CZECH TECHNICAL UNIVERSITY IN PRAGUE

Faculty of Electrical Engineering

Power Engineering Department

ENERGY BALANCE OF THE COOLING SYSTEM FOR A DATA CENTER IN METEROLOGIC CONTIDIONS OF AZERBAIJAN

Energetická bilance chladicího systému datacentra v podmínkách Azerbajdžánu

Masters's Thesis

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Guidelines:

1. Get familiar with heat transfer that takes place in cooling systems.
2. Get familiar cooling systems.
3. For the defined system in a chosen location in Azerbaijan, calculate requirements for cooling.

Bibliography / sources:

1. Lienhard, John. (2013). A Heat Transfer Textbook. J Heat Transfer.
2. WOLFRAM, Stephen. An Elementary Introduction to the Wolfram Language [online]. [cit. 2019-10-09]. Available at: https://www.wolfram.com/language/elementary-introduction/2nd-ed/
3. Kyncl, Jan Electrical Heat (Course at CTU FEE). Available at: http://www.powerwiki.cz/wiki/BE1M15ETT

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I hereby declare that this thesis is the result of my own work and all the sources I used are in the list of references, in accordance with Methodological Instructions of Ethical Principle in the Preparation of University Thesis.

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Abstract

This master thesis helps to get familiar with cooling systems and principles of heat engines and exchangers. Further, the energy balance of the cooling system and accumulated yearly electricity consumption for the data center in Baku, Azerbaijan, is calculated based on weather measurements during the past three years.

Key Words

Data center cooling, heat engines, heat exchangers, heat pump, coefficient of performance, pressure micro turbines.

Abstrakt

Tato diplomová práce přibližuje principy a fungování chladících systémů, tepelných motorů a výměníků. Dále obsahuje energetickou bilanci chladícího systému a akumulovanou roční spotřebu elektřiny data centra v Baku, Ázerbájdžán, vypočtenou na základě měření počasí během posledních tří let.

Klíčová slova

Chlazení datového centra, Točivá redukce, Tepelné stroje, Tepelné výměníky, Tepelná čerpadla, Chladicí factor.

Table of Contents

List of Equations

List of Figures

List of Table

1. INTRODUCTION

The number of individuals and businesses that rely on the internet is increasing. This increase gives rise to facilities such as internet service providers, application service providers, network operation centers, and web-hosting sites. Therefore, demand for data centers increases as well, becoming busier and more frequent (Day, T. 2009). Data centers require several components to perform, reliability, and security 24 hours a day 365 days a year, which turns them into high energy demanding infrastructures (Oro, E. & Salom, J. 2015).

Data centers are centralized locations for computing and networking equipment, which are gathered together. The main task of the data centers is collecting, storing, processing, distributing access to large amounts of data. These centers provide vital services such as data storage, backup and recovery, networking, and data management. Nowadays, almost every business needs access to data centers. Most of the companies either have their data center or require access to someone else’s. Today’s data center is more likely to have thousands of powerful and tiny servers running 24/7. There are more than 3 million data centers of different shapes and sizes in the world.

There are three subsystems that support the operations of data centers. They are IT equipment that provides services to end-user, power infrastructure that supports the IT and the cooling equipment, and the cooling infrastructure that removes the heat generated by these subsystems (Liu, Z. et al. 2012). The average data center uses a noticeable amount of electricity. Not to mention the cooling system required to maintain the ideal operating environment for all that equipment. Both together, they consume about three percent of the world’s electricity. With higher energy consumption facilities on the way in the coming years, power usage is increasing despite improvements in energy efficiency.

A data center’s electrical system must have redundant energy supply. They are reinforced with uninterrupted power supply (UPS) battery systems, and a backup generator, which can provide enough power to keep the facility functioning if the main power is disrupted for any length of time. When the device senses that the main power supply is interrupted, the UPS switches to supply mode and becomes the primary power source and keeps running long enough for the generations to start successfully (Rouse, M. 2019). Data center power infrastructure is commonly supplied with more than one electrical feeder, which provides additional redundancy and security.

1.1. Cooling systems

To obtain suitable temperature management in a data center, it is crucial to support the functionality of the equipment. Overheating in a data center can cause enormous problems for the equipment that can be avoided by using cooling techniques.

The most commonly used cooling systems are described below.

Calibrated Vectored Cooling (CVC)

This technique is mostly designed for high-density servers. It optimizes the airflow path through a device to allow the cooling system to manage heat more efficiently. It thus enables to increase the ratio of circuit boards per server chassis and utilize fewer fans.

Chilled Water System

This system is conventional for mid-to-large sized data centers which use chilled water to cool air being brought in by computer room air handlers (CRAHs). The chiller plant that supplies water is usually located in the facility.

Computer Room Air Handler (CRAH)

This unit functions as part of a more significant system associate to a chilled water plant somewhere in the facility. Chilled water goes through a cooling coil within the unit, which then uses modulating fans to carry air from outside the facility to the inside. Because, CRAHs function by chilling outside air, they are very efficient in the locations with colder annual temperatures.

Cold Aisle/Hot Aisle Design

This technique is a widespread model of data center server rack positioning that alternates the rows of “cold aisles” and “hot aisles.” On the front of the racks, the cold aisles feature cold air intakes, while on the back of the racks, hot aisles contain the hot air exhausts. Hot air is expelled into the air conditioning to be chilled, and then venter into the cold aisles. Blanking panels are used to fill the empty racks in order to prevent over-heating or wasting cold air.

Computer Room Air Conditioner (CRAC)

CRAC units, one of the most common components of any data center, are very similar to conventional air conditioners powered by a compressor that draws air across a refrigerant filled cooling unit. In terms of energy usage, they are very inefficient; however, the equipment is relatively inexpensive.

Critical Cooling Load

This is a measurement used represents the total usable cooling capacity (generally expressed in watts of power) on the data center floor that can be used for the cooling servers.

Direct-to-Chip Cooling

This is a liquid cooling method that uses pipes to transport the coolant directly to a cold plate incorporated into a motherboard’s processors to extract heat. Separated heat is fed into a chilled-water loop and send to the chiller plant of the facility. It is one of the most efficient forms of server cooling because this method cools the processors directly.

Evaporative Cooling

This method controls temperature by exposing hot air into the water, which causes the water to evaporate and attract the heat out of the air. The water can be introduced in two ways; either as a misting system or a wet material such as a filter. The system is very energy efficient as it does not use CRAC or CRAH units; however, it requires much water. Datacenter cooling towers are used to manage evaporations and carry the excess heat to the outside atmosphere.

Free Cooling

It is any data cooling system that brings cooler air into servers from the outside atmosphere, rather than continually chilling the same internal air. It is a very high energy-efficient form of a server cooling. The only limitation is that this can only be implemented in certain climates.

Immersion System

This is a new innovative data center liquid cooling solution. The system submerges hardware into a bath of non-conductive, and non-flammable dielectric fluid.

Liquid Cooling

It is any cooling technology that uses liquid, to evacuate heat from the air. Increasingly, data center liquid cooling refers to specifically direct cooling methods that expose server components such as processors to liquid to cool them more effectively.

Raised Floor

It is a frame that lifts the data center floor above the building's concrete slab floor. The area between the two is used for water-cooling pipes or increased airflow. Even though now power and network cables are sometimes run through this space as well, there newer data center cooling design and best practices place these cables overhead (Gyarmathy, K. 2020).

2. GENERAL PRINCIPLES

2.1. Conservation Laws

2.1.1. Conservation of Mass

In general, the amount of any particular substance is not conserved. Chemical reactions can change from one state to another, and nuclear reactions can even change one element into another. However, the total mass of all substances is conserved (Crowell, B. 2003).

A total mass of an isolated system is constant. Energy neither created or destroyed. It can only be transformed from one form to another or transferred from one system to another (Crowell, B. 2003).

In an equation format, the conservation of mass principle is:

$$\sum \dot{M}_{in} = \sum \dot{M}_{out} + \dot{M}_{net\ change\ in\ the\ system}$$

Equation 1 - Conservation of mass principle

According to the equation, the net change in the system has a positive sign. However, if the amount of substance in the system decreases, this will be represented with a negative net change. A decrease occurs most commonly because of imperfections in the equipment or because of incorrect process management (Salaba, J., & Šťastný Jiří. 1982).

2.1.2. Conservation of Energy

The energy balance of systems corresponds to the expression of the general law of conservation of energy, which states that the sum of energy entering the system corresponds to the sum of energy leaving the system and the net energy change in the system. This relation is described by Equation 2. When compiling balances in continuous processes, it is common to choose balancing per a unit of time (second or an hour), for discontinuous processes, one complete work cycle is often selected as the interval of balancing (Salaba, J., & Šťastný Jiří. 1982).

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} + \dot{E}_{net\ energy\ change}$$

Equation 2 - Energy balance

When it comes to energy balances, the processes can be divided into three types. The first type of process is present inflow systems in a steady state – in this case, there is no net energy change in the system, and the entry transfers correspond to the exiting transfers of energy. The second type of process is present inflow systems in a transient state – in this case, the net energy change in the system is positive, and the transfers are dependent on time. The third type of process is present in closed systems. A closed system is a system that does not allow any transfer of mass in or out of the system. One can use the modified equation of energy balance for the closed discontinuous system (Equation 3). The initial energy of the system E_0 after adding the supplied heat energy $Q_{supplied}$ corresponds to the final energy E_k and the net work W done by the system.

$$E_0 + Q_{supplied} = E_k + W$$

Equation 3 - Modified equation of energy balance for the closed system

In energy balances of the systems, if electrostatic, magnetic and surface energies are neglected, then the following components occur; internal energy, pressure energy (conservation of sum energy in piping), gravitational potential energy, kinetic energy, supplied heat energy, work and the net energy change in the system. These components are balanced in Equation 4 (Salaba, J., & Šťastný Jiří. 1982).

$$\sum m_{i1}u_{i1} + \sum m_{i1}h_{i1}g + \sum m_{i1}p_{i1}v_{i1} + \sum m_{i1}\frac{{c_{i1}}^2}{2} + \sum m_{i1}q_{i1} =$$

$$= \sum m_{i2}u_{i2} + \sum m_{i2}h_{i2}g + \sum m_{i2}p_{i2}v_{i2} + \sum m_{i2}\frac{{c_{i2}}^2}{2} + \sum m_{i2}w_{i2} + E_{net\ energy\ change}$$

Equation 4 - Balanced equation

The expression on the left side of the equation describes the entry components; the expression on the right side of the equation describes exiting components. The first members on both sides of the equation represent the internal energy of the input and output substances. The second and fourth members describe the gravitational potential energy and kinetic energy. The third member corresponds to the pressure energy, which is proportional to the work needed for pushing the substance into the system. The last member on the left side describes the energy supplied to the system in the form of heat. At the system's exit, we define the net work done with the product of $m \cdot w$ and the net energy change in the system E.

2.2. Carnot Cycle

Heat, energy consumption, cooling systems - all these have Thermodynamics in common. Thermodynamics is a branch in physics which's laws, and general principles are obeyed in every cooling system.

In 1824 Sadi Carnot published *Réflexions sur la Puissance Motrice du Feu* (Reflections on the Motive Power of Fire), which is considered the founding work of thermodynamics theory.

The Reflections contain some principles such as the Carnot cycle, Carnot's theorem, the Carnot heat engine, and thermodynamic efficiency.

A Carnot cycle is an ideal reversible closed thermodynamic cycle, which involves four successive operations, which are adiabatic compression, isothermal expansion, adiabatic expansion, and isothermal compression. During the procedure, the expansion and compression of substance can be done up to the desired point and back to the initial state (Carnot, S. 2012).

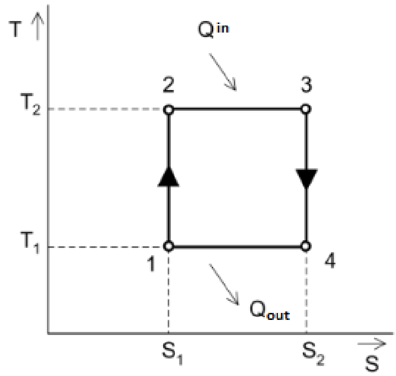

Figure 1 - Carnot Cycle

Description of the Carnot cycle can be seen below (Carnot Engine, Carnot Theorem & Carnot Cycle - Working, Efficiency. 2020).

- 1-2 the process is reversible adiabatic gas compression. The temperature rises to T_2 as the surroundings continue to do their work on the gas.

- 2-3 the process is reversible isothermal gas expansion. In this process, the amount of heat absorbed by the ideal gas is Q_{in} from the heat source, which is at a temperature of T_2. The gas expands and does work on the surroundings.

- 3-4, the process is a reversible adiabatic gas expansion. Here, the gas continues to expand, and work is done on the surroundings. Now the temperature is lower, T_l.

- 4-1 the process is reversible isothermal gas compression process. Here, the heat loss, Q_{out} occurs when the surroundings do the work at temperature T_1.

The work being done then corresponds to the difference between heat input and dissipated heat according to the law of conservation of energy (Equation 5).

$$W = Q_{in} - Q_{out}$$

Equation 5 – Work done

The heat cycle efficiency is the ratio between the transformed energy (work being done) and the supplied heat energy. The supplied and dissipated heat during the isothermal change is proportional to the difference between entropy and the temperature at which the process occurs Using this relation, we can modify the formula of thermal efficiency to the form (Equation 6) which is dependent only on the temperatures between which the Carnot cycle operates.

$$\eta_t = \frac{Q_{in} - Q_{out}}{Q_{in}} = \frac{T_2 \cdot \delta s - T_1 \cdot \delta s}{T_2 \cdot \delta s} = \frac{T_2 - T_1}{T_2}$$

Equation 6 - Modified formula of thermal efficiency

The maximal theoretical thermal cycle efficiency is determined only by the temperatures between which the cycle operates. It is impossible to achieve a cycle efficiency of 1. Although it is possible to arithmetically achieve a cycle efficiency of 1 for the zero temperature state of the cooler, it is not achievable in a finite amount of operations according to the sixth postulate of thermodynamics. The Carnot cycle is unachievable in a real heat engine, and it is only a theoretical ideal cycle. Real cycles are, however, trying to reach the cycle efficiency of the Carnot cycle, for example, by repeating the most efficient component of the cycle.

With the inverse operation of the cycle, we can change the heat motor into a heat pump. The thermodynamical processes stay the same; however, they are operating in the opposite sense. Work is not being done by the machine, it must be provided. With that, the supplied heat energy is transferred into the engine with the lower temperature level T_1 to the higher temperature level T_2, from which the heat is dissipated.

2.3. Pressure losses in piping and a pump design

2.3.1. Pipe flow

For the fluid flow in the piping, it is possible to use the balance form of the law of conservation of mass (Equation 7). Assuming a steady flow (partial time derivative is equal to zero), the law changes into the form (Equation 8).

$$\dot{m}_1 - \dot{m}_2 = \frac{\partial}{\partial t} \int_{(V)} \bar{\rho} \cdot dV$$

Equation 7 - Conservation of mass

$$\dot{m}_1 = \dot{m}_2$$

$$\overline{\rho_1} A_1 \overline{c_1} = \overline{\rho_2} A_2 \overline{c_2}$$

Equation 8 - Conservation of mass (steady flow)

If we approximate the fluid as incompressible, we can use the relation (Equation 9). This approximation is possible if there is no decrease higher than 5% in the density of the liquids during the flow. Assuming a local decrease in the density of fluids, we can determine the local velocity of the flow for the set fluid with the relation (Equation 10) (Kočárník, P., 2017).

$$A_1 \overline{c_1} = A_2 \overline{c_2}$$

Equation 9 - Incompressible fluid equation relation

$$\frac{\rho}{\rho_{c=0}} = \left(1 + \frac{\kappa - 1}{2} \cdot Ma^2\right)^{\frac{1}{1-\kappa}} \geq 0{,}95$$

Equation 10 - Local velocity flow equation relation

When we substitute the value 1,4 as the isentropic coefficient κ, we get the local value of Mach's number 0,32. Mach's number corresponds to the ratio of flow velocity to the velocity of sound in the fluid. The local velocity for air with a temperature of 0 °C then results in approximately 106 m/s. (Kočárník, P., 2017).

The extended Bernoulli equation describes the one-dimensional flow of a fluid with a constant density in a gravitational field. This equation is often presented in different forms depending on the dimensions of its members (energetic, gravitational, pressure), the Equation (11) is in the energetic form. The beta coefficients respect the error established by the flow speed and take into account the non-viscous fluid instead of the real velocity profile of a viscous fluid. The coefficient for the turbulent flow is considered as 1, and for the laminar flow, it is equal to 2 (Kočárník, P., 2017).

$$gy_1 + \beta_1 \frac{\overline{c_1}^2}{2} + \frac{\overline{p_1}}{\rho} = gy_2 + \beta_2 \frac{\overline{c_2}^2}{2} + \frac{\overline{p_2}}{\rho} + \int_{s_1}^{s_2} \frac{\partial \bar{c}}{\partial t} ds + e_{dis}$$

Equation 11 - Extended Bernoulli equation

The flow of the fluid can be divided into two types – the laminar and the turbulent. A possible criterion for the distinction of the type of the flow is Reynold's number designated as *Re* which can be calculated according to the following relation (Equation 12), where ν is the kinematic viscosity, μ is the dynamic viscosity, D is the diameter of the pipe and $\bar{c}$ is the flow speed.

$$Re = \frac{\bar{c}D}{\nu} = \frac{\bar{c}D\rho}{\mu}$$

Equation 12 - Reynold's number

For a pipe with a circular cross section, the critical value of Reynold's number is 2300, and lower values show that the flow is laminar. The area between 2300 and 10^4 is a transition area, values higher than 10^4 represent a turbulent flow. The flow speed is influenced by the velocity profile, the size of the losses, and the heat transfer between the fluid and the wall of the pipe (Kočárník, P., 2017).

2.3.2. Pressure losses

Pressure losses can be divided into frictional losses (longitudinal) and local losses. Losses (through dissipated energy e_{dis}) are expressed as the multiple ζ of the value of the specific kinetic energy, according to the relation (Equation 13), where λ is the coefficient of frictional losses, *D* is the diameter of the pipe and *l* is the pipe's length.

$$e_{dis} = \zeta \cdot \frac{\bar{c}^2}{2}$$

$$\zeta_{frictional} = \lambda \frac{l}{D}$$

Equation 13 - Losses

The relation between pressure losses and the dissipated energy is described by the Equation (14), where ρ is the density of the flowing fluid.

$$p_z = e_{dis} \cdot \rho$$

Equation 14 - Relation between pressure losses and the dissipated energy

The longitudinal losses are strongly dependent on the flow speed of the fluid. The coefficient of the frictional losses for the laminar flow is a function of only Reynold's number, and with its increase, the coefficient decreases according to the relation (Equation 15) (Kočárník, P., 2017).

$$\lambda = \frac{64}{Re}$$

Equation 15 - Longitudinal losses

The coefficient for the turbulent flow is a function of Reynold's number, the roughness of the surface δ, and the diameter of the pipe D. The process is determined experimentally, and it can be calculated by the function (Equation 16).

$$\lambda = \left\{2\ log\left[\left(\frac{6,97}{Re}\right)^{0,9} + 0,27\frac{\delta}{D}\right]\right\}^{-2}$$

Equation 16 - Turbulent flow function

The local losses are caused by changes in the shape of the pipe or, for example, by a change of the direction of the flow. The values of the coefficient ζ are determined experimentally for various general types of changes and are tabulated (Kočárník, P., 2017).

Calculating the pressure losses is similar to calculating the frictional losses according to the relation (Equation 13).

3. HEAT ENGINES AND HEAT EXCHANGERS

With the knowledge of thermodynamical processes and cycles, we can construct heat engines. A heat engine is an engine that uses mutual transformations of heat energy and internal energy into work according to the first equation of thermodynamics, and at the same time, it respects the second equation of thermodynamics. This chapter contains principles of compressor, adsorption and absorption units, heat exchangers, and micro reduction turbines.

3.1. Heat pumps

The common feature of these pumps is, for example, the use of electrical energy for increasing the temperature level of heat energy. This capability allows us to use a medium of higher temperature to cool down a medium of a lower temperature. The uses of both of these features of heat pumps in the industry are extensive. In the design of the cooling system, the chapter will be dealing with compressor cooling systems and ab-sorption cooling systems.

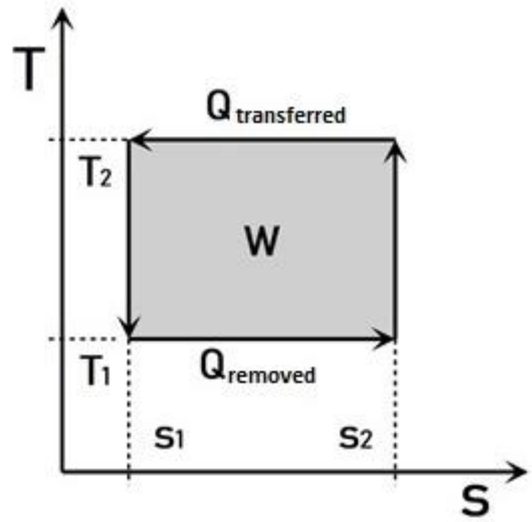

Figure 2 - Carnot cycle in the direction of the heat pump

The ideal theoretical heat pump is a machine which would apply the Carnot heat cycle in an opposite sense, according to Figure 2. The heat pump removes the heat $Q_{removed}$, which has a low temperature level T_1 and outputs the heat $Q_{transferred}$, which has a higher temperature level T_2. During this process, the heat pump uses the work W. From the law of conservation of energy, the following equation (17) can be used, which describes the energy balance of the heat pump.

$$Q_{transferred} = W + Q_{removed}$$

Equation 17 - Energy balance of the heat pump

The transferred and removed heat during the isothermic process is determined by Equation 18.

$$Q_{transferred} = T_2 \cdot \delta s$$

$$Q_{removed} = T_1 \cdot \delta s$$

Equation 18 - Transferred and removed heat

For this direction of the Carnot cycle, it is possible to determine two different heat factors. The factor which corresponds to the use of the set cycle, be it cooling or heating, is then applied. The factor in the sense of cooling is named as the cooling factor (COP_R), for heating, it is the heating factor (COP_H). The cooling factor corresponds to the ratio of removed heat to supplied work, whereas the heating factor is the ratio of transferred heat to supplied work. The process of deriving these efficiencies for the Carnot cycle is shown in Equation 19.

$$COP_R = \frac{Q_{removed}}{W} = \frac{Q_{removed}}{Q_{tranferred} - Q_{removed}} = \frac{T_1 \cdot \delta s}{T_2 \cdot \delta s - T_1 \cdot \delta s} = \frac{T_1}{T_2 - T_1}$$

$$COP_H = \frac{Q_{transferred}}{W} = \frac{Q_{transferred}}{Q_{transferred} - Q_{removed}} = \frac{T_2 \cdot \delta s}{T_2 \cdot \delta s - T_1 \cdot \delta s} = \frac{T_2}{T_2 - T_1}$$

Equation 19 - Derived efficiencies for cooling and heating factors

The following simple relation between these factors is apparent (Equation 20).

$$COP_H = 1 + COP_R$$

Equation 20 - Relation between heating and cooling factors

3.1.1. Compressor cooling systems

Compressor cooling systems are heat pumps that use electrical energy for the transfer of heat energy from the cooling circuit of a low temperature level into the circuit of the cooler with a higher temperature level. The compressor cooling systems use electrical energy to perform work – the electrical energy is then changed into mechanical energy in the compressors. The mechanical energy then raises the pressure level in the heat pump.

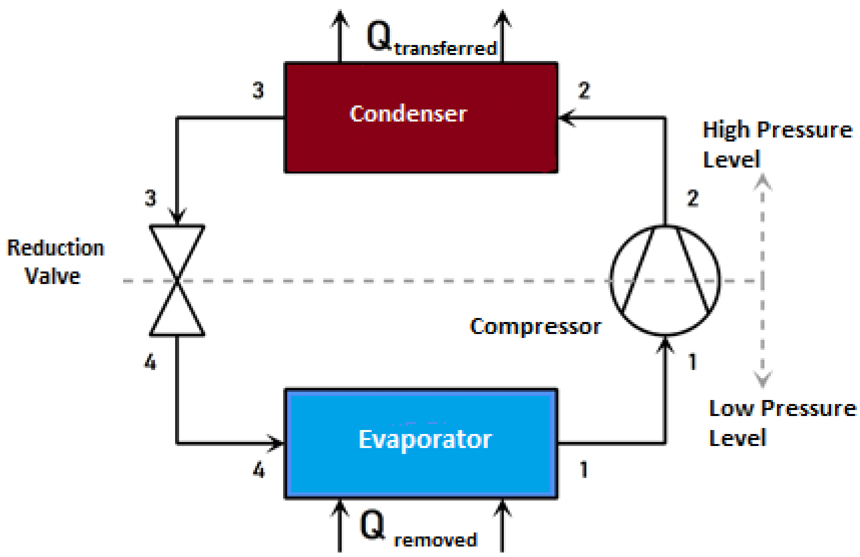

Figure 3 - Diagram of the heat compressor pump

The circuit of the heat compressor pump (Figure 3) works between two separate temperature levels. At a lower temperature level, the heat is removed from the set medium – the cooling occurs. The removed heat causes the evaporation of the work medium from the evaporator. The created vapor is then compressed in the compressor and is subsequently transferred to the condenser, where condensation occurs. The condensation is connected with the transfer of heat energy, which is then led out of the pump. The condensate is connected with the evaporator through the expansion valve. In the expansion valve, a reduction of pressure occurs – the pressure is reduced to a lower level corresponding to the pressure in the condenser. The consumption of the heat compressor pump is concentrated on only the compressor itself and can be supplied with mechanical or electrical energy.

The comparison cycle in the T-s diagram corresponding to the heat compressor pump also corresponds to Figure 4, assuming a reversible adiabatic operation of the compressor. The cycle is then composed of the isentropic compression, isobaric condensation, isenthalpic throttling process, and the isothermal-isobaric evaporation. The numbering of points in the T-s diagram of the comparison cycle corresponds to the numbering of the states in the diagram (Figure 3).

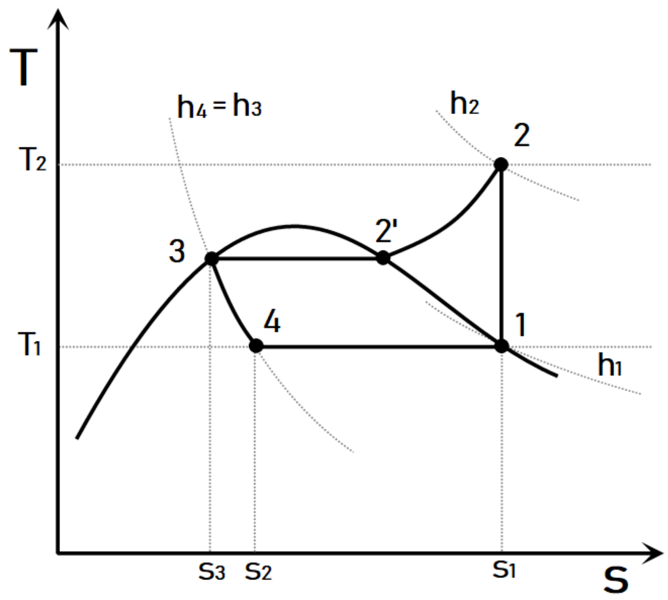

Figure 4 -The cycle of the heat compressor pump

3.1.2. Sorption systems

Sorption devices can be described as heat pumps that do not primarily use electrical energy for their operation – they use energy in the form of heat. Both systems work at a lower pressure level to enable cooling with low temperatures. On the basis of heating, we can divide the sorption devices into directly and indirectly heated categories. Directly heated devices use the combustion of gaseous or liquid fuels (natural gas, LPG, oil, fuel oil, etc.) to get the required propulsive heat output. Indirectly heated devices use the supply of heat energy into the sorption device through the heat transferring medium (hot water, steam, flue gases). Sorption devices are also divided into adsorption and absorption devices. Devices with an adsorption cycle are more likely to be applied to lower cooling outputs, and in comparison to absorption devices, they also have a lower COP factor.

3.1.2.1. Adsorption devices

An adsorption cooling device is based on the principle of physical adsorption. The adsorption process occurs on the phase boundary of the solid, gaseous or liquid substance, where molecules of gaseous or liquid substances gather and form layers. These layers of molecules are then bound with intermolecular Van der Waals forces. This type of intermolecular bond is reversible. The solid porous phase with a large reaction surface is called an absorbent, and the gaseous or liquid phase is called an absorbate.

The fundamental principle of an adsorption device is based on Figure 5. The cooled water is led into the evaporator (marked with the number 3 in the picture), where a transfer of heat energy to the liquid and its evaporation occurs. The vapor is led through the open valve into a dry adsorber (marked with the number 1 in the picture), where it then bonds. The adsorber is cooled with a circuit of cooling water. The adsorption process is active until the adsorber reaches a saturated state. When the process reaches this point, the adsorber is no longer able to keep binding the vapor. Subsequently, desorption occurs, and the adsorbent is dried with heating water. The bonded water vapor is gradually released, and it is led through an open valve into the condenser (marked with number 4 in the picture). The vapor inside the condenser is then liquefied using the circuit of cooling water, which then leads the heat transferred from the vapor out of the adsorption device. To ensure smooth operation, two adsorbers are being used in the adsorption device (marked with the numbers 1 and 2 in the picture). The two adsorbers cyclically alternate between adsorption and desorption.

The cooling factor of the adsorption units (COP_R) is usually not too high, common values are around 0,6.

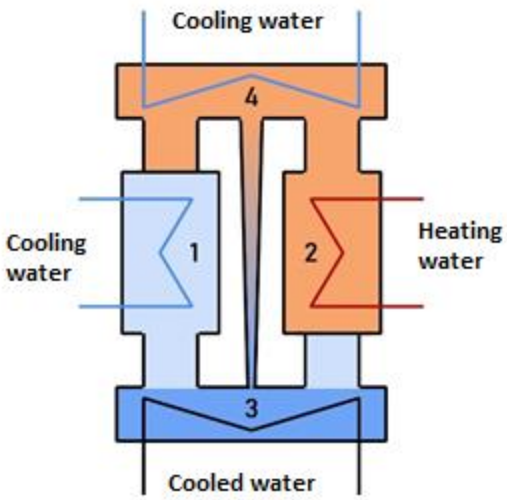

Figure 5 - The adsorption device

3.1.2.2. Absorption devices

The absorption device uses the physical principle of absorption. This process consists in absorbing the gaseous phase into the liquid phase. In this process, the liquid is absorbent, whereas the absorbed gas is the absorbate. The pair of water and ammonia ($NH_3 + H_2O$) is commonly used as the working substance. The aqueous lithium bromide solution ($LiBr + H_2O$) is also widely used. The technology using the ammonia allows working with negative temperatures of the cooling medium. Devices using the aqueous lithium bromide solution often work with a medium cooled down to about 7°C.

The absorption units can also be divided into two basic types according to the arrangement of the system. Both of these types work between three temperature levels and are here, to simplify, modeled as pairs of Carnot cycles – a right-handed and a left-handed Carnot cycle.

In the type marked with the roman numeral I (Type I Absorption Heat Pump), the Carnot motor works on a higher temperature level (T_2 and T_1), and supplies the work done to the left-handed cycle between the temperatures T_1 and T_0. The cycle of type *I* in the T-s diagram is shown in Figure 6. The heat (Q_2) with the higher temperature level (T_2) is supplied to the system, and because of this, we can then provide the heat (Q_0) with a lower temperature level (T_0). Moreover, we can also remove the heat Q'_1 and Q''_1 with the temperature level T_1 (Herold, K. et al. 1996).

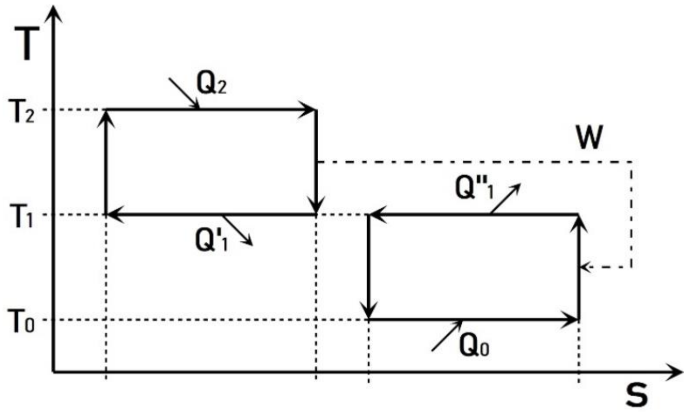

Figure 6 - The absorption heat pump of type I in the T-s diagram

This type of absorption device allows us to describe the COP_R (for cooling) and COP_H (for heating) factors with the relations (Equation 21). As the mentioned cycles are Carnot cycles, these relations can be explained only with temperatures between which these cycles work. With this, we can achieve the maximal cycle efficiency of the absorption cycle of type I. The aforementioned Equation 21 can still be used (Herold, K. et al. 1996).

$$COP_H = \frac{Q'_1 + Q''_1}{Q_2} = \frac{T_2 - T_0}{T_2} \cdot \frac{T_1}{T_1 - T_0}$$

$$COP_R = \frac{Q_0}{Q_2} = \frac{T_2 - T_1}{T_2} \cdot \frac{T_0}{T_1 - T_0}$$

Equation 21 - Maximal cycle efficiency of the absorption cycle type I

The type marked with the roman numeral II (Type II Absorption Heat Transformer), Figure 7, is composed of the right-handed Carnot cycle working between the temperatures (T_1 and T_0). The work done corresponds to the work consumed by the left-handed Carnot cycle between the temperatures T_2 and T_1. In this manner, the heat Q_2 with a higher temperature level T_2, and the waste heat Q_0 with the temperature level T_0 is being removed. For the operation of the type II absorption cycle, the heat $Q'_1 + Q''_1$ with the temperature T_1 needs to be supplied (Herold, K. et al. 1996).

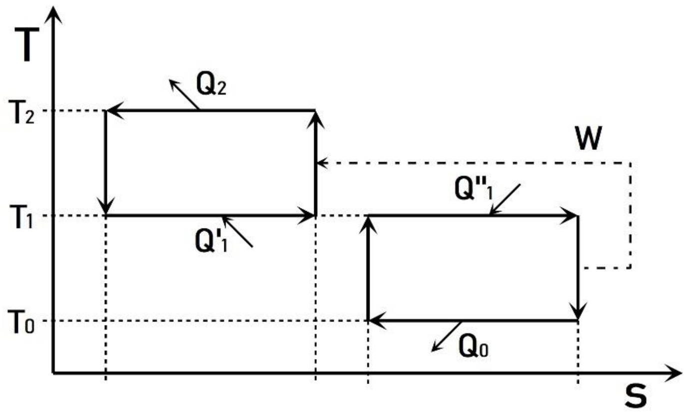

Figure 7 - The absorption heat transformer of type II in the T-s diagram

The relation for the heating factor COP_H in the type II absorption heat transformer can be described similarly to the ratio of dissipated heat to propulsion heat (Equation 22) (Herold, K. et al. 1996).

$$COP_H = \frac{Q_2}{Q'_1 + Q''_1} = \frac{T_1 - T_0}{T_1} \cdot \frac{T_2}{T_2 - T_0}$$

Equation 22 - Heating factor in the type II absorption heat transformer

The variant I is used for cooling purposes, so this paper will be dealing with this type of connection for the absorption device from now on. This type of absorption device mainly uses single-stage and two-stage connections. The simpler single stage-cycle, including its main components and temperature and pressure levels, is shown in the Dühring diagram in Figure 8. It is apparent from the diagram that the propulsion heat Q_d is led into the desorber (generator, ejector). The cooling output in the diagram is represented by the heat supplied into the evaporator. The removed heat from the absorber and condenser corresponds to the waste heat, which needs to be transferred out of the system. The evaporator and the absorber both have lower pressure levels, whereas the condenser and the desorber both have higher pressure levels. These pressure levels are separated with expansion valves and the pump. The description of the individual components of the system is listed below.

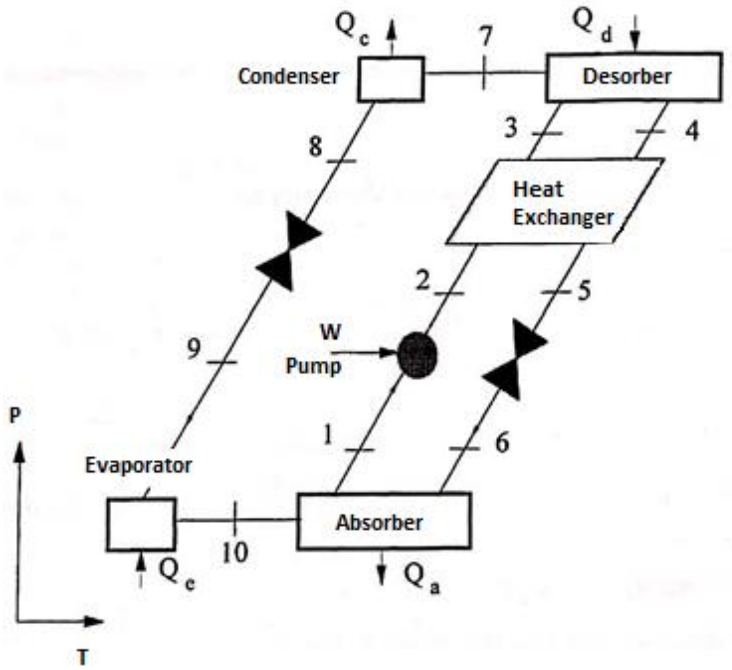

Figure 8 – Dühring diagram of the single stage absorption cooling system (LiBr + H_2O) (Herold, K. et al. 1996)

The individual components of the single-stage system can be described with the following equations of mass and energy balances. The indexation of these equations corresponds to the states from Figure 8 (Herold, K. et al. 1996).

Pump:

$$m_1 = m_2$$

$$x_1 = x_2$$

$$m_1 h_1 + W = m_2 h_2$$

Heat exchanger:

$$m_2 = m_3$$

$$x_2 = x_3$$

$$m_4 = m_5$$

$$x_4 = x_5$$

$$m_2h_2 + m_4h_4 = m_3h_3 + m_5h_5$$

Expansion valve of the solution:

$$m_5 = m_6$$

$$x_5 = x_6$$

$$h_5 = h_6$$

Absorber:

$$m_{10} + m_6 = m_1$$

$$m_{10}x_{10} + m_6x_6 = m_1x_1$$

$$m_{10}h_{10} + m_6h_6 = m_1h_1 + Q_a$$

Desorber:

$$m_3 = m_4 + m_7$$

$$m_3x_3 = m_4x_4 + m_7x_7$$

$$m_3h_3 + Q_d = m_4h_4 + m_7h_7$$

Condenser:

$$m_7 = m_8$$

$$x_7 = x_8$$

$$m_7h_7 = m_8h_8 + Q_c$$

Expansion valve:

$$m_8 = m_9$$

$$x_8 = x_9$$

$$h_8 = h_9$$

Evaporator:

$$m_9 = m_{10}$$

$$x_9 = x_{10}$$

$$m_9 h_9 + Q_e = m_{10} h_{10}$$

Equation 23 - Equations of mass and energy balances of the individual components of the single stage

The two-stage system is created by doubling the high pressure part of the single-stage system. In comparison with the single-stage system, the two-stage system is composed of two desorbers (high temperature and low temperature), which are connected with two condenser (high temperature and low temperature). The pumps, expansion valves, and heat exchangers are doubled as well. The propulsion heat in the system is first supplied to the high temperature desorber. The heat also transfers between the condenser with a high temperature level and the desorber with a low temperature level. The Dühring diagram of the two-stage absorption system is shown in Figure 9 (Herold, K. et al. 1996).

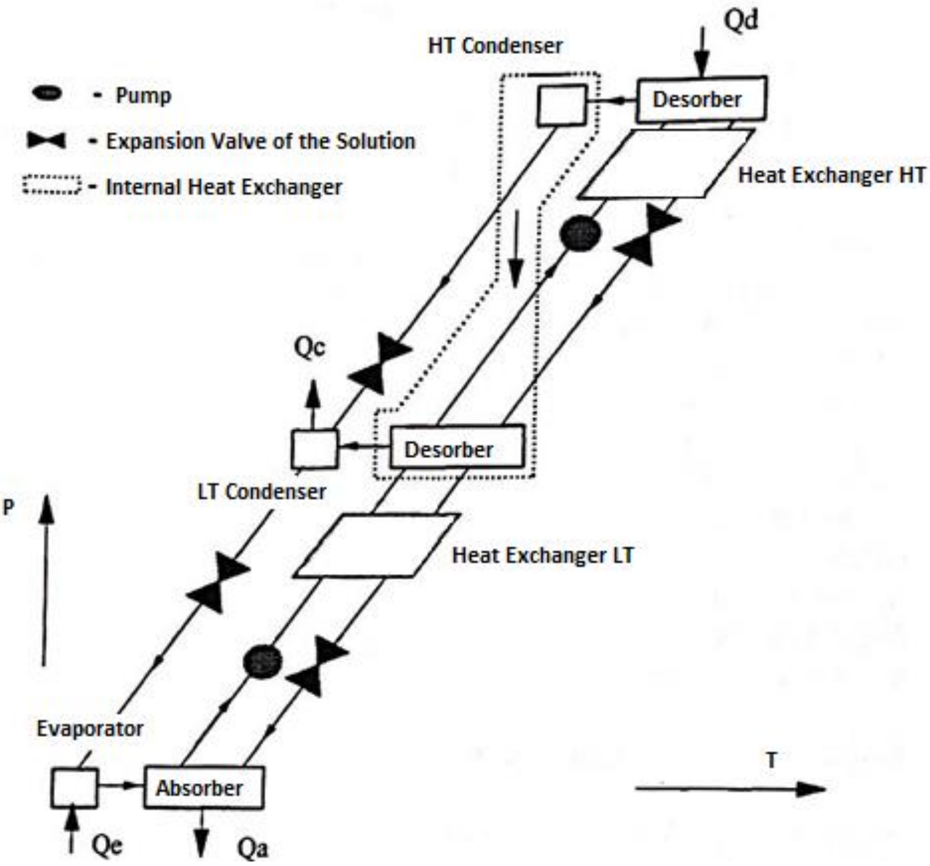

Figure 9 - Dühring diagram of the two-stage absorption cooling system (LiBr + H_2O) (Herold, K. et al. 1996).

The standard single-stage system works with a COP_R factor of around 0,72. The higher level pressure is usually about 7 kPa, the lower level roughly ten times less. The two-stage system uses propulsion mediums with higher pressure and temperature levels thanks to the division of the desorber into higher and lower temperature parts. With this, the cycle efficiency is raised. The raised cycle efficiency affects the cooling factor COP_R, nearly doubling it. This means that it reaches a value of 1,4. The highest-pressure level is around 64 kPa. The pressure of the middle part is usually approximately 4 kPa, and the low pressure part works at a level of 0,8 kPa (Herold, K. et al. 1996).

The cooling cycle of the absorption unit is described in the diagram of the component placement in the absorption unit (Figure 10). The cooled water enters the pipes of the evaporator (marked with the number 3 in the figure). The pipes in the evaporator are sprayed with a refrigerant (marked with the green color in the figure), which is led from the refrigerant container through the refrigerant pump into the evaporator. The refrigerant absorbs the heat from the cooled water and evaporates. In the joint container, the refrigerant vapor then passes into the area of the absorber (marked with the number 4 in the figure). The refrigerant vapor is then absorbed in the absorber with the gradually sprayed concentrated aqueous lithium bromide solution (marked with dark red in the figure). As a result of the absorption of the refrigerant vapor, the solution heats up and becomes diluted. The solution then transfers the heat into the pipes with cooling water, which enters the absorber. The cooling water then leads the waste heat out of the cooling unit. The diluted LiBr solutions settle at the bottom of the absorber (marked with bright red in the figure). It is then pumped into the desorber (generator/ejector; marked with the number 1 in the figure) through the low-temperature regenerator (regenerative heat exchanger, marked with the number 5 in the figure) with the LiBr solution pump (marked with number 6 in the figure). A pipe with the heating medium (marked with red in the picture) enters the desorber (generator/ejector), where it transfers its heat to the diluted LiBr solution. Because of this, the diluted LiBr solution then heats up and releases water vapor, which then passes through the joint container into the area of the condenser (marked with the number 2 in the figure). The concentrated LiBr solution has a higher density, and it settles at the bottom of the solution container (in the generator/ejector), from where it is pumped through the low-temperature regenerator back into the absorber. In the condenser, the water vapor from the desorber (generator/ejector) transfers its heat energy to the pipes with the cooling water. The water vapor then changes its state into a liquid (and is further used for cooling – the refrigerant from the condenser then also enters the evaporator, where it sprays the pipes with the cooled water). Then the heat is led out of the cooling unit with the circuit of cooling water. The low-temperature regenerator is vital for achieving a high cycle efficiency. The low-temperature regenerator removes the excess heat from the concentrated LiBr solution which is passing from the generator into the diluted LiBr solution in the absorber. With this, the consumption of energy required for heating is reduced, and at the same time, the cooling output required for the removal of waste heat is lowered.

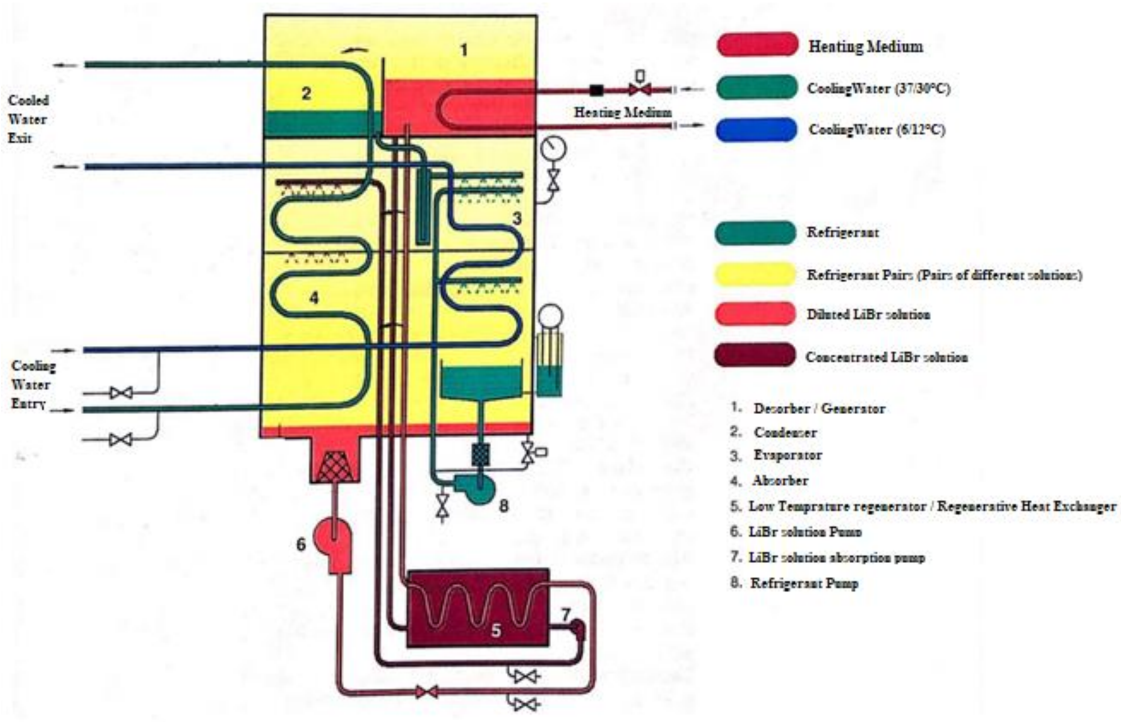

Figure 10 – The structural arrangement of the single stage absorption device (Sokra)

The following figure shows a diagram of a directly heated two-stage absorption system. When compared to the single stage absorption system, the vapor ejected from the high-temperature desorber is led into the area of the low-temperature desorber, where it heats up the solution and only after that is it led into the area of the condenser. Because the two-stage absorption system has two desorbers, it needs two regenerators to function accurately.

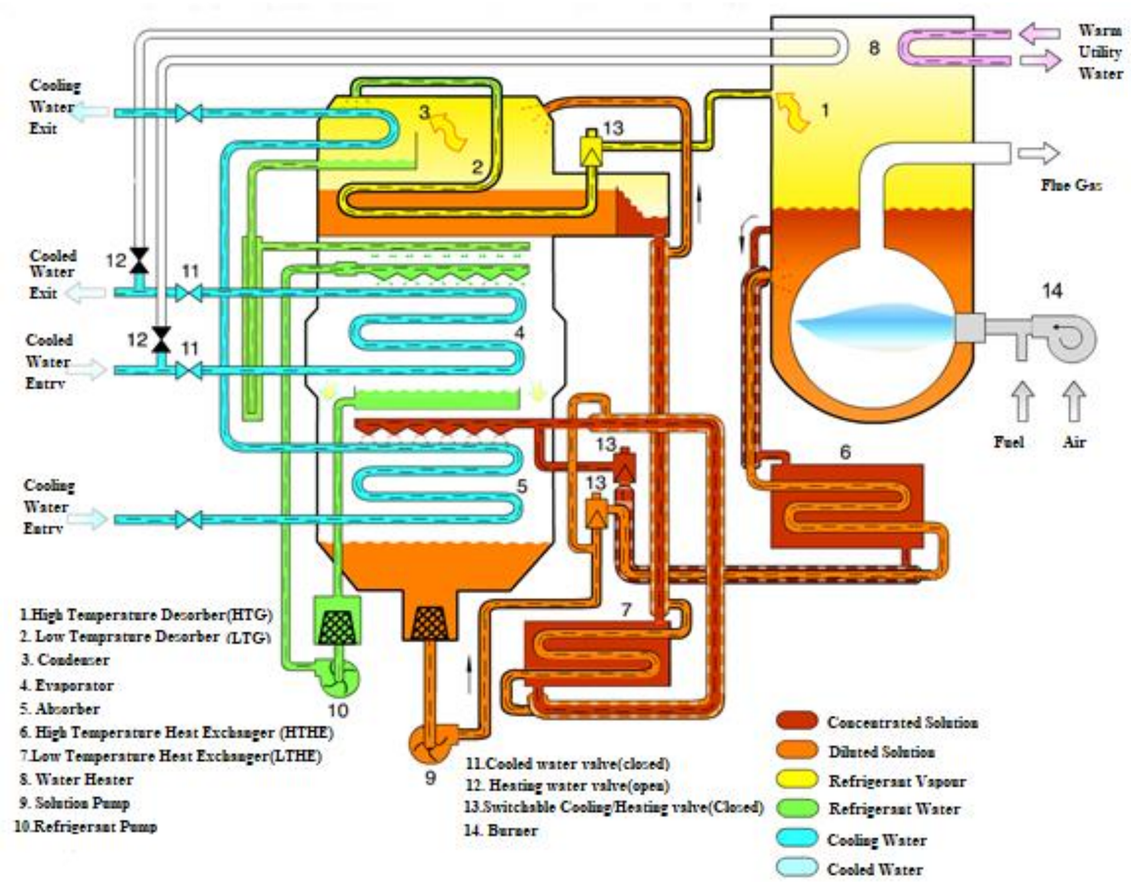

Figure 11 – The structural arrangement of the two-stage absorption device (Broad.cz**)**

3.2. Pressure micro turbine

In order to lower the pressure of the technological steam, industrial enterprises often use an outdated but simple and cheap solution, which consists in applying a reduction valve. This way, the parameters of the steam at the entry pressure level are changed (the pressure is determined by the parameters of the steam generator or by the sampling stages of heat turbines) into a lower pressure needed in technology. The reduction valve (RV) connected, as shown in Figure 12, changes the steam with the mass flow M from the pressure p_1 into the pressure p_2. Because the throttling can be considered as an isenthalpic change, the enthalpy h_1 is identical to the enthalpy at the exit of the reduction valve.

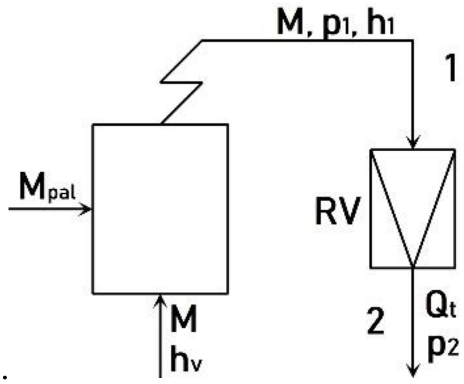

Figure 12 - Reductive valve

The reduction valve is substituted with a micro turbine, which is a turbine machine that can achieve the needed pressure reduction of the working medium for performing electrical work. The reduction of the pressure causes a lower qualitative level of the working medium's energy – this decrease in the qualitative level, however has a minimal influence on the heat supply, which mainly uses the condenser heat of the heat-carrying medium.

3.2.1. Micro turbine balance

The pressure micro turbine is generally connected in parallel connection to the reduction valve, as shown in Figure 13. The micro turbine, connected between points 1 and 3, changes the parameters of the steam with the mass flow M' from the pressure p_1 to the pressure p_2 and enthalpy h_3. A part of the steam energy corresponding to the difference between the enthalpy h_1 and h_3 with the set mass flow is then changed with the micro turbine and the generator into electrical energy. In order to preserve the supply of the heat output Q_t, the mass flow of the fuel M'_{fuel} has to be increased for the generation of a higher steam mass flow M'. In the event of a power outage of the micro turbine or the generator, the parallel reduction valve is used to prevent an interruption of the steam supply.

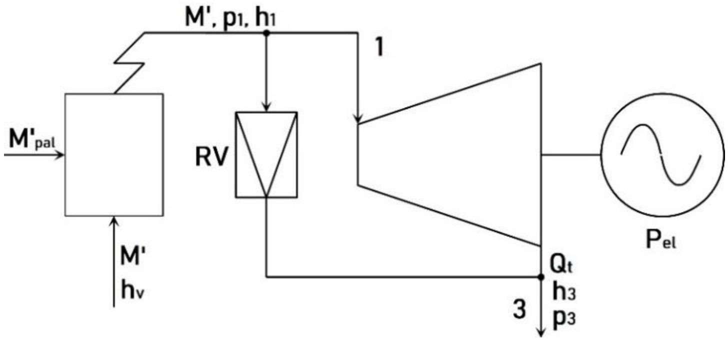

Figure 13 - Pressure micro turbine

The following picture describes a situation in the h-s diagram of the steam. The state marked with the number 1 corresponds to the entry parameters of the steam entering the reduction valve or the micro turbine. The point marked with the number 2 is the state at the exit of the reduction valve. The point marked with the number 3 corresponds to the exit of the micro turbine. The point marked with the number 4 corresponds to the exit from the micro turbine with a perfect efficiency, which is represented with an isentropic change from the state 1 into the state 4. The supplied heat output into the technology Q_t then corresponds to the transfer from state 3 (or state 2 during the usage of the reduction valve) into state 5.

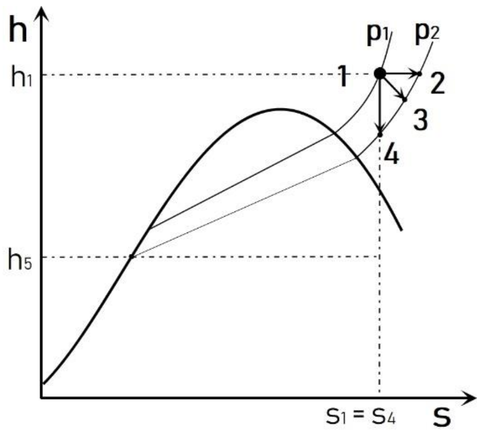

Figure 14 - H-s diagram of the change of pressure with throttling and with the micro turbine

The supplied heat Q_t has to be preserved after the installation of the micro turbine as shown in Equation 24. The supplied heat is proportional to the steam mass flow and to the difference between enthalpies (Šťastný, J., 2006).

$$Q_t = M \cdot (h_1 - h_5) = M' \cdot (h_3 - h_5) = (M + \Delta M) \cdot (h_3 - h_5)$$

Equation 23- Supplied heat Q_t proportionality

The following relation (Equation 25) describes the electrical output on the generator clamps, which is determined by the steam mass flow, the efficiency, and the utilized enthalpy gradient in the micro turbine.

$$P_{el} = \eta \cdot M' \cdot (h_1 - h_3)$$

Equation 24 - The electrical output on the generator clamps

With a few adjustments, it is possible to get the relation (Equation 26), which describes the essential advantage of the production of electrical energy using the pressure micro turbine. The electrical output, after neglecting the generator efficiency, corresponds to the change of the fuel consumption (Šťastný, J., 2006).

$$P_{el} = \eta \cdot (M' - M) \cdot (h_1 - h_5) = \eta \cdot \Delta M \cdot (h_1 - h_5) = \eta \cdot \Delta Q_{pal}$$

Equation 25 - Essential advantage of the production of electrical energy

The price of the micro turbine is determined mainly by its efficiency. This corresponds to the ratio of the utilized enthalpy gradient to the theoretical enthalpy gradient during an isentropic change between the entry state of the steam and the required exit pressure. Although the efficiency limits the produced electrical energy, it has no influence on the pressure reduction, and with higher thermodynamical efficiency, the additional costs of steam are raised. The choice of efficiency is then based on the economic evaluation of investment costs for the micro turbine, costs for the increased steam generation, and the income from the created electrical energy.

3.3. Recuperative heat exchangers

The heat exchangers are critical elements of heat systems. In heat exchangers, a part of the heat energy of the heat-carrying medium with a higher temperature level (heating medium) is transferred to the heat-carrying medium with a lower temperature level (heated medium). The calculation of the heat exchanger is based on the energy balance of the exchanger and on the equation for calculating the heat flow using the permeation of heat through the wall of the heating surface between both heat-carrying mediums (Černý, V., 1986).

3.3.1. Energy balance of the heat exchanger

The energy balance of the heat exchanger assembled according to the Figure 15 corresponds to the Equation 27, where $\dot{m}_1$ and $\dot{m}_2$ are mass flows of the flowing heat-carrying fluids, c_1 and c_2 are their specific heat capacities, $t_{1,1}$ and $t_{2,1}$ are their entry temperatures and $t_{2,1}$ and $t_{2,2}$ correspond to their exit temperatures. In the equation, heat dissipation is the member Q_z. Usually, the heat dissipation of the exchangers is not considered.

$$\dot{m}_1 \cdot c_1 \cdot (t_{1,1} - t_{1,2}) = \dot{m}_2 \cdot c_2 \cdot (t_{2,1} - t_{2,2}) + Q_z$$

Equation 26 - Heat dissipation

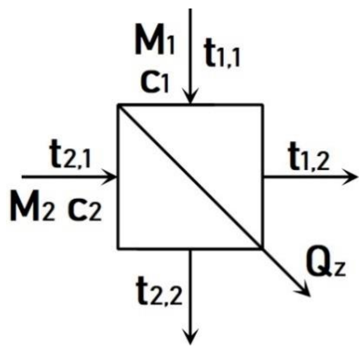

Figure 15 – Energy balance of the heat exchanger

3.3.2. Heat exchange surface area and logarithmic mean temperature difference (LMTD)

In regenerative heat exchangers, the heat output transfers from the heat-carrying fluid to the heat exchange surface area using convective heat transfer. The transfer through the wall from one surface to another is realized with heat conduction, where the heat output is transferred to the heat-carrying fluid again using convective heat transfer.

The heat flux density q in a steady state can be described with the heat resistances R_1, R_s and R_2, which are determined with the coefficients of the heat transfer by convective heat transfer α_1 and α_2 ($W \cdot m^{-2} \cdot K^{-1}$), with the wall thickness d and with the heat conductivity of the wall λ ($W \cdot m^{-1} \cdot K^{-1}$). The heat resistances can be collectively marked as the heat transfer coefficient h (Kyncl, J., n.d.).

$$q = \frac{(t_1 - t_2)}{R_1 + R_s + R_2} = \frac{(t_1 - t_2)}{\frac{1}{\alpha_1} + \frac{d}{\lambda} + \frac{1}{\alpha_2}} = k \cdot (t_1 - t_2) \qquad (W \cdot m^{-2} \cdot K^{-1})$$

Equation 27 - Heat flux density

The transferred heat output Q from the heat-carrying medium to the heat-carrying medium is described by Equation 29, where $\overline{\Delta t}$ is the logarithmic mean temperature difference (LMTD).

$$Q = k \cdot S \cdot \overline{\Delta t} = k \cdot \int_0^S (t_1 - t_2) \cdot dS = k \cdot S \cdot \frac{(\Delta_1 - \Delta_2)}{ln\frac{\Delta_1}{\Delta_2}} \quad (W)$$

Equation 28 - The transferred heat output

The logarithmic mean temperature difference (LMTD) is dependent on the type of heat exchangers. The general connection of the regenerative heat exchangers is the co-current flow and countercurrent flow. The course of the countercurrent and co-current flow temperature difference is shown in Figure 16. For the countercurrent flow heat exchanger, the temperature difference $\Delta_1 = (t_{1,2} - t_{2,1})$ and $\Delta_2 = (t_{1,1} - t_{2,2})$, whereas for the cocurrent flow heat exchanger, the temperature difference $\Delta_1 = (t_{1,1} - t_{2,1})$ and $\Delta_2 = (t_{1,2} - t_{2,2})$.

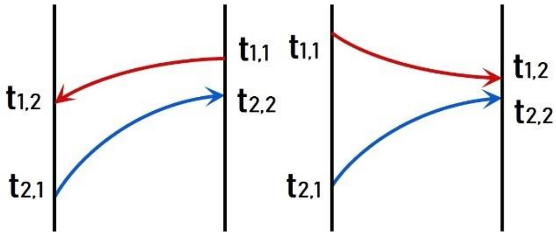

Figure 16 – The temperature gradient of the countercurrent connection (left) and co-current connection (right)

Crossflow heat exchangers have more complex temperature ratios, as the temperatures do not only change in the direction of the flow of the set medium, but also in the direction of the flow of the second heat-carrying medium. In practice, the crossflow heat exchangers are calculated similarly as the countercurrent flow heat exchangers with the use of the LMTD correction coefficient factor ψ according to the Equation 30, in which $\overline{\Delta t_{cf}}$ is the logarithmic mean temperature difference (LMTD) of the countercurrent flow heat exchanger (Černý, V., 1986).

$$\overline{\Delta t} = \psi \cdot \overline{\Delta t_{cf}}$$

Equation 29 - LMTD countercurrent heat exchanger

The LMTD correction coefficient factor $\psi = f(P, R)$ depends on the way of the flow of the heat-carrying mediums and on their temperature ratios, which are described by the values of P and R according to Equation 31. The value of the LMTD correction coefficient factor can be determined by reading the graph (Figure 18), where the dependencies of the two types of heat exchangers are shown.

$$P = \frac{t_{2,2} - t_{2,1}}{t_{1,1} - t_{2,1}}$$

$$R = \frac{t_{1,1} - t_{1,2}}{t_{2,2} - t_{2,1}}$$

Equation 30 - Dependencies of the 2 types of heat exchangers

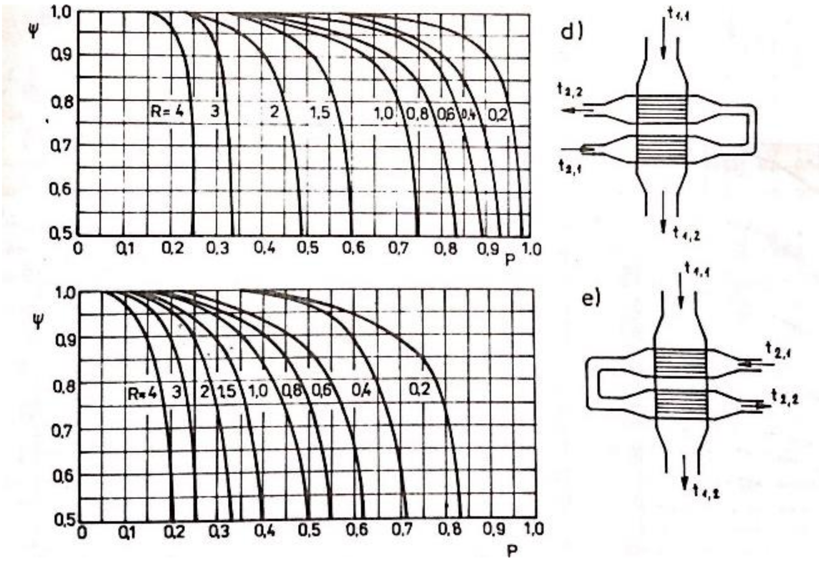

Figure 17 - Dependency of the LMTD correction coefficient ψ on P and R (Černý, V., 1986).

4. AZERBAIJAN CLIMATE

Azerbaijan is mainly located in the subtropic climate zone. While 65 percent of the country locates in the subtropic climate zone, 32 percent locates in the temperature zone, and the rest in the cold climate zone (Sputnik, 2020). This creates a possibility for a high number of sunny days in the country. Approximately, there are 5-8 hours daily sunshine and 1900-2900 hours of sunshine annually (Museyibov, M. A., 1998).

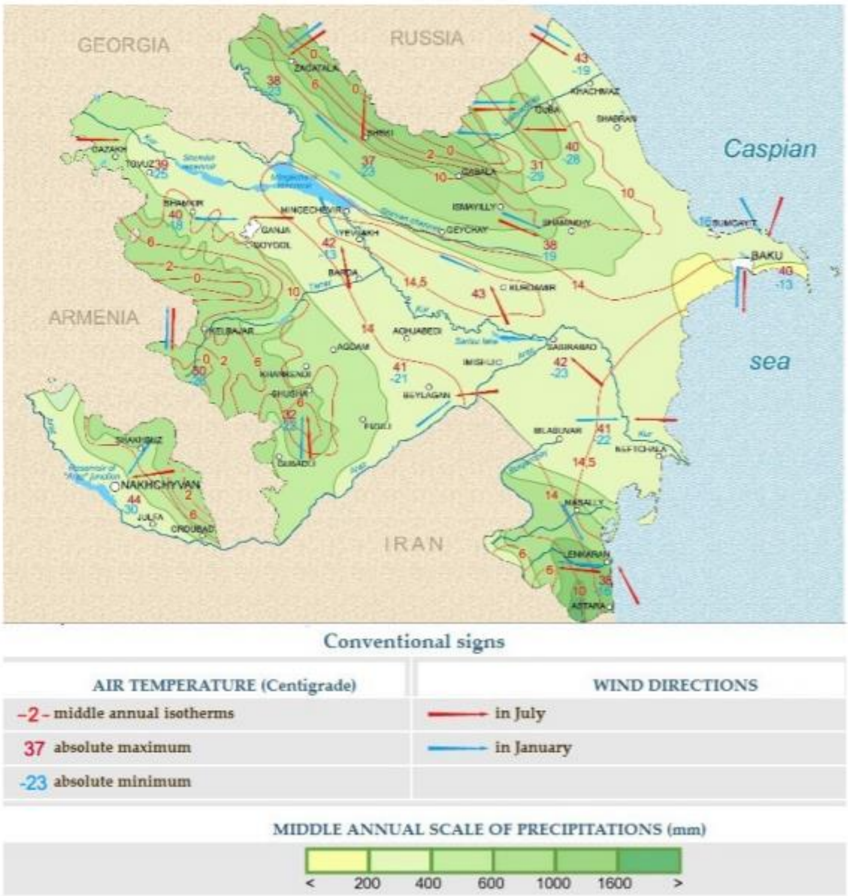

Figure 18 - Climate Map of Azerbaijan (Safaraliyev, R., 2015).

Azerbaijan's capital city is Baku. Summers are warm, humid, and dry, and winters are cold and occasionally wet in Baku. The city is also known for its strong winds all year along (Guide to Baku). Daily temperature in summer averages 26.4 °C. The warmest month is July, with an average of 27.9 °C. Daily temperature in winter averages 4.3 °C. The coldest month is January, with an average of 1.7 °C (Safaraliyev, R., 2015).

5. CALCULATIONS

For this thesis, Heat Pump Electricity Consumption and Accumulated Energy consumption for the last three years (2017, 2018, 2019) are calculated for the data center located in Baku. In order to make these calculations, Energy Balance of the cooling system is also calculated. The demand power for the chosen data center is provided as 2x893,6kW.

To calculate the Energy Balance weather data for the 2017, 2018 and 2019 were used (Weather Archive in Baku). The weather was measured every 30 minutes at Heydar Aliyev International Airport in Baku. In 2017, the coldest temperature was measured on 18 February with -5,9 °C, and the warmest temperature was measured on 8 August with 39,8 °C (Figure 19). In 2018, the coldest temperature was measured on 02 February with -1,7 °C, and the warmest temperature was measured on 1 July with 49 °C (Figure 20). In 2019, the coldest temperature was measured on 20 January with -3 °C, and the warmest temperature was measured on 24 June with 40 °C (Figure 21).

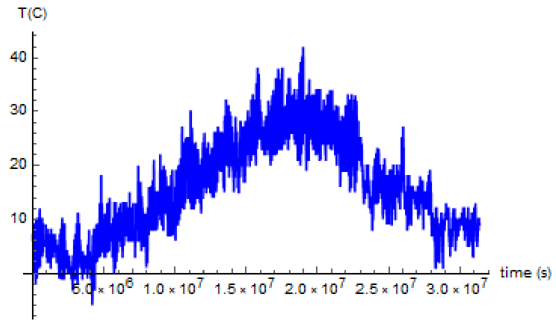

Figure 19 - Temperature in Baku during 2018

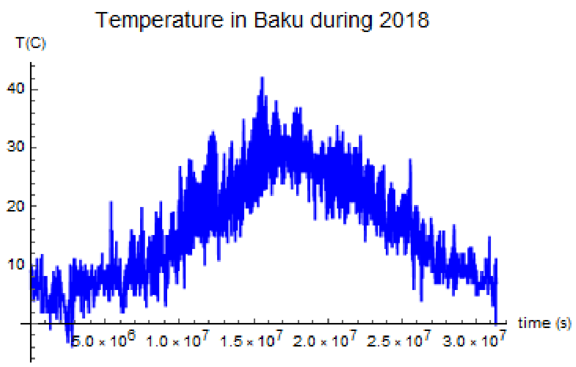

Figure 20 - Temperature in Baku during 2018

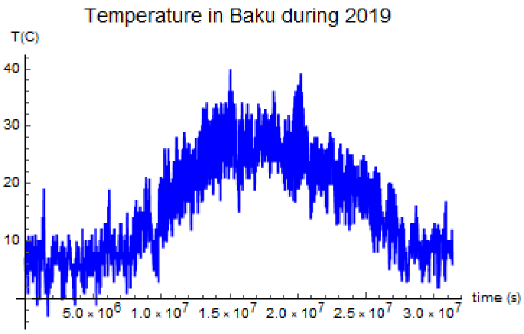

Figure 21 - Temperature in Baku during 2019

As a cooling method, CRAC (Computer Room Air conditioner) will be used to conduct further calculations. The average suitable temperature for the data center is considered as 20 °C. The air conditioner is going to be used as a ventilator, while the outside temperatures lower than 20 °C, which uses relatively less energy, and therefore, will be neglected in further calculations. As soon as the outside temperature rises more than 20 °C, the air conditioner will start cooling to keep devices at a suitable temperature.

Formulas for the cooling factor $COP_{R,}$ for this case and the relation between P_{el} and $P_{cooling}$ (compressor heat pump), are shown in Equation 32.

$$COP_R = 0{,}5 * \frac{T_{outside} + 273}{\Delta T} - 1$$

$$COP_R * P_{el} = P_{cooling}$$

$$P_{el} = \frac{P_{cooling.required}}{COP_R(T_{outside})}$$

Equation 31 - Cooling Factor and its relation between P_{el} and $P_{cooling}$

The accumulated energy consumptions for 2017, 2018, and 2019 are shown in the below figures (Figure 22,23,24). The highest energy consumption is observed in 2018, while the lowest energy consumption is observed in 2019 (Table 1). The average accumulated energy consumption for these three years is 218,099MWh.

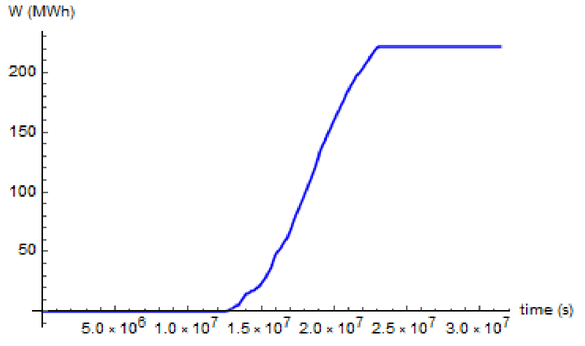

Figure 22 - Accumulated energy consumption in 2017

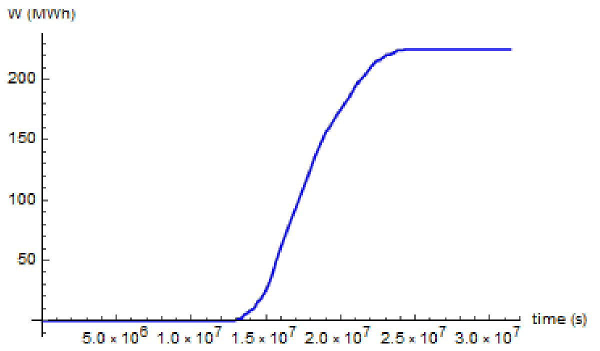

Figure 23 - Accumulated energy consumption in 2018

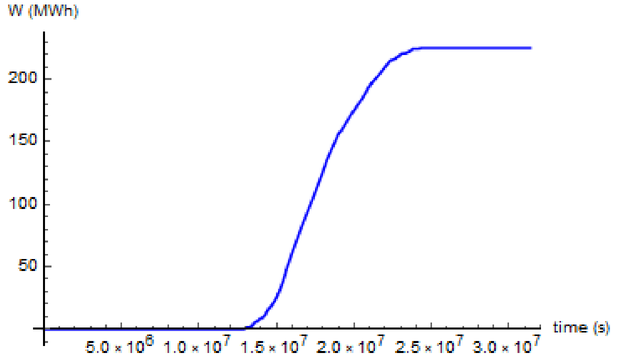

Figure 24 - Accumulated energy consumption in 2019

Year	2017	2018	2019	Mean
Energy (MWh)	221.634	225.073	207.591	218.099

Table 1 - Yearly accumulated energy consumption

The heat pump electricity consumptions in 2017, 2018, and 2019 are shown in below figures (Figure 25, 26, 27).

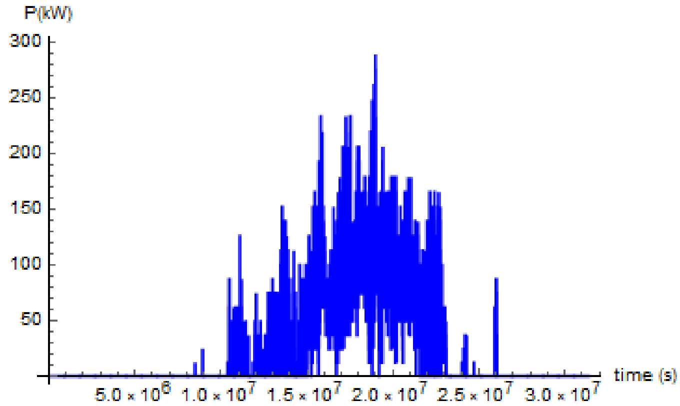

Figure 25 - Heat pump electricity consumption in 2017

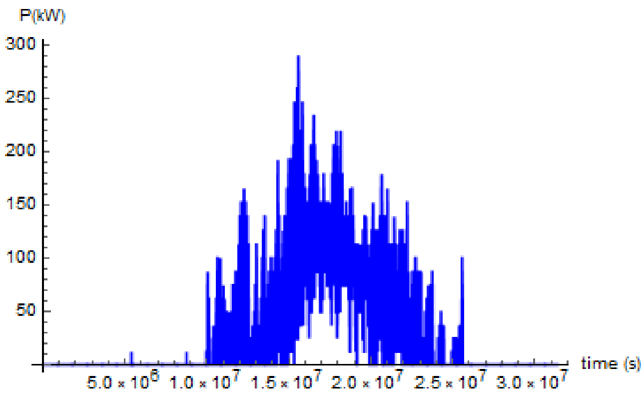

Figure 26 - Heat pump electricity consumption in 2018

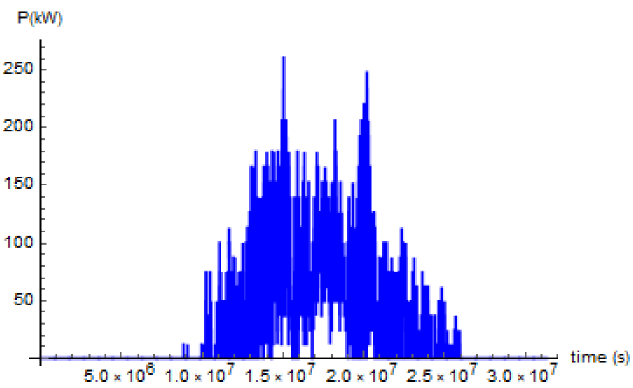

Figure 27 - Heat pump electricity consumption in 2019

6. CONCLUSION

Nowadays, data centers are crucial equipment for many businesses. They consume a lot of energy, and they need a cooling system to keep them at a suitable temperature, especially when the outside weather is above 20°C. Many different cooling systems can be applied to the data center. The calculations on this thesis are mainly focused on the Computer Room Air Conditioner (CRAC) method, which is inefficient. However, it is also very inexpensive to install.

In the second chapter, general principles such as conservation laws, Carnot cycle, and pressure losses are discussed. These principles are the foundation of thermodynamics. Therefore, they are always considered during calculating energy balance and energy consumption.

In the third chapter, heat engines and heat exchangers are explained. Also, the necessity of heating and cooling factors for the devices needed to build the data center are discussed.

In this thesis, the calculations were focused on a data center located in Baku, the capital city of Azerbaijan. While winter is cold in this city, the summers are very hot.

In calculations, it was observed that the highest energy consumption occurred in 2018, while the lowest consumption was in 2019. The overall average consumption is considered to be high. One of the reasons for this is the chosen cooling system CRAC. Another reason is the high temperatures during summertime in Baku.

For this data center, it can be suggested to use another cooling system, or move the data center to less warm areas in Azerbaijan.

7. REFERENCES

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APPENDIX

```
In[=]:= Remove["Global`*"]

In[=]:= SetDirectory[NotebookDirectory[]];

        data2017 = Import["baku2017.xls", "Data"]; (* data import *)

        data2018 = Import["baku2018.xls", "Data"]; (* data import *)

        data2019 = Import["baku2019.xls", "Data"]; (* data import *)

In[=]:= data2017temp = data2017[[1, 8 ;;, 2]];

In[=]:= t2017 = Range[Length[data2017temp] * 30 * 60 - 30 * 60, 0, -30 * 60];

In[=]:= temp20172 = Thread[{t2017, data2017temp}];
        temp2017 = temp20172[[Range[Length[temp20172], 1, -1]]];

In[=]:= ListPlot[temp2017]

In[=]:= data2018temp = data2018[[1, 8 ;;, 2]];
        t2018 = Range[Length[data2018temp] * 30 * 60 - 30 * 60, 0, -30 * 60];
        temp20182 = Thread[{t2018, data2018temp}];
        temp2018 = temp20182[[Range[Length[temp20182], 1, -1]]];

In[=]:= ListPlot[temp2018]

        data2019temp = data2019[[1, 8 ;;, 2]]; (* select temperatures from data*)
        t2019 = Range[Length[data2019temp] * 30 * 60 - 30 * 60, 0, -30 * 60];
        temp20192 = Thread[{t2019, data2019temp}];
        temp2019 = temp20192[[Range[Length[temp20192], 1, -1]]];

In[=]:= temp2019 // Length

In[=]:= minlen = Min[Length /@ {temp2017, temp2018, temp2019}];

In[=]:= pload = 893.6 * 2 * 1000; (* load consumption of data center *)

In[=]:= k[t_] := 0 /; t <= 20 (* heat pump works  if outside temperature is less than 20 deg*)
        k[t_] := 1 /; t > 20
```

$$\texttt{cop[t\_] := 0.5}\ \frac{t+273}{t-20} - 1\ /;\ t > 20\ \texttt{(* COP is a f of temperature*)}$$

```
        cop[t_] := 1 /; t ≤ 20

In[=]:= pel[temp_] := k[temp] * pload / cop[temp]

In[=]:= power2017 = {#[[1]], pel[#[[2]]]} & /@ temp2017;

In[=]:= intpower2017 = Interpolation[power2017]

In[=]:= pltcop = Plot[cop[t], {t, 20.1, 50}, PlotRange → {All, {0, 80}},
          AxesOrigin → {20, 0}, AxesLabel → {"Tout  (C)", "COP  (-)"}, AxesStyle → Black]

In[=]:= Export["cop.png", pltcop]

In[=]:= tmax2017 = power2017[[-1, 1]];

In[=]:= energy2017 = x /. First@NDSolve[{x'[t] == intpower2017[t], x[0] == 0}, x, {t, 0, tmax2017}]
            (* energy consumed during year*)
```

```
In[•]:= pltenergy2017 = Plot[energy2017[t] / (1000000 * 3600), {t, 0, tmax2017},
          AxesLabel → {"time (s)", "W (MWh)"}, AxesStyle → Black, PlotStyle → Blue,
          PlotLabel → Text[Style["Accumulated energy consumption in 2017", Black, 14]]]

In[•]:= Export["pltenergy2017.png", pltenergy2017]

In[•]:= power2018 = {#[[1]], pel[#[[2]]]} & /@ temp2018;

In[•]:= intpower2018 = Interpolation[power2018]

In[•]:= tmax2018 = power2018[[-1, 1]];

In[•]:= energy2018 = x /. First@NDSolve[{x'[t] == intpower2018[t], x[0] == 0}, x, {t, 0, tmax2018}]

In[•]:= pltenergy2018 = Plot[energy2018[t] / (1000000 * 3600), {t, 0, tmax2018},
          AxesLabel → {"time (s)", "W (MWh)"}, AxesStyle → Black, PlotStyle → Blue,
          PlotLabel → Text[Style["Accumulated energy consumption in 2018", Black, 14]]]

In[•]:= Export["pltenergy2018.png", pltenergy2018]

In[•]:= power2019 = {#[[1]], pel[#[[2]]]} & /@ temp2019;

In[•]:= intpower2019 = Interpolation[power2019, InterpolationOrder → 2]

In[•]:= tmax2019 = power2019[[-1, 1]];

In[•]:= energy2019 = x /. First@NDSolve[{x'[t] == intpower2019[t], x[0] == 0}, x, {t, 0, tmax2019}]

In[•]:= pltenergy2019 = Plot[energy2018[t] / (1000000 * 3600), {t, 0, tmax2018},
          AxesLabel → {"time (s)", "W (MWh)"}, AxesStyle → Black, PlotStyle → Blue,
          PlotLabel → Text[Style["Accumulated energy consumption in 2019", Black, 14]]]

In[•]:= Export["pltenergy2019.png", pltenergy2019]

In[•]:= energy2017[tmax2017]

In[•]:= energy2018[tmax2018]

In[•]:= energy2019[tmax2019]

In[•]:= pltpower2017 =
         ListLinePlot[power2017 /. {t_, p_} :→ {t, p / 1000}, AxesLabel → {"time (s)", "P(kW)"},
          AxesStyle → Black, PlotRange → All, PlotStyle → Blue, AxesOrigin → {0, 0},
          PlotLabel → Text[Style["Heat pump electricity consumption in 2017", Black, 14]]]

In[•]:= Export["pltpower2017.png", pltpower2017]

In[•]:= pltpower2018 =
         ListLinePlot[power2018 /. {t_, p_} :→ {t, p / 1000}, AxesLabel → {"time (s)", "P(kW)"},
          AxesStyle → Black, PlotRange → All, PlotStyle → Blue, AxesOrigin → {0, 0},
          PlotLabel → Text[Style["Heat pump electricity consumption in 2018", Black, 14]]]

In[•]:= pltenergy2019 = Plot[energy2019[t] / (1000000 * 3600), {t, 0, tmax2019},
          AxesLabel → {"time (s)", "W (MWh)"}, AxesStyle → Black, PlotStyle → Blue,
          PlotLabel → Text[Style["Accumulated energy consumption in 2019", Black, 14]]]

In[•]:=
```

```
In[•]:= Export["pltpower2018.png", pltpower2018]

In[•]:= pltpower2019 =
         ListLinePlot[power2019 /. {t_, p_} :> {t, p / 1000}, AxesLabel → {"time (s)", "P(kW)"},
          AxesStyle → Black, PlotRange → All, PlotStyle → Blue, AxesOrigin → {0, 0},
          PlotLabel → Text[Style["Heat pump electricity consumption in 2019", Black, 14]]]

In[•]:= Export["pltpower2019.png", pltpower2019]

In[•]:= plttemp2017 = ListLinePlot[temp2017, AxesLabel → {"time (s)", "T(C)"},
          AxesStyle → Black, PlotStyle → Blue, AxesOrigin → {0, 0},
          PlotLabel → Text[Style["Temperature in Baku during 2017", Black, 14]]]

In[•]:= Export["plttemp2017.png", plttemp2017]

In[•]:= plttemp2018 = ListLinePlot[temp2018, AxesLabel → {"time (s)", "T(C)"},
          AxesStyle → Black, PlotStyle → Blue, AxesOrigin → {0, 0},
          PlotLabel → Text[Style["Temperature in Baku during 2018", Black, 14]]]

In[•]:= Export["plttemp2018.png", plttemp2018]

In[•]:= plttemp2019 = ListLinePlot[temp2019, AxesLabel → {"time (s)", "T(C)"},
          AxesStyle → Black, PlotStyle → Blue, AxesOrigin → {0, 0},
          PlotLabel → Text[Style["Temperature in Baku during 2019", Black, 14]]]

In[•]:= Export["plttemp2019.png", plttemp2019]

In[•]:= table = TableForm@Grid[{{"Year", "2017", "2018", "2019", "Mean"}, {"Energy (MWh)",
              1/(3600 * 1 000 000) energy2017[tmax2017], 1/(3600 * 1 000 000) energy2018[tmax2018],
              1/(3600 * 1 000 000) energy2019[tmax2019], 1/(3600 * 1 000 000) Mean[{energy2017[tmax2017],
                energy2018[tmax2018], energy2019[tmax2019]}]}}, Frame → All]

In[•]:= Export["tableenrgy.png", table]
```