of the condenser.
2. The condensation zone accounts for around 85-90% of the total heat transfer of the condenser.
3. In the supercoiling zone, the completely condensed refrigerant is super cooled in the temperature of 5°C. Supercooling guarantees that there is no gas before the expansion valve, increases the capacity on the evaporator side and improves the coefficient of the heat pump performance.

5.3.3. Third stage – Isenthalpic throttling

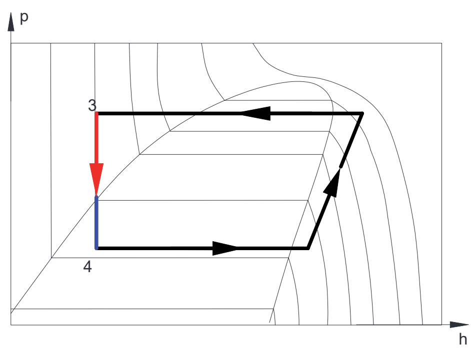

Fig. 5.7. Heat pump throttling in a *p-h* diagram (Source: own elaboration)

Completely condensed refrigerant must be supplied to the evaporator at low pressure. This occurs during an isenthalpic change of state during throttling (Fig. 5.7).

'Isenthalpic' means that neither heat nor work is exchanged with the environment. In a real process, entropy increases as the energy is dissipated inside the throttle during the process. A certain proportion of the refrigerant is evaporated during the throttling process. The value of this proportion can be identified from p-h diagrams. They contain lines that show the constant vapour content x in the wet steam area. This basically depends on the evaporation temperature and the supercooling of the refrigerant in the condenser.

5.3.4. Fourth stage – Isobaric evaporation

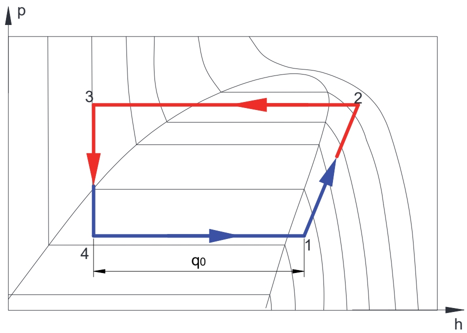

Fig. 5.8. Heat pump evaporation in a *p-h* diagram (Source: own elaboration)

The liquid refrigerant enters the evaporator, where it starts boiling (Fig. 5.8). The heat energy is taken from the surroundings or heat medium. The total output of the evaporator can be determined from the log p-h diagram and expressed by Eq. (5.6):

$$Q_o = (h_1 - h_4) \cdot m_R \tag{5.6}$$

The specific refrigerating capacity is calculated by Eq. (5.7):

$$q_o = Q_o/m_R = h_1 - h_4 \tag{5.7}$$

where:

Q_O – evaporator output (kW),
h_2 – refrigerant enthalpy before evaporator (kJ/kg),
h_3 – refrigerant enthalpy after evaporator (kJ/kg),
m_R – refrigerant mass flow rate (kg/s),
q_o – specific evaporator output (kJ/kg).

5.3.5. Determining the output coefficient from a *p-h* diagram

In order to compare different heat pumps, an output coefficient is used. It is consistent with the engine performance and is equal to the ratio between work and condenser output. The amounts of energy converted in the cyclic process can be taken directly as enthalpy differences from the p-h diagram. Thus, the output coefficient for the ideal process can be expressed by Eq. (5.8):

$$\varepsilon = \frac{Q_c}{P_{in}} = \frac{h_2 - h_3}{h_2 - h_1} \tag{5.8}$$

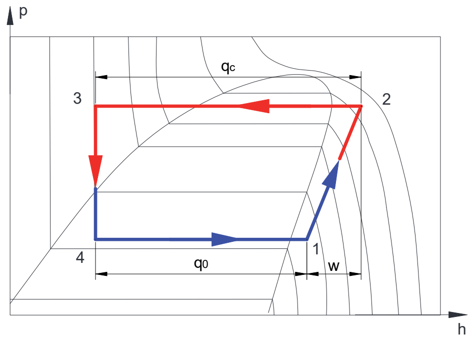

Fig. 5.9. Heat pump output coefficient in a *p-h* diagram (Source: own elaboration)

The output coefficient increases as the temperature difference between the inlet and outlet part decreases.

5.3.6. Real heat pump cyclic process in a *p-h* diagram

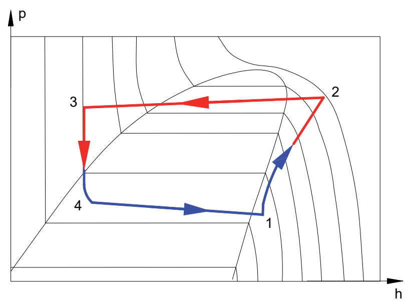

Fig. 5.10. Real heat pump cyclic process in a p-h diagram (Source: own elaboration)

The descriptions till now have been focused on an ideal heat pump process. In an ideal process, where t is assumed that no losses take place in the system. In practice, this is not possible. In a real cycle, the isentropic compression can only be approximately achieved heat exchange with the environment (Fig. 5.10). This also applies to the throttling process, which is not isenthalpic. During the evaporation and condensation pressure losses are inevitable (Eicker, 2014).

5.4. Compressors

A compressor is one of the main components of a heat pump system. Its task is to supply mechanical energy for refrigeration cycle performance. The compressor takes in gaseous refrigerant at low pressure and temperature, compresses it at increased temperature and discards it again at high pressure (Kreider et al., 2010).

Compressors are classified according to pressure, compressible medium, environmental conditions, structure and type.

There are two basic compressor types – positive displacement compressors and dynamic type compressors.

Positive displacement compressors operate by pressing a fixed volume of fluid from the inlet pressure section of the compressor into the discharge zone of the compressor. Mechanical pistons can be single acting, double acting or of a diaphragm type. In the rotary compressor, a fixed amount of fluid is transferred with each revolution. Rotary compressors have two revolving elements, like wheels between which the refrigerant is compressed. These compressors are very efficient because the actions of taking in the refrigerant and compressing it take place at the same time. These compressors are divided into: lobe, liquid ring, vane, scroll, screw and centrifugal compressors (Fig. 5.11).

The screw compressor uses a pair of screw rotors which operate together to compress the refrigerant between them.

The scroll compressor uses two intermeshed spiral discs operating together to compress the refrigerant. The upper disc is stationary while the lower-one moves in an orbital way.

The rotary lobe compressor: inside the pump housing there are two rotors, each has two or three lobes. One of the rotors is driven by the motor and the other is geared to the driven one. When one revolves, the other revolves in the opposite direction.

The liquid ring compressor has a rotor with several fixed blades that rotate eccentrically in a cylinder. The cylinder is partially filled with low viscosity liquid. Due to the centrifugal force, the liquid forms a solid ring around the inside of the cylinder. The liquid ring isolates each cell from another and the volume of the cells decreases as the rotor rotates, compressing the gas until it is finally removed through the outlet.

The unbalanced vane compressor contains spring-loaded vanes placed in the slots of the rotor. The compressing action takes place due to the motion of the vanes along a cam ring. The rotor is eccentric to the cam ring. When the rotor rotates, the vanes follow the inside surface of the cam ring. The space between the vanes is reduced near the outlet due to the eccentricity. This force compresses the fluid.

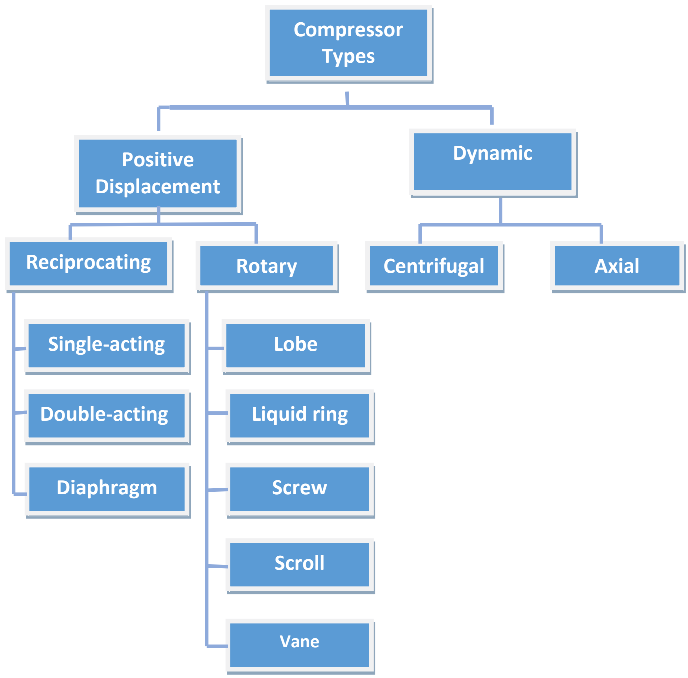

Fig. 5.11. Compressor classification (Source: own elaboration)

Centrifugal compressors use the rotating movement of an impeller wheel to create the centrifugal force. A diffuser converts the velocity energy to pressure. Centrifugal compressors are well adjusted to compress large volumes of refrigerant to a relatively low pressure. The compressive pressure generated by the impeller wheel is small, therefore centrifugal compressors usually use two or more stages in a series to generate high compressive forces.

The axial flow compressor is a dynamic rotating compressor continuously accelerating the fluid. The fluid flows in an axial direction through a series of rotating blades and stationary vanes that are concentric with the axis of rotation. The flow trajectory decreases in a cross-sectional area in the direction of the flow. This decreases the volume of the air as compression increases gradually. Air pressure rises each time it passes through a unit of rotors and stators (Althouse et al., 2013).

5.5. Condenser

The task of a condenser is to release the heat flows absorbed in the evaporation process to the heat user. This is the heat flow absorbed on the evaporator side and the compressor power, which during compression was transferred to the cooler as a heat flow. The heat flow to be discharged on the condenser can be calculated by Eq. (5.9):

$$Q_c = Q_o + P_{in} \tag{5.9}$$

where:

Q_c – total output of the condenser (kW),
Q_o – total output of the evaporator (kW),
P_{in} – total compressor output (kW).

To be able to release the absorbed heat flow, the surface temperature of the condenser must be higher than the temperature of the heated building (Fig. 5.12).

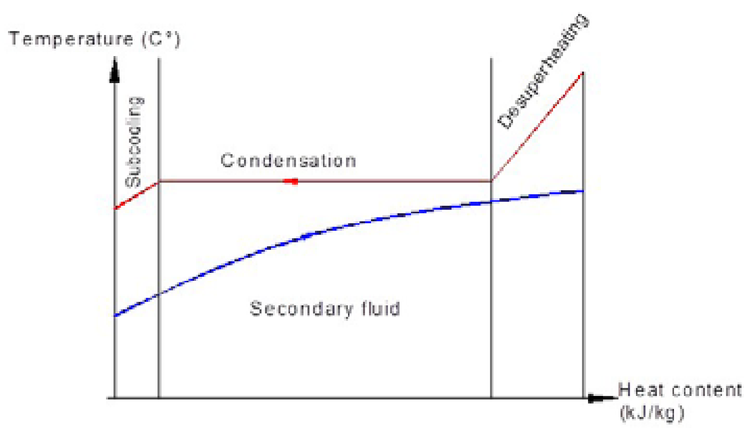

Fig. 5.12. Condensation process (Source: own elaboration)

Heat dissipation zone: The superheated, gaseous refrigerant cools at a constant pressure p_c from the superheated temperature to the condensation temperature t_c. The heat flow released is absorbed by the fluid from the heated building. That heat zone accounts for 5-15% of the total heat transmission of the condenser.

Condensation zone: Condensation takes place in this zone. The saturated refrigerant vapour condenses at a constant pressure p_c and a constant temperature t_c while releasing latent heat. The condensation zone accounts for around 85-90% of the total heat transmission of the condenser.

Supercooling zone: In the supercooling zone, the completely condensed refrigerant is then supercooled to around 5°C. Supercooling guarantees that there are no bubbles before the expansion valve and increases the process capacity on the evaporator part, thus upgrading the coefficient of performance of the whole system (Kreider et al., 2010).

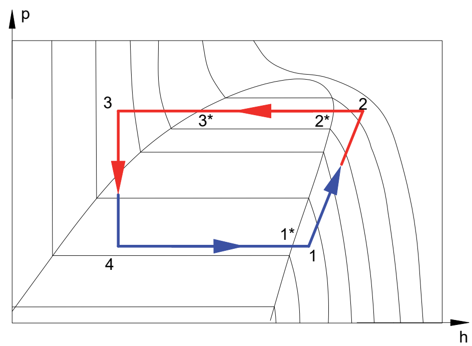

Fig. 5.13. Condensation process (Source: own elaboration)

Fig. 5.13 shows the condensation process in a p-h diagram:

- line 2-2* indicates the heat dissipation zone,
- line 2*-3* indicates the condensation zone,
- line 3*-3 indicates the supercooling zone.

The output of the condenser depends on the area of the condenser surface, the heat transfer coefficient and the temperature difference. These parameters can be used to calculate the output of a condenser by Eq. (5.10):

$$Q_c = A \cdot k \cdot \Delta t_m \tag{5.10}$$

where:

Q_c – total output of the condenser (kW),
A – the area of the condenser surface (m^2),
k – coefficient of heat transmission (W/m^2K),
Δt_m – logarithmic temperature difference (K).

The above formula can only be used to perform the calculation for liquid-cooled condensers which operate in a uniflow mode. However, the formula cannot be used for the condensers operating in a cross flow mode, nor for the air-cooled condensers.

The k coefficient of heat transfer indicates the amount of heat that can flow through the condenser per second and per square meter at a temperature difference of 1K. The k value is qualified in the plant, for each condenser. In practice, the coefficient of heat transmission can be reduced due to soiling on the surface of the condenser. If the heat transmission goes down, the temperature difference between the condensation temperature and the fluid temperature rises. As the temperature of the heating is constant, the condensation temperature must be higher. The compressor must operate at a higher pressure ratio when its volumetric efficiency is reduced and creates a risk of overload. To avoid this, the condenser must be constantly cleaned of dirt and soiling.

The logarithmic temperature difference Δt_m between the condensation temperature and the temperature of the heated building is used to calculate the condenser output. The temperature difference between the cooling medium and the refrigerant changes depending on the distance of the flow, but this change is not linear and cannot be used for the calculation.

The temperature difference can be described by Eq. (5.11):

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\ln\left(\frac{\Delta t_1}{\Delta t_2}\right)} \tag{5.11}$$

Neither heat dissipation nor supercooling is taken into account. The logarithmic temperature difference is therefore the basis for calculations in the condensation area.

The heat flow to be discharged by the condenser is made up of the compressor power intake and the heat resources capacity. If this data is unknown, the condenser output can be recorded by measurement. For the underfloor heating (water cooled) condenser, the output is calculated by Eq. (5.12):

$$Q_c = V_F \cdot \rho_F \cdot c_F \,(t_{F1} - t_{F2}) \tag{5.12}$$

where:
Q_c – total output of the condenser (kW),
V_F – underfloor heating water flow rate (m^3/s),
ρ_F – underfloor heating water density (kg/m^3),
c_F – underfloor heating water specific heat capacity (kJ/(kg·K)),
t_{F1} – underfloor heating water temperature at condenser inlet (°C),
t_{F2} – underfloor heating water temperature at condenser outlet (°C).

Measuring the output of an air-cooled condenser is similar. It should be noted that the specific heat capacity and density of the air depends on the temperature. Values can only be constant for small temperature differences. The condenser output is calculated using the air flow rate by Eq. (5.13):

$$Q_c = V_A \cdot \rho_A \cdot c_A \,(t_{A1} - t_{A2}) \tag{5.13}$$

where:

Q_c – total output of the condenser (kW),
V_A – air flow rate (m^3),
ρ_A – air density (kg/m^3),
c_A – air specific heat capacity (kJ/kg·K),
t_{A1} – air temperature at condenser inlet (°C),
t_{A2} – air temperature at condenser outlet (°C).

Usually it is difficult to determine the air flow rate because it requires recording the air speed. The air flow rate can be calculated by measuring the dynamic air pressure. The air is normally non homogeneous, temperature and speed distributions from the heat exchanger are also difficult to measure. The real output can be calculated only by measurement of liquid cooling medium (Kreider et al., 2010).

5.6. Throttle valves

A throttle valve reduces the freon pressure from the condenser to the evaporation pressure. The low pressure is required so that the refrigerant can evaporate at low temperatures. It is an important significance in terms of optimum system recovery that the evaporator is always supplied with as much refrigerant as it can process in the correct operating conditions. If too much freon reaches the evaporator, the liquid will not be fully evaporated and liquid lock can occur in the compressor.

If the evaporator is provided with too little refrigerant, it will not operate correctly and excessive overheating of the refrigerant can take place, which will result in too high a final compression temperature.

The change of state in the throttle element can be viewed as isenthalpic. Practically, there is a slight increase in enthalpy in the valve (Althouse et al., 2013).

Expansion valves do not control the evaporation temperature straightforwardly. They regulate the superheating by controlling the mass flow of freon into the evaporator and upholding the pressure difference between the condenser (high pressure) and the evaporator (low-pressure). The evaporation temperature depends on the capacity of the compressor and efficiency of the evaporator.

Throttle elements are divided into three categories depending on their regulation of freon mass flow rate:

1. non-regulating throttle elements,;
2. mechanical self-regulating throttle elements,
3. electric self-regulating throttle elements.

Non-regulating throttle elements or capillary tubes are the simplest of all freon flow controls. They contain no moving parts; normally they consist only of a copper pipe with the diameter of 0.5 to 1.5 mm and the length from 1.5 to 4 m. The fall in the pressure of the refrigerant takes place due to the long and narrow tube. The mass flow through the tube depends on the pressure difference between the condensing and evaporating sides. Capillary tubes can be set up on small and high-volume commercial systems if the operating conditions are relatively fixed. Often, the low pressure side has a liquid separator that acts as a receiver in the frontal part of the compressor.

Mechanical self-regulating throttle elements:

A pressure control throttle valve (Fig. 5.14) is a manually operated needle valve. The needle position is fixed, and the mass flow through it depends on the pressure difference over the valve. A manual throttle is a non-regulating valve and it should not be used as an expansion valve with a BPHE (Brazed Plate Heat Exchangers) evaporator, because some changes in functioning conditions would immediately change the evaporation process inside the plate heat exchanger. Regulating the flow manually would require direct regulation, which is not practically executable.

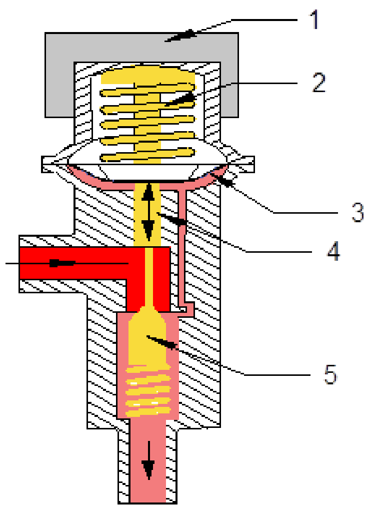

Fig. 5.14. Pressure control throttle valve (Source: own elaboration)

Thermostatic expansion valves (TEVs) are the expansion devices used most commonly with BPHE evaporators. TEVs are good expansion devices due to their relatively good sensitivity in settings. The large choice of expansion valve sizes, capacity and temperature ranges is very positive. The main disadvantage of TEVs is the requirement for a relatively high superheating**.**

There are two different types of thermostatic expansion valves with internal pressure compensation (Figs. 5.15 and 5.16) with external pressure compensation (Fig. 5.17).

(TEV) with internal pressure compensation:

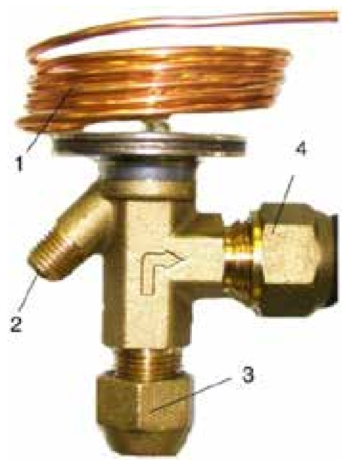

Fig. 5.15. Thermostatic expansion valve (Source: own elaboration)

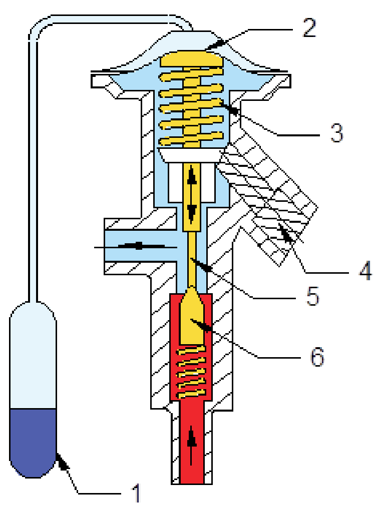

Fig. 5.16. Thermostatic expansion valve with internal pressure compensation – cross-section: 1 – temperature sensor, 2 – membrane, 3 – membrane spring, 4 – ajustting screw, 5 – tappet, 6 – nozzle insert with valve cone (Source: own elaboration)

The valve consists of: (Fig. 5.16, pos. 1) a temperature sensor which is installed at the evaporator outlet; a valve cone, opening and closing valve (Fig. 5.16, pos. 6); a regulating spring (Fig. 5.16, pos. 3) which increases the force on the valve cone; and a regulating screw which can be used to adjust the spring resistance and thus the superheating. A membrane (Fig. 5.16, pos. 2) is used to transfer the pressure force. The functioning of the valve is determined by three forces: sensor pressure in a temperature sensor (Fig. 5.16, pos. 1) which generates force in the valve (Fig. 5.16, pos. 6) opening direction at the membrane (Fig. 5.16, pos. 2). The pressure in temperature sensor is mainly determined according to the temperature of the evaporator outlet. The spring force (Fig. 5.16, pos. 3) and the force generated by the evaporation pressure at the membrane act in the closing direction of the valve (Fig. 5.16, pos. 6).

When these three forces are in balance, the opening cross-section of the valve remains constant. If the evaporator is supplied with too little refrigerant, the temperature at the evaporator outlet increases the gas pressure in the temperature sensor (Fig. 5.16, pos. 1) as well. The result of this is that the opening cross-section in the valve increases and more refrigerant is injected into the evaporator, which means that the superheating is reset. If the compressor shuts down, the evaporation pressure increases and the valve cone (Fig. 5.16, pos. 6) closes the valve. If greater pressure losses take place between the evaporator inlet and outlet, the evaporation pressure at the membrane (Fig. 5.16, pos. 2) is higher than the pressure at the evaporator outlet. The lower evaporation temperature at the outlet results in a lower refrigerant temperature here. This lower refrigerant temperature causes a lower sensor pressure (Fig. 5.16, pos. 1) and the valve is half closed. This reduces the freon flow and increases the superheating. The superheating increases with the pressure loss in the evaporator (Althouse, 2013).

(TEV) with external pressure compensation:

This type differs from a TEV with internal pressure compensation (Fig. 5.16) in that the valve outlet and the membrane surface (Fig. 5.16, pos. 2) are separated from one another by a divider, usually a gland. This means that the evaporator inlet pressure is no longer present at the membrane (Fig. 5.17, pos. 2). The evaporator outlet and the space below the membrane are connected with a pressure line. The superheating is not affected by the pressure drop in the evaporator. The evaporator outlet pressure is now present at the membrane (Fig. 5.17, pos. 2) as the control pressure instead of the evaporator inlet pressure (Althouse, 2013).

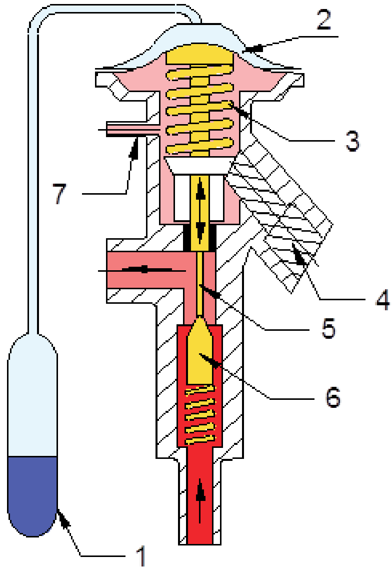

Fig. 5.17. Thermostatic expansion valve with external pressure compensation – cross-section. 1 – temperature sensor, 2 – membrane, 3 – membrane spring, 4 – ajustting screw, 5 – tappet, 6 – nozzle insert with valve cone (Source: own elaboration)

Throttle valves with electrical auxiliary energy. Electronic expansion valves technology has been developed to deal with increasingly complicated requirements. The cost of electronic valves, which includes the regulator, sensors, actuator and the valve, is still high. Electronic expansion valves are therefore mostly found on very large systems. Electronic expansion valves are controlled by temperature or pressure sensors. The electronic regulating unit can be programmed to correct differences in temperature and pressure at any point of the system. Because the electric actuator reacts only to signals from the regulator, there are good possibilities for achieving a lower level of superheating than with a thermal expansion valve (Hadorn, 2015).

Thermo-electric throttle valves are equipped with a bimetallic strip. It contains two metal layers with different heat transfer coefficients. If the band is heated, it deforms in proportion to the temperature change. Heating is carried out using a heating wire wound around the strip. In the vapour intake line behind the evaporator, there is a thermistor. The capacity of the expansion valve (the amount of refrigerant flowing through it) is determined by the relationship between the opening and closing times. A regulator controls the opening and closing of the valve in order to reach the correct level of superheating. The inputs to the regulator are the temperature and pressure at the evaporator outlet. The inputs could also be the inlet and outlet evaporator temperatures, as for an electronic valve with a continuous control.

5.7. Evaporator

The task of the evaporator is to withdraw heat energy from the heat source by a heat medium. Heat transmission can only take place if there is a temperature difference between the heat medium and the refrigerant. Temperature difference of around 10°C is enough to guarantee heat transmission. The liquid freon enters the evaporator after throttling. The liquid evaporates completely as it passes through the evaporator. During this process, the freon absorbs heat energy from the heat medium.

Depending on the heat source, the heat medium evaporator can be: air-cooled, liquid-cooled and flooded.

Air-cooled evaporators finned tube coil-pipe systems are used as evaporators with the freon flowing through the tubes. The purpose of the fins is to increase the evaporator surface area on the air side. In air-cooled evaporators, the heat transmission on the freon side is higher than on the air side.

Liquid evaporators combine good heat transmission with a compact design. At low outputs, plate and coaxial heat exchangers are used. The refrigerant can either be evaporated in the tubes or flooded in the lateral surface.

Flooded evaporator operates in connection with a receiver. The receiver acts as a separator of gaseous and liquid refrigerant behind the expansion valve and provides 100% of the liquid refrigerant flow to the evaporator. Direct expansion evaporator and the refrigerant are not fully evaporated and superheated at the flooded evaporator outlet. The leaving refrigerant flow is a two-phase mixture containing typically 50-80% of gas (Hadorn, 2015).

5.8. Other components

Filter-dryer – Moisture (water) in the system may freeze the refrigerant control valves. This may clog or partially jam the control. Humidity in some refrigerants at high compressor temperatures can cause decay of the refrigerant to form harmful acids. It may lead to corrosion, or oil mudding, which could result in a motor failure. Refrigerants must be stored in a sealed container and kept dry.

A filter-dryer is a device used to prevent refrigerant contamination in heating and refrigeration cycle systems. The filter can restrict the free flow of refrigerant into the expansion valve. It effectively prevents the impurities from flowing rearwards when the system is having a reversible flow and eliminates moisture or acids during the refrigeration and heating cycles.

The function is based on the very large surface area of ultrafine pores inside the substance. The water absorption of the filter dryer depends on the temperature and it increases as the temperature is reduced. From this point, a filter dryer is fitted in the cold inlet line. Relatively high flow velocities take place in the intake line, filter dryers with a very large volume would have to be used to keep the pressure loss in acceptable limits. That is why, the filter dryer is normally installed in the liquid line between the evaporator and the expansion valve (Althouse, 2013).

Refrigerant receivers are used to store the refrigerant. They allow to maintain the required variable refrigerant amount under various working conditions. The refrigerant receiver also has a supplementary protection function. Because of its position between the condenser and the expansion valve, it can separate vapour bubbles in the refrigerant from the liquid. Receiver volume must ensure all the heat pump freon volume. Therefore the repair work can be carried out by transferring the refrigerant into the refrigerant receiver, and no refrigerant can leak.

The sight glass allows to check if the freon flowing to the throttle valve is liquid. It is necessary to install it before the expansion valve. If vapour bubbles can be seen through the sight glass, it indicates the lack of refrigerant or insufficient supercooling in the system. In addition, sight glasses often include indicators showing the water content in the refrigerant.

Oil separator – the heat pump compressors receive oil from a small amount of special lubricating oil. It is distributed to a variety of compressor components. When the compressor is running, small amounts of oil come in hot, compressed freon vapour. The oil can be separated from the compressed hot steam by installing an oil separator between the compressor outlet and the condenser. The oil in the heat pump system is cooled down to a lower temperature parameter. However, it must be able to maintain high temperature in the compressor. It must remain liquid in all parts of the system. The liquidity of the oil-refrigerant mixture is determined by several factors. These include the temperature, the oil properties, the solubility of the oil in refrigerant and the solubility of the refrigerant in the oil. The internals of good refrigerant oil are:

- **low wax amount** -separation of the wax from the refrigerant oil blend can clog up the refrigerant control openings;
- **good thermal stability** - it should not form heavy coal deposits at the hot spots of the compressor;
- **good chemical stability** - there should be little or almost no chemical reaction with the refrigerant or substances that are commonly found in the system;

- **low pour point** - the ability of the oil to remain in the liquid state at the lowest system temperature;
- **low viscosity** - the ability of the oil to maintain good lubricating properties at high temperatures, as well as good low temperature viscosity always provides a good lubricating film.

Oil separators are very effective. Very little oil goes into the system. They are usually used in large commercial facilities. Integration of an oil separator is particularly important in systems with a flooded evaporator, as the oil precipitates at the base of the evaporator. Other positive result of the oil separator is that it can compensate pressure pulses in pressure lines.

A liquid line, commonly copper tubing, is used to carry the liquid refrigerant from the condenser to the evaporator. The lines are soldered or brazed to fittings. It is important to avoid pinching or buckling of these lines. They should be regularly maintained to prevent wear or breaking due to vibration (Hadorn, 2015).

5.9. Refrigerant

The refrigerant is responsible for transferring energy in the heat pump system. In theory, any material could be used as the refrigerant, if it can be condensed at technically possible pressures and evaporated at low temperatures. There is no ideal refrigerant for every application. Which refrigerant is used in a system depends on its purpose. The choice of refrigerant is primarily influenced by its physical, chemical and physiological properties. Scientists have found that the release of some refrigerant chlorofluorocarbons (CFCs) may damage the ozone layer.

Most refrigerants used today are classified in four areas:
- chlorofluorocarbons (CFCs),
- hidrochlorofluorocarbons (HCFCs),
- hidrofluorocarbons (HFCs).
- Refrigerant blend (azeotropic and zeotropic).

The refrigerant is identified by number (Table 5.1). The number goes under the letter R which means the refrigerant. The refrigerant cylinder is often colour coded to make it easier to identify the type of refrigerant in it.

Table 5.1. The most commonly used refrigerants (Source: own elaboration)

Cylinder colour	Number	Refrigerant name	Chemical composition
Orange	R-11	Trichlorofluoromethane	CFC
White	R-12	Dichlorodifluoromethane	CFC
Light blue	R-13	Chlorotrifluoromethane	CFC
Coral	R-13B1	Bromotrifluoromethane	CFC
Light green	R-22	Chlorodifluoromethane	HCFC
Light grey	R-23	Trifluoromethane	HFC
Purple	R-113	Trichlorotrifluoromethane	CFC
Dark blue	R-114	Dichlorotetrafluoromethane	CFC
Light grey	R-123	Dichlorotetrifluoromethane	HCHF
Deep green	R-124	Chlorotetrafluoromethane	HCHF
Medium brown	R-125	Pentafluoroethane	HFC
Light (sky) blue	R-134a	Tetrafluoroethane	HFC
Coral red	R-401A	R-22+R-152A+R-124	Zeotropic (HCFC)
Mustard yellow	R-401B	R-22+R-152A+R-124	Zeotropic (HCFC)
Blue- green (aqua)	R-401C	R-22+R-152A+R-124	Zeotropic (HCFC)
Pale brown	R-402A	R-22+R-125+R-290	Zeotropic (HCFC)
Green-brown	R-402B	R-22+R-125+R-290	Zeotropic (HCFC)
Orange	R-404A	R-125+R-143a+R-134a	Zeotropic (HCFC)
Light grey-green	R-406A	R-22+R-142b+R-600a	Zeotropic (HCFC)
Bright green	R-407A	R-32+R-125+R-134a	Zeotropic (HFC)
Peach	R-407B	R-32+R-125+R-134a	Zeotropic (HFC)
Chocolate brown	R-407C	R-32+R-125+R-134a	Zeotropic (HFC)
Rose	R-410A	R-32+R-125	Zeotropic (HFC)
Yellow	R-500	Refrigerants 152a/12	Azeotropic (CFC)
Light purple	R-502	Refrigerants 22/115	Azeotropic (CFC)
Aquamarine	R-503	Refrigerants 23/13	Azeotropic (CFC)
Teal	R-507A	Refrigerants 125/143a	Azeotropic (CFC)
Silver	R-717	Ammonia	Inorganic Compound

Chlorofluorocarbons (CFCs) refrigerants – the first ones were developed sixty years ago. These refrigerants include chlorine, fluorine and carbon. These refrigerants are low toxic, non-corrosive and compatible with other substances. They are non-flammable, non- explosive, but heat can induce them to decompose into their elements and cause harm to human tissue. They are particularly damaging to the respiratory system.

Simple refrigerants (CFCs) include R-11, R -12, R -113, R -114, R -115, R-500, R-502, and R-503. R-500, R-502, and R-503.

Simple refrigerants (CFCs) are thought to be one of the main reasons for ozone depletion. According to the international agreement, since 1995 they have not been produced, but they are still used in the existing units.

Due to laws forbidding the release of CFCs to the atmosphere, the new agent has been developed.

These are used to recover, recycle and reclaim the refrigerants containing (CFCs).

Hydrochlorofluorocarbons (HCFCs) refrigerants. These are molecules that consist of methane or ethane together with halogens. They form a new molecule that is considered partially halogenated. HCFCs refrigerants have shorter lives and cause less ozone depletion than the fully halogenated (CFCs). For this reason, they have reduced impact on the global warming (Althouse, 2013).

5.10. THE ABSORPTION SYSTEMS

5.10.1. Introduction

The absorption system uses heat energy, not mechanical energy. This heat energy is used to create necessary conditions for the cooling cycle completion. Compression systems use a compressor and a refrigerant to create a heat pump cycle. Absorption heat pump uses a chemical process to change low-temperature vapour into a high-pressure vapour. Most of the vapour compression systems used chlorofluorocarbon refrigerants whose use is now limited due to ozone depletion (Hadorn, 2015). The most commonly used refrigerants for absorption systems are ammonia/water or lithium bromide/water, in which case the absorption system is better. In order to encourage the use of absorption systems, it is necessary to improve their performance and reduce costs. The absorption system can use different sorts of heat sources. The condenser, expansion valve and evaporator are similar to the compression system. The compressor is replaced by a generator and an absorber.

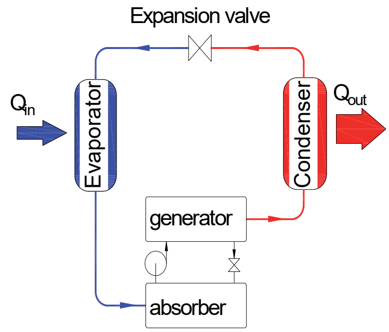

Fig. 5.18. The process of absorption in an absorption heat pump (Source: own elaboration)

Compare the functioning of the absorption heat pump (Fig. 5.18) with the compression heat pump (Fig. 5.3).

The absorption cooling cycle is a combination of two processes, as shown in Fig. 5.19.

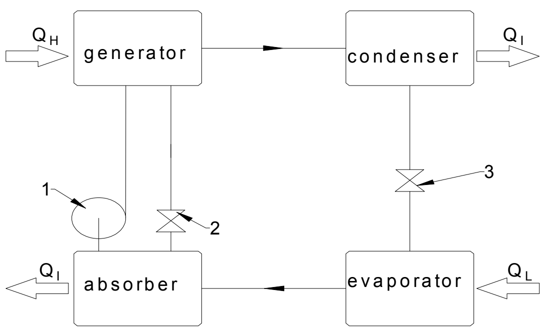

Fig. 5.19. Absorption refrigeration cycle: 1 – pump, 2, 3 – expansion valve, QL, QI, QH – low, intermediate and high temperature heat sources (Source: own elaboration)

Absorption cooling cycle changes the heat by three external sources; low Q_L, intermediate Q_I and high Q_H temperature scale. The high temperature source heat Q_H is used for running the absorption system. For the refrigerator, the useful heat Q_L transfer is at a low temperature, for the heat pump at an intermediate Q_I temperature. Since the separation process in the generator takes place at a higher pressure than the absorption process in the absorber, the circulation of the solution requires a circulation pump (Fig. 5.19, pos. 1). The working fluid in absorption refrigeration systems is a binary solution made up of a refrigerant and an absorbent. High temperature heat Q_H is supplied to the generator tank to remove the refrigerant from the solution. The performance of absorption systems is highly dependent on chemical and thermodynamic fluids characteristics. The essential requirement of the combination of the absorbent/refrigerant is that they must be mixed in the liquid phase during the cycle temperature range.

The following requirements apply to the mixture:

- the difference in boiling temperature must be as high as possible;
- the refrigerant should have high latent heat of vaporization and good absorption in absorbent material in order to keep low circulation range between the generator and the absorber;
- favourable viscosity, thermal conductivity and diffusion coefficients;
- refrigerants and absorbents must be cheap, non-corrosive, environmentally friendly.

The most commonly used liquids are water/NH_3 and LiBr/water.

The water/NH_3 fluid is widely used for both cooling and heating purposes. Both NH_3, refrigerant and water, absorbent are very stable for a large range of temperature and pressure. NH_3 has a high latency evaporation temperature, which is essential for the effective operation of the system. The freezing point of NH_3 is -77°C, therefore it can be used for low temperature applications. There are several disadvantages such as high pressure, toxicity and corrosion of copper and copper alloys. The NH_3 and water solution are volatile during the process, so it is necessary to reactivate the solution. Without the return of NH_3, water will accumulate in the evaporator and degrade the system performance (Althouse, 2013). The other most commonly used liquid for the absorption cooling and heating purposes is LiBr/water. Two extraordinary LiBr/water properties are: LiBr absorbency of material, regeneration and a very high solution evaporation temperature. The use of water as a refrigerant limit the temperature of the sublimation to 0°C. Since water is the refrigerant, the system must operate under vacuum conditions to achieve lower evaporation temperature. In addition to the extensive use of LiBr/water and water/NH_3 for many years, new functional fluids are being tested, such as fluorocarbon refrigerant-based working fluids, a mixture using inorganic salt absorbent NaOH/water and a mixture of LiBr+$ZnBr_2$/water.

5.10.2. Types of absorption systems

Single-effect absorption system

It is the most often used scheme. This system uses a non-volatile absorbent such as LiBr/water.

The high temperature heat Q_H supplied to the generator (Fig. 5.20) is used to evaporate the refrigerant from the solution. The refrigerant is supplied to a condenser (Fig. 5.20) that emits heat Q_I. The heat of the generator is used to heat the solution from the absorber, then the heat Q_l is rejected from the absorber. The high temperature heat at the generator is released at the absorber and the condenser. In order to improve the coefficient of performance COP, heat exchange HX is inserted as shown in (Fig. 5.20). The heat exchanger allows the solution from the absorber to be preheated before entering the generator. In this way part of the heat is returned. The heat input to the generator is less than the size of the generator and can be smaller. The COP can be increased by up to 60%.

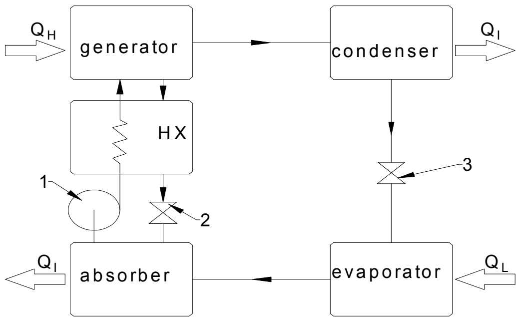

Fig. 5.20. Single-effect absorption system: 1 – pump, 2, 3 – expansion valve, HX – heat exchanger, QL, QI, QH – low, intermediate and high temperature heat sources (Source: own elaboration)

Absorption heat transformer. This system uses heat from the intermediate temperature sources Q_l and discards heat Q_L at a low temperature level. The useful heat Q_H output is obtained at the highest temperature level. Using an absorption heat transformer, any amount of waste heat Q_l can be increased to a higher temperature level Q_H without any other heat source, except for some work required for the working fluid to flow.

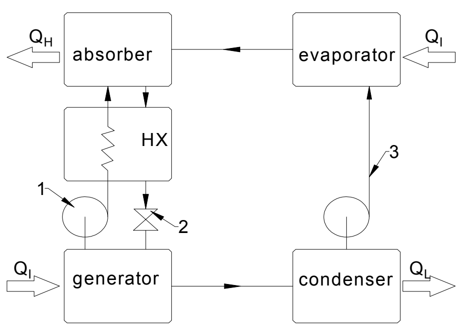

Fig. 5.21. Absorption heat transformer system: 1 – pump, 2, – expansion valve, 3 – pump, HX – heat exchanger, QL, QI, QH – low, intermediate and high temperature heat sources (Source: own elaboration)

The system (Fig. 5.21) has the same units as a single absorption cycle. The difference is that a pump (Fig. 5.21, pos. 3) is installed between the condenser and the evaporator.

The intermediate temperature heat Q_I supplied to the generator (Fig. 5.21) is used to evaporate the refrigerant from the solution. The refrigerant is supplied to the condenser (Fig. 5.21) that emits heat Q_L. Liquid refrigerant is pumped from the condenser to the evaporator. The pump raises the pressure in the evaporator. Liquid refrigerant is evaporated using the same intermediate waste heat Q_I used to drive the generator. The vapour refrigerant is absorbed by the solution in the absorber which discarded the heat in a high temperature environment.

5.11. Examples of practical applications

Ground heat pumps

Heat pumps often use ground as a lower heat source. Heat exchangers are installed in a horizontal or vertical position as shown in Fig. 5.22.

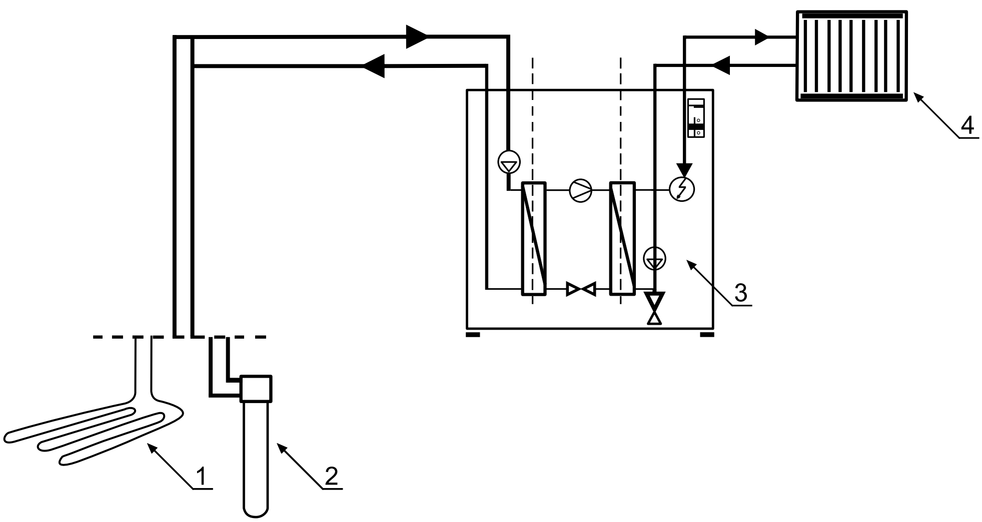

Fig. 5.22. Scheme of the system with a heat pump: 1 – horizontal heat exchanger, 2 – vertical heat exchanger, 3 – heat pump, 4 – radiator (Source: own elaboration)

Vertical ground heat pumps (Fig. 5.23) can be used even in small areas, however the knowledge of ground layers is necessary. The boreholes are mostly drilled to depths from 30 to 120 m. Fig. 5.24 shows the equipment for drilling vertical boreholes.

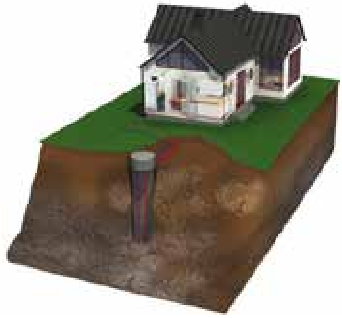

Fig. 5.23. A schema of the vertical ground heat pump (Source: Smuczyńska, 2015)

Fig. 5.24. Drilling of boreholes (Source: Smuczyńska, 2015)

In case of horizontal closed loops (Fig. 5.25) a sufficient land area is needed. The PVC, PB and PE pipes are used most frequently. The distance between pipes depends on the technology used and mostly is in a range between 0.8 and 1.0 m (Fig. 5.26). The advantages of this system are: low cost of work connected with shallow depth of pipes (about 1.5 m) and the possibility of installation within the foundations of the house. However, it should be noted that the amount of heat that can be obtained depends strongly on the ground type and can differ even 3-4 times in various locations.

Fig. 5.25. An example of the ground heat pump installation (Source: Smuczyńska, 2015)

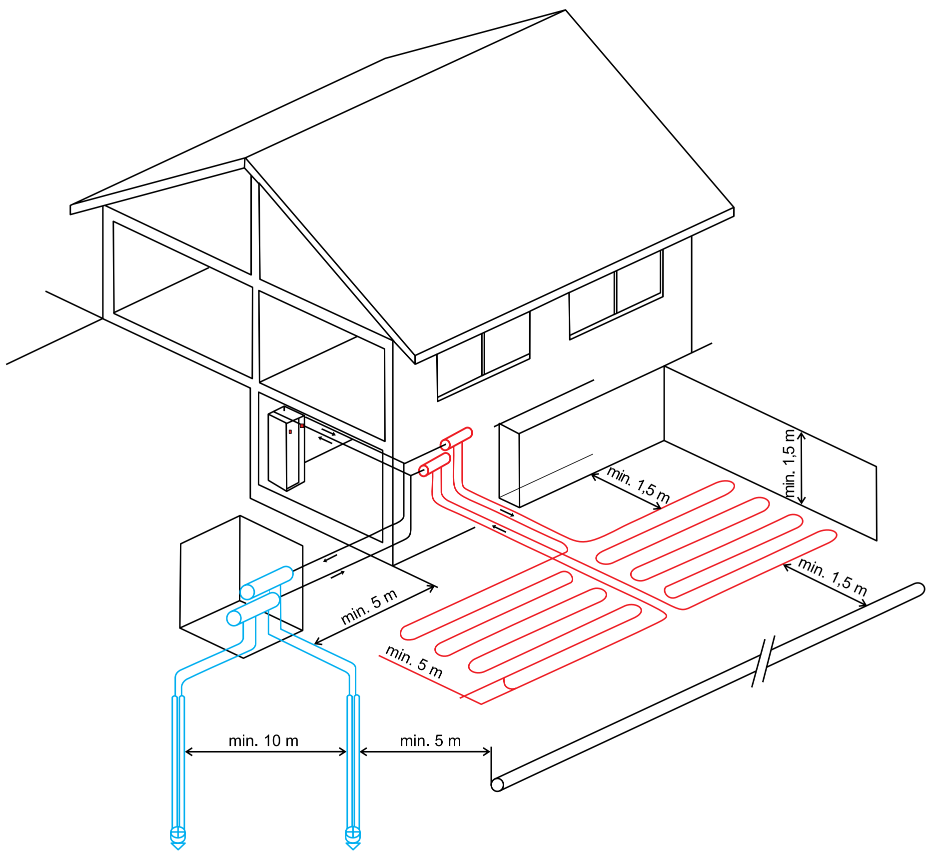

Fig. 5.26. A recommended distance between pipes (Source: own elaboration)

As shown by Rubik (2006), the necessary length of pipes (L) can be estimated from the following formula:

$$L = Q_o \cdot \ln\left(\frac{4x}{D_z}\right) / 2\Pi\lambda_A \left(t_x - t\right) \tag{5.14}$$

where:

Q_0 – total heat from the ground (W),
x – depth of pipe (m),
D_z – diameter of pipe (m),
λ_A – thermal conductivity of the ground (W/(m·K)),
t_x – ground temperature at depth x (°C),
t – medium temperature at condenser outlet (°C).

Thermal conductivity of the ground can be considered from 1.16-1.33 W/(m·K) for sand, to 2.3-2.44 W/(m·K) for wet sea sand or clay. Fig. 5.27 shows photos of the equipment (internal units) installed in existing buildings with ground heat pumps.

Fig. 5.27. Examples of ground heat pumps in buildings (Source: Smuczyńska, 2015)

Air heat pumps

Nowadays also air heat pumps are becoming more and more popular, although in Polish or Lithuanian conditions they are not efficient in severe winter period, so additional heat source is required (Figs. 5.28 and 5.29).

Fig. 5.28. An example of the air heat pump in a building (Source: Smuczyńska, 2015)

Fig. 5.29. An example of the air heat pump – external unit (Source: Smuczyńska, 2015)

Water heat pumps

Water, with its high thermal conductivity, could be used for high power systems (Fig. 5.30).

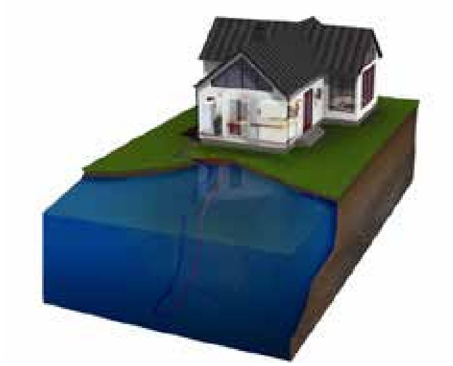

Fig. 5.30. An example of the water heat pump – external unit (Source: Smuczyńska, 2015)

Fig. 5.31. An example of the air heat pump – external unit (Source: Smuczyńska, 2015)

The groundwater-based systems that consist of two wells (for collection and discharge of water), as well as closed water loops using dams (Fig. 5.31) or lakes, require detailed assessment of the location of heat source, its capacity, local hydrogeological conditions and water quality. Due to the content of common groundwater contaminants, most heat pump manufacturers recommend the use of an additional indirect heat exchanger to protect the evaporator in the heat pump from precipitation and corrosion (Smuczyńska, 2015).

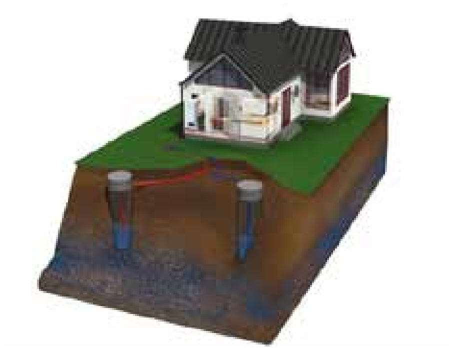

Fig. 5.32. An example of the water heat pump – external unit (Source: Smuczyńska, 2015)

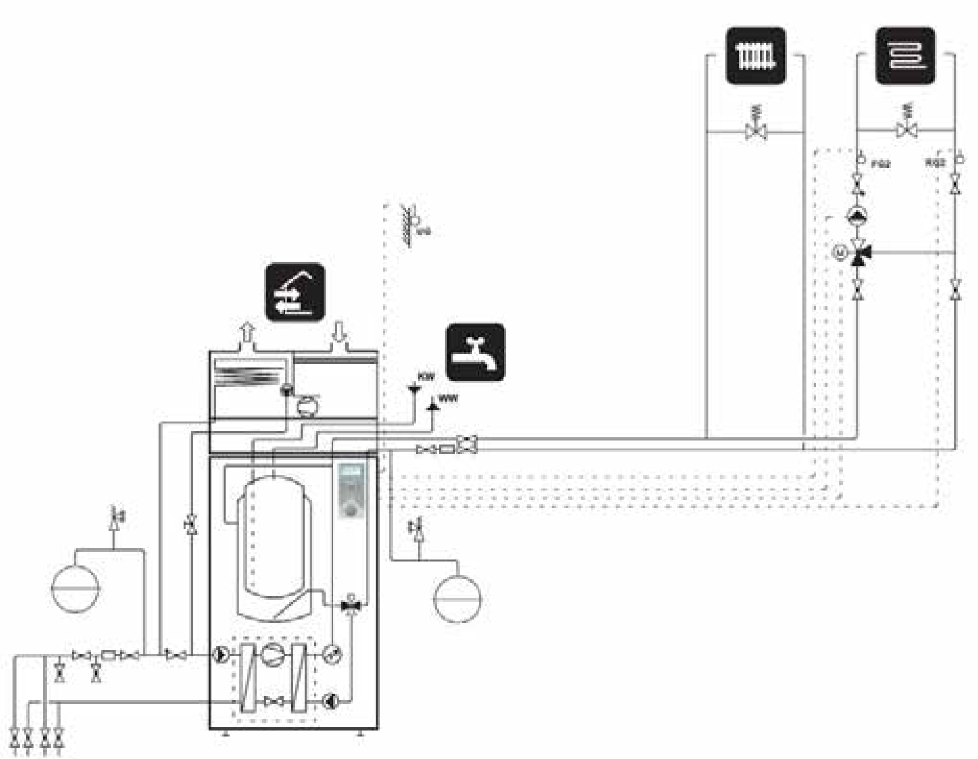

Fig. 5.33. A schema of system with heat pump working for heating, DHW, ventilation and cooling (Source: Smuczyńska, 2015)

Moreover, systems with two wells need to be installed properly (Fig. 5.32), and if the level of mineral and chemical impurities exceeds the specified values or the water intake is too small, the use of a heat pump as the source of heat may be impossible. In addition, the discharge well shall be designed in such a way that the discharge of cold water is carried to the same aquifer at a minimum distance of 15 m from the intake. If the groundwater level is below the depth of 30 m, a water permit is required to use it. The most common mistakes made during the design process and the installation of heat pumps are: wrong dimension and/or length of the ground collector pipes – caused by unawareness of the ground features (λ) or improper distance between the pipes. These slips could result in too long work of heat exchangers with high load and lack of time for regeneration of the soil.

Heat pumps started to be more and more popular. They are used in buildings to prepare hot water, for heating, cooling and ventilation (Fig. 5.33).

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