Department of Physics and Technology

Thermal Energy Recycling at Elkem Salten Verk

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Abstract

This thesis elucidates thermal energy transfer at Elkem Salten Verk, and tries to determine the most favorable approach on how to provide indoor heating by thermal energy for a building at the plant site. My goal for this thesis was to create some suggestions, regarding the type of heating systems and their energy rates in order to provide enough heating power for Miljøbygget.

Acknowledgments

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Erik Ro Rundhaug
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Chapter 1

Introduction

The background for this thesis is the United Nation's (UN), the European Union's (EU) and the Norwegian Government's focus on decreasing the global CO_2 emissions. As a result of this Norway has set some long term goals and commitments to decrease the nations pollution emissions. One segment of these commitments and goals is Enova. The Norwegian Parliament enacted to establish Enova in 2001 in order to enhance the usage of green energy sources applied in domestic and industrial heating. Enova is managing the energy fund, and has as one of its main-goals to encourage a shift to green energy sources for heating through support funding, research, and working with consciousness- raising in schools. As an additional support funding option a norwegian bank called "Husbanken" has created a special mortgage called "grunnlån" to motivate the utilization of green energy heating in new building projects.. The mortgage have some very specific demands for heat loss of buildings. These standards are higher than any current building standards for new buildings (TEK-10). Both of these above mentioned financial aids are a result of the Norwegian Parliaments goals to reduce the Norwegian electrical consumption by half for heating by the year 2040. With these guidelines from the government in mind Elkem Salten wants to decrease their own consumption of electrical power for heating by utilizing the thermal energy available at the plant [6, 26].

Elkem Salten Verk is a part of Elkem AS, which again are owned by China National Bluestar. Elkem Salten is located in the Sørfold municipality, about 82km north east of the city of Bodø.

Figure 1.1: Overview of Elkem Salten verk [16]

Figure 1.1 shows an overview picture of Elkem Salten Verk [16].

The plants three submerged arc furnaces were built in the mid 1960's, the first furnace started production in 1967 and the third furnace in 1972. The initial production was mainly ferrosilica (FeSi75). [15]

The main products currently produced at the plant are Silicon 97 and Silicon 99 used in chemical industry, Microsilica products for concrete and refractories and ®Sidistar mainly used for rubber and thermoplastics. The plant is one of the world's largest silicon plant with a production capacity of about 80,000 tons silicon/year [14]. Elkem Salten Verk produces vast amounts of heat as a by-product of their production. This thermal energy is currently utilized for some heat recycling, but Elkem Salten Verk wants to enhance the percentage of recycled thermal energy, in order to reduce the electrical consumption for heating at the plant site.

For my thesis I chose the problem of a feasibility study at Elkem Salten Verk on the possibilities to create a heat central which uses thermal energy from furnace #1, furnace#2 and furnace#3. The thermal energy is then used for heating Miljøbygget. Throughout this thesis I investigate the different methods on how to transfer thermal energy, and try to determine the most favorable approach to heat Miljøbygget with thermal energy.

1.1 Structure of the thesis

This thesis is consists of seven chapters.

Chapter one is the introduction. Chapter two is the theoretical foundation of the thesis. Chapter three is a description of Elkem Salten Verks plant, and the cooling system. Chapter four treats the energy balance for the cooling system and Miljøbygget. Chapter five treats the suggested heating systems more detailed, it also contains the economical analysis. Chapter six is the discussion of my results from chapter four and five. Chapter seven is the conclusions for this thesis.

Chapter 2

Theory

This chapter of the thesis contains the theory associated with the calculations and the general foundation of this thesis.

2.1 Water as refrigerant

Water is one of the oldest and least expensive refrigerants. In addition to a low global warming potential (GWP) and a low ozone depletion potential (ODP) in comparison with refrigerants such as R12(synthetic), R22(synthetic), R717(Ammonia), R290(synthetic), R134a (synthetic)and R152a (natural). Table 4.1 displays a comparison of water (R718) with the above mentioned refrigerants [20]. Hydrocarbon and halocarbon refrigerants are named by their contents of Carbon, Hydrogen and Fluorine in a Rxyz system. R is refrigerant, x is the number of Carbon atoms minus one atom, y is the number of Hydrogen atoms pluss one atom, and z is the number of Fluorine atoms. This system applies for refrigerants consisting of only one type of working fluids. If two different refrigerants are mixed in such a concentration that they have the same boiling and condensation temperature as another single refrigerant, they are called azeotropes. And have the numbers from R500 and upwards. Azeotrope refrigerants will have condensation and evaporation temperatures at a fixed pressure. Zeotrope refrigerants are refrigerants that are soluble with each other and will therefore have a condensation- and evaporation temperature interval. Zeotrope refrigerants are number between R400 and R500. Refrigerants with the letters A,B and C after their refrigeration numbers are isomers. Inorganic refrigerants such as water, Carbon-dioxide, Ammonia and Sulfuric-acid has the designation R7 followed by their molecular weight[29].

Refrigerant name	ODP	GWP	Safety group
R718	0	0	A1
R717	0	0	B2
R12	1	8500	A1
R22	0,034	1900	A1
R290	0	20	A3
R134a	0	1600	A1
R152a	0	190	A2

Table 2.1: A table comparing different types of refrigerants. Toxicity increases from A to B. Flammability increases from 1 to 3 [20].

2.2 Heat transfer

Heat transfer ($\dot{Q}$) is defined as a transfer of energy where the temperature is the driving force in the system. If temperature is not the driving force, the energy transfer is defined as work ($\dot{W}$) [12]. The first law of thermodynamics is:

$$\Delta U = \dot{Q} - \dot{W} \tag{2.1}$$

Where U is the internal energy, $\dot{Q}$ is the heat transfer rate or heat per unit time, and $\dot{W}$ is the power or work per unit time [12]. If there is no work done on the system ($\dot{W} = 0$) , the first law of thermodynamics can be rewritten as:

$$E_{in} - E_{out} = \Delta E \tag{2.2}$$

Expression (1-10) [13]. States that the total energy into a system minus the total energy out of a system equals the change in total energy of the system. Where:

$$E = \int_0^t \dot{Q} dt \tag{2.3}$$

The international unit for energy is Joule (J). For simplicity I use the rates of thermal energy given in Watts and the total energy consumed given as Wh. Where $1W = 1J/s$ so $1Wh = 3600J$, and $1kWh = 3,6 \cdot 10^6 J$ as reference value.

The total thermal energy rate transferred could in general under steady operating conditions be expressed as:

$$\dot{Q} = \dot{m} c_p \Delta T \tag{2.4}$$

Equation 2.4 is expression (1-18) in [13], where $\dot{m}$ is the flow rate, c_p is specific heat capacity with constant pressure, and ΔT is the temperature gradient inside the pipelines. There are three possibilities for thermal energy transfer. These are conduction, convection and radiation. Conduction is transfer of thermal energy by the movement of particle between to mediums such as solids, liquids or gases. Where one of these mediums holds a higher movement of particles then the adjacent ones. The rate of conduction is given as:

$$\dot{Q}_{cond} = \kappa A \frac{dT}{dx} \tag{2.5}$$

Fourier's law of conduction equation 2.5 is expression (1-22) [13]. κ is the thermal conductivity, A is the area, and $\frac{dT}{dx}$ is the derivative of temperature with respect to distance. Convection: Quote" Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion. The faster the fluid motion, the grater the convection heat transfer. "Unquote [13] p.25. The rate of convection heat transfer is given as:

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) \tag{2.6}$$

Equation 2.6 is expression (1-24) [13] Newton's law of cooling. h is the heat transfer coefficient, A_s is the area of the surface, T_s is the temperature of the surface and T_∞ is the temperature far away from the surface. Radiation is energy transferred by waves (e.g. x-ray waves), in the same way as a body is emitting heat to the surroundings and receiving thermal energy from the surroundings [13]. The rate of thermal radiation is given as:

$$\dot{Q}_{rad} = \epsilon\sigma A_s(T_s^4 - T_{surr}^4) \tag{2.7}$$

Equation 2.7 is expression (1-28) [13]. ϵ is the emissivity of a body, σ is the Stefan-Boltzmann constant, A_s is the area of the surface, T_s is the temperature of the surface and T_{surr} is the temperature of the surroundings. In practice the Newton's law of cooling with a combined heat transfer coefficient is used to estimate both the rate of convection and the rate of radiation in the same expression. This expression is:

$$\dot{Q}_{combined} = h_{combined}A_s(T_s - T_\infty) \tag{2.8}$$

Equation 2.8 is expression (1-29) [13]. $h_{combined}$ is here the combined heat transfer coefficient for radiation and convection. Here $h_{combined}$ is given as:

$$h_{combined} = h_{conv} + h_{rad} \tag{2.9}$$

Equation 2.9 is expression (1- 29) in [12]. And h_{rad} is expressed as:

$$h_{rad} = \epsilon\sigma(T_s + T_{surr})(T_s^2 + T_{surr}^2) \tag{2.10}$$

Where T_s is the surface temperature, and T_{surr} is the surrounding temperature. Equation 2.10 is also from expression (1-29) in [12]. The Nusselt number (Nu) which is the dimensionless convection heat transfer coefficient [12] is expressed as:

$$Nu = \frac{hL_c}{\kappa} \tag{2.11}$$

Where h is the convection heat transfer coefficient, L_c is the characteristic length, and κ is the thermal conductivity. Equation 2.11 is expression (6-5) from [12]. The characteristic length is defined as:

$$L_c = \frac{A_s}{p} \tag{2.12}$$

Here A_s is the area of a surface and p is the perimeter for the surface.Expression 2.12 is expression (9-29) in [12]. In natural convection over flat horizontal surfaces the Nusselt number is given as:

$$Nu = C(Gr_L Pr)^n = CRa_L^n \tag{2.13}$$

Equation 2.13 is expression (9-16) in [13]. Here the Nusselt number is given by the Rayleigh number (Ra_L), which is the product of the Grashof number (Gr_L) and the Prandtl number (Pr). The constants C and ^{n} are both determined by the geometry of the object for which the Nusselt number is calculated for, table 9-1 page 528 in [12]. The Grashof and Prandtl numbers are defined as: Quote"The Grashof number describes the relationship between buoyancy and viscosity within the fluid."Unquote [12] p.527. Quote" The Prandtl number describees the relationship between momentum diffusivity and thermal diffusivity" Unquote [12] p.527. And expressions:

$$Gr_L = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2} \tag{2.14}$$

Equation 2.14 is expression (9-15) from [12] where g is the gravitational acceleration, β is the coefficient of volume expansion ($1/T_{idealgas}$), and v is the kinematic viscosity of the fluid. The Prandtl number is given by:

$$Pr = \frac{\mu C_p}{\kappa} \tag{2.15}$$

Equation 2.15 is expression (6-12) in [12], and μ is the dynamic viscosity.

The one-dimension thermal conduction equation: This equation is based on equation 2.5.

$$\dot{Q}_x - \dot{Q}_{x+\Delta x} + \dot{E}_{gen} = \frac{\Delta E_{element}}{\Delta t} \tag{2.16}$$

$$\Delta E_{element} = E_{t+\Delta t} - E_t = \rho c A \Delta x (T_{t+\Delta t} - T_t) \tag{2.17}$$

$$\dot{E}_{gen} = \dot{e}_{generated} A \Delta x \tag{2.18}$$

Substituting (2.17) and (2.18) into (2.16) and dividing by $A\Delta x$ gives:

$$-\frac{1}{A}\frac{\dot{Q}_{x+\Delta x}-\dot{Q}_x}{\Delta x}+\dot{e}_{generated}=\rho c\frac{T_{t+\Delta t}-T_t}{\Delta t} \tag{2.19}$$

Taking the limits as $\Delta x \longrightarrow 0$ and $\Delta t \longrightarrow 0$ and utilizing the definition of the derivative of Fouriers law of heat conduction:

$$\lim_{\Delta x\to 0}\frac{\dot{Q}_{x+\Delta x}-\dot{Q}_x}{\Delta x}=\frac{\partial \dot{Q}}{\partial x}=\frac{\partial}{\partial x}(-\kappa A\frac{\partial T}{\partial x}) \tag{2.20}$$

I then have:

$$\frac{1}{A}\frac{\partial}{\partial x}(\kappa A\frac{\partial T}{\partial x})+\dot{e}_{generated}=\rho c\frac{\partial T}{\partial t} \tag{2.21}$$

If A is constant I get:

$$\frac{\partial}{\partial x}(\kappa\frac{\partial T}{\partial x})+\dot{e}_{generated}=\rho c\frac{\partial T}{\partial t} \tag{2.22}$$

These expressions and equations are from expression (2-6) to expression (2-13) in[13]. The multidimentional thermal conduction equation is:

$$\frac{\partial}{\partial x}(\kappa\frac{\partial T}{\partial x})+\frac{\partial}{\partial y}(\kappa\frac{\partial T}{\partial y})+\frac{\partial}{\partial z}(\kappa\frac{\partial T}{\partial z})+\dot{e}_{generated}=\rho c\frac{\partial T}{\partial t} \tag{2.23}$$

In cylindrical coordinates:

$$\frac{1}{r}\frac{\partial}{\partial r}(\kappa r\frac{\partial T}{\partial r})+\dot{e}_{generated}=\rho c\frac{\partial T}{\partial t} \tag{2.24}$$

2.3 Heat exchangers

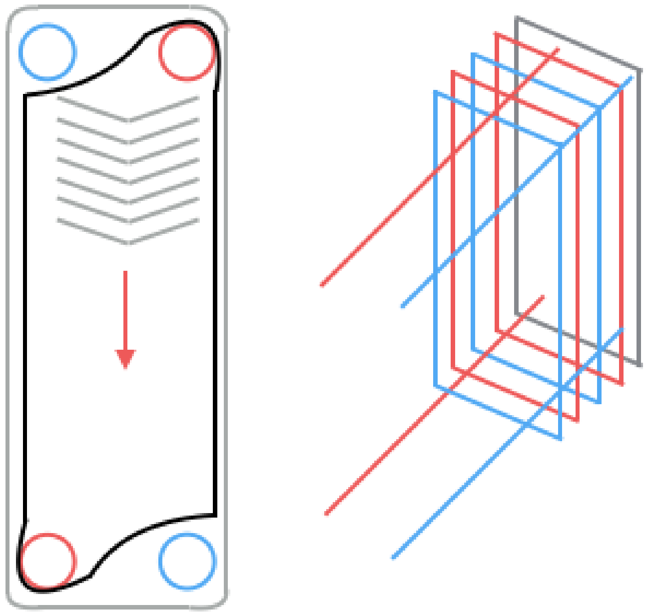

Figure 2.1: Sketch illustrating the basic composition of a plate heat exchanger.

The heat exchangers utilized at Elkem Salten are plate exchangers. This type of heat exchangers are built with a frame holding the large amount of plates in place. Each plate have four larger holes in them. Two on the upper part of the plate and two on the lower part. These holes are to create a circulation of refrigeration medium or heating medium over each side of the plates. A leak-proofing material is placed over two of these holes diagonally one on the upper part and one on the lower part. In the assembly of plates, each plate is placed together in pairs of two, in order to create a space between them for the medium to flow, ensuring that there is no space where the two mediums are meeting. This type of heat exchangers utilizes the small spacing to get the mediums in close contact with each other. The size of the exchanger dictates the size of the area where the two mediums are meeting in cross current. By having for example one inlet of absorption medium (the medium which is to absorb heat) on the upper part of the exchanger. Then the diagonal leak-proofing material ensures that the output for the absorption medium is in the lower part on the last plate in the exchanger. The warmer medium which is to transfer heat, has the opposite connections [16].

There are two main types of heat exchangers, these types are separated on which way the two currents are traveling. The first type is co-current, where the warm and cold water inside the heat exchanger are traveling in the same direction. The other type is counter current, where the two flows are traveling against each other. For the co-current option the warmest medium and the warm side of the cold medium are at one end. With the coldest medium, and the cold side of the warm medium at the other end. The counter current option has the coldest medium in contact with the warmest medium, making the warm side of the cold medium in contact with the cold side of the warm medium [13].

There are also several different designs of heat exchangers such as: plate exchangers, plate fin exchangers, crossflow tubular exchangers, concentric tube exchangers, shell exchangers and tube exchangers. The shell and tube exchangers utilizes baffles to direct the flow of cooling water inside the exchanger. Plate- fin exchangers are plates with mounted fins on the side. Water is being directed through the fins. Each plate is mounted in 90 degree angle to each other making the flow directions perpendicular to each other. The crossflow tubular exchanger uses small tubes to guide the warm water through the exchanger, and is simultaneously sending the cold water over the tubes inside the heat exchanger [13].

The heat transfer rate of a heat exchanger is given by:

$$\dot{Q} = U \cdot A(\Delta T_{lm}) \tag{2.25}$$

Equation 2.25 is expression (11-15) in [13]. Here $\dot{Q}$ is the power of the exchanger, U is the heat transfer coefficient, and ΔT_{lm} is the logarithmic temperature difference between the warm side and the colder side:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} \tag{2.26}$$

A is the heat transfer area. [13] U denotes the overall heat transfer coefficient:

$$U = \frac{1}{A_1 R} + \frac{1}{A_2 R} \tag{2.27}$$

Where:

$$R = \frac{1}{h_1 A_1} + \frac{R_{f,1}}{A_1} + R_{wall} + \frac{R_{f,2}}{A_2} + \frac{1}{h_2 A_2} \tag{2.28}$$

Where R_{wall} is the thermal resistance of the wall between the two mediums, and R_f is the fouling factor due to sediments or biological elements such as algae in the heat exchangers, and R is the thermal resistance.

2.4 Pipelines

In pipelines it is experienced two types of heat transfer. The convection heat transfer the water, and conduction heat transfer to the pipe and the soil surrounding the pipe. The loss rate of thermal energy in the pipe given as:

$$\dot{Q}_{pipe} = \dot{Q}_{cond} + \dot{Q}_{combined} \tag{2.29}$$

Where $\dot{Q}_{pipe} =$ 2.4, $\dot{Q}_{cond} =$ 2.24 and $\dot{Q}_{combined} =$ 2.8. A reducing factor for the thermal loss in the pipe is insulation. If the pipe is insulated this will primarily reduce the loss through conduction, since the pipe is covered in a low density material which holds air between the pipe an the surroundings.

2.5 Water based surface heating systems

Hot water surface heating is a system of one pipeline, or a series of smaller pipelines used for surface heating, such as floors, walls and ceilings. These applications are most commonly found in domestic heating. For floor heating the pipelines are built into the flooring and is either casted in concrete flooring or built in wooden flooring. By continuously circulating hot water into a building in a loop the thermal energy transfers from the pipeline to the surroundings by conduction and convection following newton's law of cooling 2.6. By varying the distance between the loops of pipeline the rate of thermal energy transferred varies. Bathrooms needs a higher temperature than living rooms, and therefore the density of pipeline is increased in the bathroom versus the living room. Normally the pipes carrying the warmest water are placed closest to the outer walls of the building. This is to increase the thermal transfer in these areas so that the areas where the heat loss from the building is greatest are heated more than the other areas, and therefore increases the comfort of the building. This system is quite common with geothermal heat pump systems, and in boiler heating systems. Inside the building heat is transferred to the indoors climate by conduction and convection. Benefits of this system is that it gives a very comfortable heating system with a very good temperature gradient from floor to ceiling. The system distributes heat throughout the entire surface, and is considered a better source for domestic heat than wall or ceiling heat. Where especially the ceiling heat needs a higher temperature which might be considered to reduce the comfort experienced with the system. All systems (floor, wall and ceiling) are built by the same principles, and the same kind of pipelines where diffusion tight plastic pipes are the most common material for pipelines. There are two ways of controlling these systems. The first is to control the amount of fluid into each zone of the building. A thermostat is located in each zone, which controls a valve which then is shut or open depending on the desired temperature for the zone. The other way of controlling this system is to regulate the temperature of the fluid by mixing with colder fluid until the preferred temperature is found [29].

From Prof. Faxen (page 31-32) in [29]. The formula for thermal transfer from hot water pipes casted in a concrete floor, to the floor surface is:

$$\dot{q}_{up} = \frac{\Delta T}{\frac{1{,}15 \cdot a \cdot (R_{up} + R_{down}) \cdot \log \frac{a}{d \cdot \pi}}{\pi \cdot \kappa_{concrete} \cdot R_{down}} + R_{up}} \tag{2.30}$$

For the upwards direction, and:

$$\dot{q}_{down} = \frac{\Delta T}{\frac{1{,}15 \cdot a \cdot (R_{up} + R_{down}) \cdot \log \frac{a}{d \cdot \pi}}{\pi \cdot \kappa_{concrete} \cdot R_{up}} + R_{down}} \tag{2.31}$$

For the downwards direction. Here $\dot{q}_{up}$and $\dot{q}_{down}$ are given in $\frac{W}{m^2}$, $R_{up}$and R_{down} is the overall heat transfer coefficient in upwards and downwards direction from the pipe, a is the distance between the pipes, and d is the diameter of the pipes.

The overall heat transfer coefficient for the floor in the upwards direction is given as:

$$R_{up} = \frac{1}{h_{combined}} + \Sigma_{i=1}^{n} \frac{s_i}{\kappa_i} \tag{2.32}$$

For the downwards direction from the pipes:

$$R_{down} = \frac{1}{h_{combined}} + \Sigma_{i=1}^{n} \frac{s_i}{\kappa_i} \tag{2.33}$$

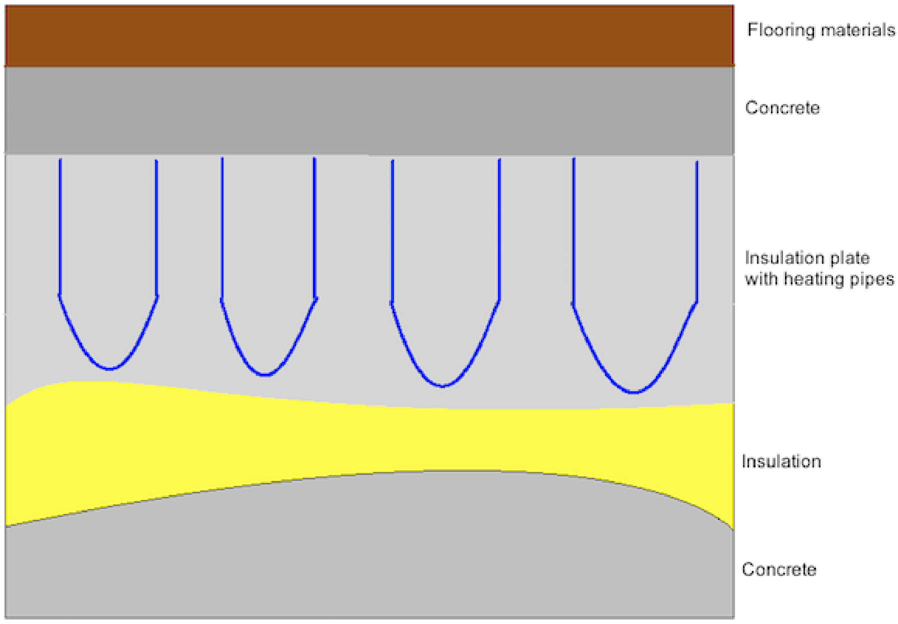

Figure 2.2: Sketch illustrating the components of the water based heating system

2.6 Spot heating sources

This section will describe the most common sources for spot heating and the theory associated with these. The term spot heating is meant for heat sources which are not heating a surface of the building itself, but rather smaller sections of the building [29].

2.6.1 Radiators

A radiator is a spot heat source which transfers thermal energy by convection and radiation. The ratio of energy transfer by radiation is 30-60% of the total energy transferred. The radiator is normally built by two metal plates that are welded together. The panel radiator is two larger plates; one as the front and the other is the back. A radiator of this design is called a panel radiator. Another design of radiators is where several of these panels are set together in a frame. The radiators heat transfer is controlled by a valve adjusting the amount of hot water into the radiator. Another possibility is to equip the radiator with a thermostat measuring the air temperature and then adjusting the amount of hot water flow by the set temperature. From equation 2.6 the relation between the temperature difference and the flow rate is given. Therefore by comparing two radiators with individual temperature differences the flow rate and hence the volume of the radiators must not necessarily be the same for both[29].

2.6.2 Convectors

Quote"Convectors are built by round or squared finned pipes or copper pipes with aluminum wafers placed in a enclosure open in the top and bottom." Unquote Zijdemans [29] (page 51). The heat is transferred by air which are heated and then rises from the top of the convector. In order to increase the heat transfer fans could be added to the convector, the fans would make the convection inside the convector a forced convection, in contrast to natural convection when the convector is without fans [29].

2.6.3 Radiating heat strips

A radiating heat strips is a wafered pipe without insulation. The wafers contributes to increase the transferred heat from the cornice to the surroundings by increasing the surface from where the energy is transferred. Radiating heat strips come in two different models, where one is the single radiating heat strips, and the other is the double radiating heat strips. The two designs differ in that single radiating heat strips has one pipe with the fluid running through it, while the double radiating heat strips has two pipes and the entry and exit of fluid is on the same side. Radiating heat strips are delivered in standard lengths of 2,5m and can be chopped to desired length, by cutting the radiating heat strips the area of thermal transfer is naturally decreased. A great disadvantage with radiating heat strips is that they are very sensitive to the hot water temperature, and only utilizes natural convection for thermal transfer [29].

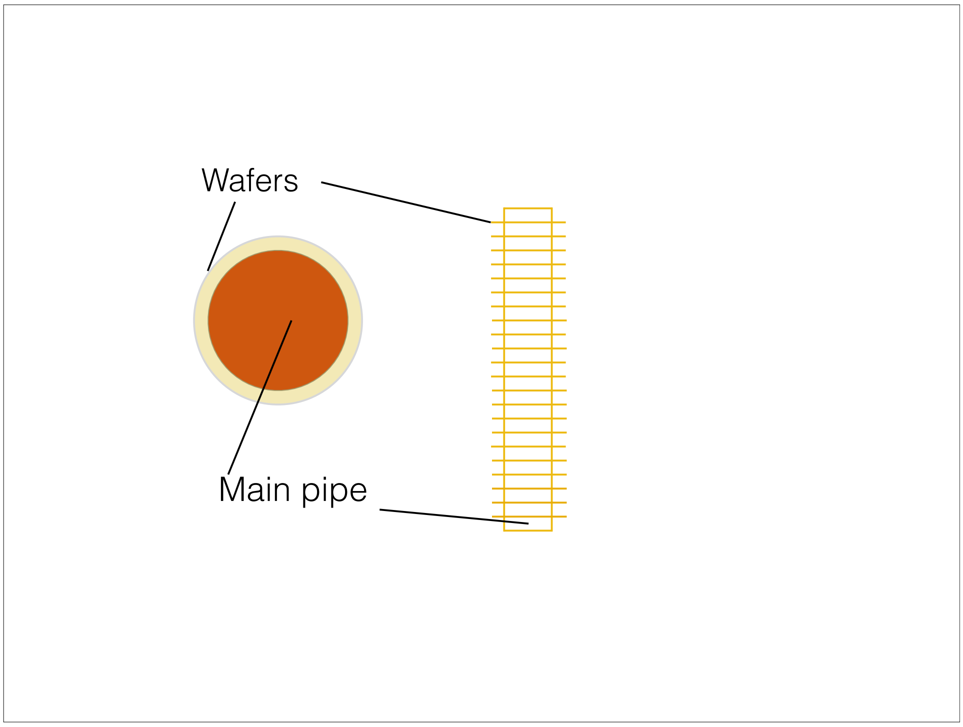

Figure 2.3: A principle sketch of a heat strip

2.7 Ventilation heating

Ventilation heating is commonly done with a heating section in the ventilation system.. The heating section is placed after the inflow fan, and is heating the cold air to the set temperature desired for the air into the building. The temperature is adjusted by altering the airflow into the building. The heating section uses electrical energy to transfer thermal energy to the air, although other energy sources are applicable. Newer ventilation systems are usually equipped with a heat recoverer and or in addition to a heat battery in order to transfer thermal energy from the outflow air to the inflow air. The heat battery functions in principle as a heat exchanger between liquid and water. In this application the heat battery is a replacement or as an addition for the electrical heater inside the ventilation system. The heat battery utilizes forced convection to transfer heat. This is normally used together with a heat recoverer for the ventilation which in turn takes heat from the warm air going out of the building and transfers the heat to the ingoing air in the ventilation system [29].

$$\dot{Q} = \dot{v} \cdot C_p \cdot \rho \cdot (T_i - T_{DUT}) \tag{2.34}$$

This is the rate of power power for the ventilation system. This is equation 1.13 from [29]. Here $\dot{v}$ is the volume of air in $\frac{m^3}{h}$.

$$\dot{Q} = \dot{v} \cdot C_p \cdot \rho \cdot (T_i - T_{DUT}) \cdot (1 - \eta) \tag{2.35}$$

Is equation 1.14 from [29]. This equation takes into consideration the loss of efficiency of the ventilation-systems efficiency, here given as η.

2.8 Heat pumps

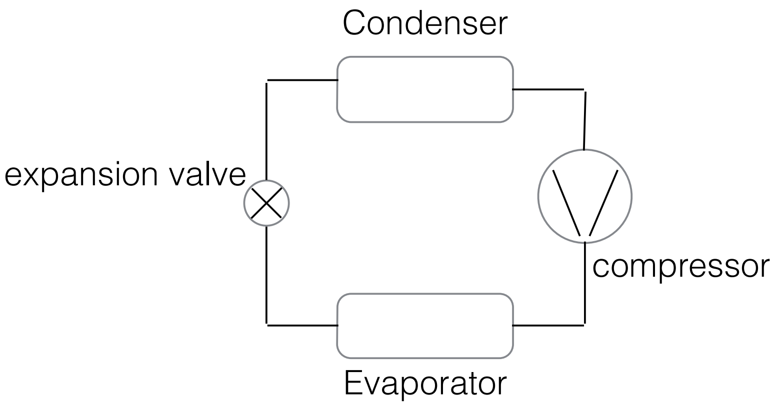

Figure 2.4: Sketch illustrating the components of a heat pump.

A heat pump is a device that is capable of transferring thermal energy against the natural "direction" of energy flow. An object or enclosed environment which at one point is heated to hold a higher temperature than the surroundings will be cooled. If not, thermal energy is continuously added to the object or enclosed environment. Heat pumps counteracts the cooling effects by gathering thermal energy from the surroundings and transferring it to the object or enclosed environment. This process requires additional energy in the form of work, as stated in the first law of thermodynamics. 2.1 A heat pump consists of: evaporator, compressor, expansion valve, and a condenser. The condenser and evaporator are heat exchangers that transfers heat from the working medium used in the heat pump to the surroundings, and from the surroundings to the working medium. The principle cycle of a single stage heat pump: 1. The evaporator gains heat from the surroundings and the working fluid evaporates. The vapor is then leaded to the compressor 2. The compressor compresses the vapor of the working fluid. The now high pressure vapor is led by pipeline to the condenser. 3. In the condenser the high pressure vapor transfers the thermal energy to the surroundings and the working fluid condenses to liquid again. The working fluid is now led to the expansion valve. 4. The expansion valve releases working fluid from the high pressure side with the condenser to the low pressure side with the evaporator. This is done with a valve that regulates the pressure of the high pressure side by releasing liquid to the low pressure side as soon as the compressor creates an overpressure on the high pressure side. When the pressure on the high pressure side exceed the desired set pressure created by the compressor, the expansion valve opens and releases a small stream of working fluid to the low pressure side. The high pressure side now has a lower pressure than the set pressure and the valve closes again. The working fluid released has now entered the evaporator again and the cycle is complete[29]. Heat pumps are very versatile in areas for application, such as air to air heat pumps where the temperature of air is increased. Fluid to fluid where thermal energy is transferred between two different fluids with different temperatures. Fluid to air is a combination of air to water where thermal energy is transferred between different mediums [29].

The power of an evaporator/ condenser is given as:

$$\dot{Q} = \dot{m}_{fluid} \Delta H \tag{2.36}$$

Here $\dot{m}_{fluid}$ is the mass flow rate of the working fluid, and:

$$\Delta H = c_p \Delta T \tag{2.37}$$

Equations 2.36, 2.37 are an alternative presentation of equation 2.4. Equation 2.36 is also found in [29] as formula 3.5. ΔH is here the enthalpy difference over a component of the heat pump. For the condenser, evaporator and compressor equation 2.36 is more specific:

$$\dot{Q}_{cond} = \dot{m}(h_2 - h_3) \tag{2.38}$$

$$\dot{Q}_{evap} = \dot{m}(h_1 - h_4) \tag{2.39}$$

$$\dot{W}_{teo} = \dot{m}(h_2 - h_1) \tag{2.40}$$

Where $\dot{W}_{teo}$ is the theoretical work done by the compressor. Equations 2.38 to 2.40 are found in [10]. Where h_1, h_2, and h_4 are numerical enthalpy values found by interpolating from a logarithmic pressure enthalpy diagram. A logarithmic pressure enthalpy diagram is a diagram which displays all the different phases of a working fluid. The phases are indicated with the boiling line and the dew line. Starting from the left of figure 2.5 the liquid phase is from zero enthalpy in the diagram and follows the boiling point line up to the critical point for the medium. The critical point indicates the point at which enthalpy and pressure the medium will change phases from liquid to gas directly. On the right side of the critical point following the curve which indicates the pressure and enthalpy of the condensation point also known as the dew point line is the gaseous phase for the medium. Between the boiling point curve, the dew point curve, and below the critical point is the mixed zone where the medium is in mixed phases of gas and liquid. The pressure on the y- axis is presented as logarithmic pressure and is in absolute pressure (bara). The enthalpy is on the x- axis and is the heat content of the medium typically given for the medium at liquid state at zero degrees celsius. It is therefore only enthalpy differences that is used, when the diagram is utilized for interpolating numerical values for heating to cooling processes. In these processes the temperature which is constant with constant pressure in the two phase area is the set point. To the right side of the condensation line some linear lines which presents constant entropy are located [29]. An example of a heating process is shown in figure 2.5.

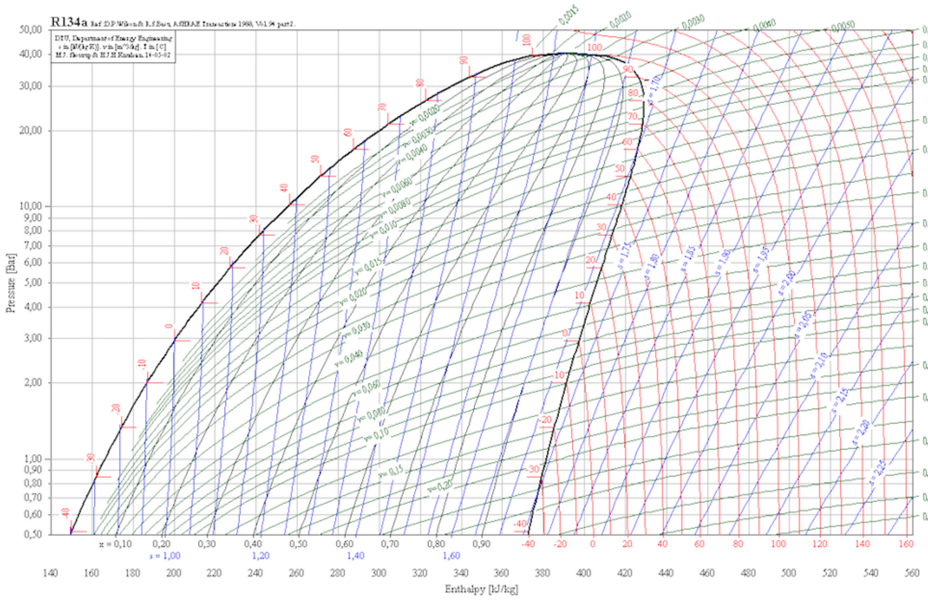

Figure 2.5: A screen dump from cool pack with the logarithmic pressure enthalpy diagram for the medium R-134a [22].

In figure 2.6 a heating cycle with the names of each stage in the cycle

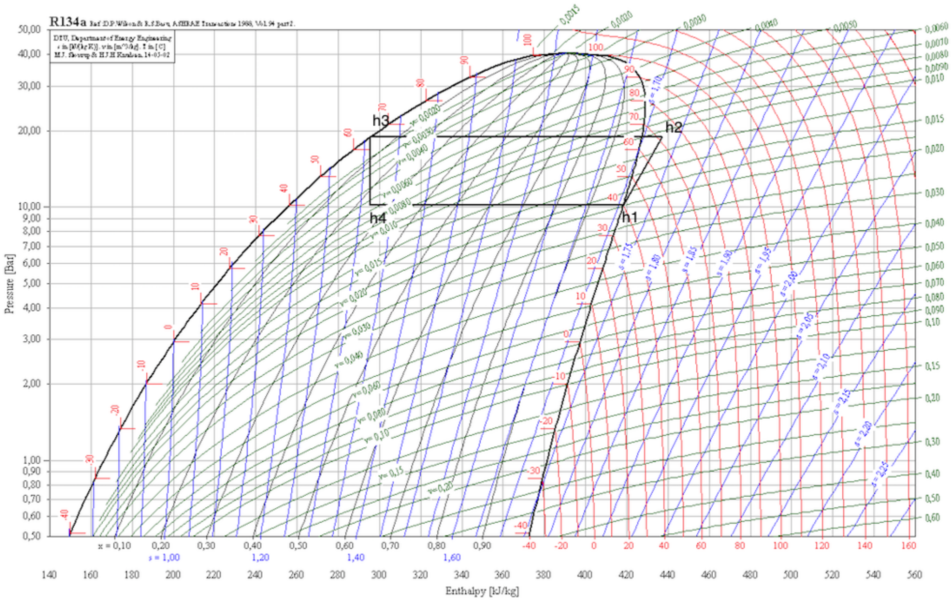

Figure 2.6: A screen dump from cool pack with a heating cycle in the logarithmic pressure enthalpy diagram for the medium R-134a [22].

is marked from h_1 to h_4. This heating cycle has ideally an isentropic compression, meaning that the line between h_1 and h_2 is a straight line with the same slope as the entropy line. The point h_3 is saturated liquid, and h_1 is the saturated vapor point [10] and [29].

The Coefficient Of Performance (COP) is given as:

$$\text{COP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} = \frac{\Delta H_{cond}}{\Delta H_{comp}} \tag{2.41}$$

Equation 2.41 The first part of the equation is found in [10] page 4, the second part of the equation is formula 3.7 in [29]. ΔH_{cond} is the enthalpy difference over the condenser, and ΔH_{comp} Is the enthalpy difference over the compressor. This is true due to the mass flow rate, which must be the same for each component of the system.

The minimum work for a heat pump is given as:

$$\dot{W}_{min} = \frac{\dot{Q}_{cond}(T_{min,cond} - T_{max,evap})}{T_{min,cond}} \tag{2.42}$$

Equation 2.42 is from [10] page 4, the original expression does not account for a temperature difference inside the condenser and evaporator. The actual work of a compressor is:

$$\dot{W}_{comp} = \frac{\dot{W}_{teo}}{\eta_{is}} \tag{2.43}$$

η_{is} is the isentropic efficiency of the compressor. Equation 2.43 is from [10].

2.9 Wastewater recycling

Wastewater is a generic term for liquid waste from industrial and domestic installations, such as cooling water, domestic wastewater which includes sewage, and sewage alone. Wastewater heat recycling is to recycle the thermal energy in wastewater. This form for thermal recycling for residential application is not very common in Norway because there is a requirement for separation of cold wastewater from the wastewater. Also, this type of recycling is not considered to be very cost-effective for older houses. But when compared to the future planned building standard for passive houses this technique is considered to be more cost-effective due to the increased need for hot water for the buildings [29].

Wastewater recycling is divided in two categories, which are active and passive wastewater recycling. Active wastewater recycling is distinguishable with the need for additional energy for the recycling process, while passive wastewater heat recycling does not require additional energy for the process, but usually involves a accumulator. Although they both might need circulation pumps to run the process the active wastewater recycling usually involves a heat pump [29].

The recycling process itself is divided into two methods, these are delayed thermal recycling and instant thermal recycling. Delayed thermal recycling is a recycling method where the thermal recycling process is delayed from the consumption of hot water, meaning that the recycling does not necessarily appear simultaneous with the hot water consumption. In order to recycle the thermal energy with this method require a accumulation reservoir. The accumulation reservoir has as primary function to preheat the cold water going into the hot water cistern [29].

Chapter 3

Plant layout

All Elkem Saltens furnaces are located in the same building and are sources for hot cooling water. The cooling system at Elkem Saltens furnaces is a one phase system (only liquid), which operates with freshwater as the cooling agent. The system is designed for temperatures up to $35°C$ in to the manifold and $45°C$ out of the sink [16].

Each furnace has a separate cooling system consisting of pumps and pipelines carrying cold water from the basement of the main factory building to a manifold located at the same height as the furnace itself. The manifold then distributes water out to the different water-cooled equipment. Because of the extreme heat from the process, cooling all exposed components are needed to avoid destruction of the equipment. Such equipment would typically be raw material feeders, heat shields and the furnace roof. Each component has its own separate loop of cooling water from the manifold. On return from the furnace, the hot cooling water is gathered in sinks. From the Manifold to the sink there are approximately 5 km of pipelines. From the sink the water goes back to the basement of the main factory building and are then being pumped to the heat exchangers. In the heat exchangers the hot fresh water is exchanged against cold seawater and the cooled fresh water is pumped back into the loop again.

3.1 Inside the main factory building

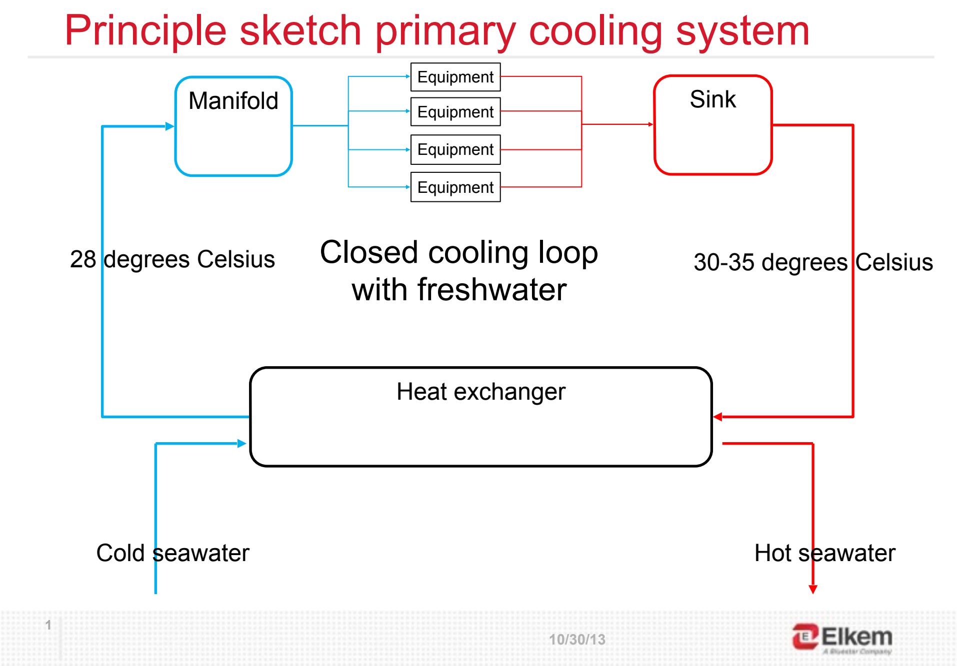

Figure 3.1: Shows a principle sketch of the cooling system [16].

In order to transport the cooling water around in the system large pumps are installed. There are four main pumps on each furnace, two of them always running while two are backup. These pumps have a capacity of $567m^3$/h. In addition there are two separate pumps, called booster pumps, installed. These pumps are needed to gain enough water pressure in the highest located cooling circuits. The capacity of the booster pumps are $236m^3$/h, one always running and one being backup. If needed there is also an opportunity to add cold fresh water to the system. This cold freshwater comes from a nearby freshwater lake and always holds a pressure of 10 bars. This high pressure is due the location of the fresh water pumping station [16].

For all furnaces the basic cooling system is the same with smaller variations in the way the hot water is transported from the sinks to the outside of the main factory building. According to Figure 3.1 the difference between the three furnaces is mainly in the distribution system from the manifolds and back to the sinks again. From the sink the rest of the cooling circuits are basically the same except variation in diameter of the main pipeline from the main factory building to the heat exchanger building.

3.2 Individual differences between the furnaces

3.2.1 Furnace #1

For furnace #1 there is a sink collecting the hot cooling water from the various equipment. This sink is then directly connected to a larger pipeline that transport the cooling water to and from the heat exchangers. This furnace cooling system is different from the others in the way that it has a completely closed cooling system, meaning that there is no air getting into the system. Located in the basement there is a pressure tank installed. This pressure tanks task is to maintain the overpressure in the system so that the circulation of water is maintained and stable for all the equipment.

3.2.2 Furnace #2

Furnace #2 has an open cooling water system, meaning that the hot cooling water from the sink is collected in an open tank in the basement. Sensors automatically monitor the level of water in the tank. From this tank the water is pumped to the heat exchangers and back to the manifolds again.

3.2.3 Furnace #3

Furnace #3 does not have a pressure tank or a gathering tank, but the sinks for this furnace are larger. These sinks are also open. From the sinks the hot water is then sent down to the heat exchanger building.

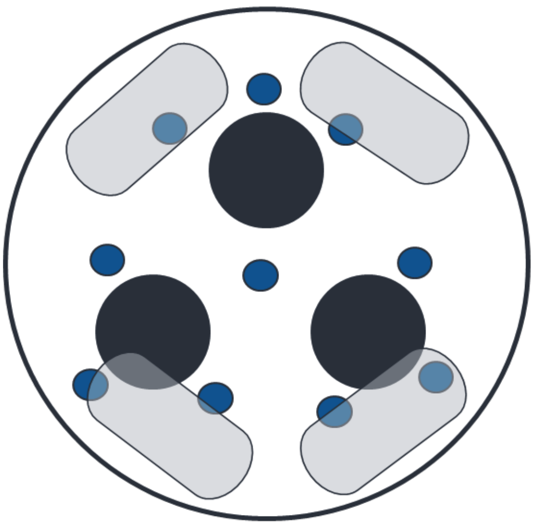

Figure 3.2: A principle sketch from top view of the furnace with some components

In figure 3.2 a sketch illustrates the top or the "hat" of the furnace, which is situated on top of the furnace itself. The black circles indicate the electrodes in the furnace. The small blue filled circles indicate the feeding tubes for feeding raw material to the furnace and the rectangles with their rounded edges indicate the four chimneys for each furnace [16].

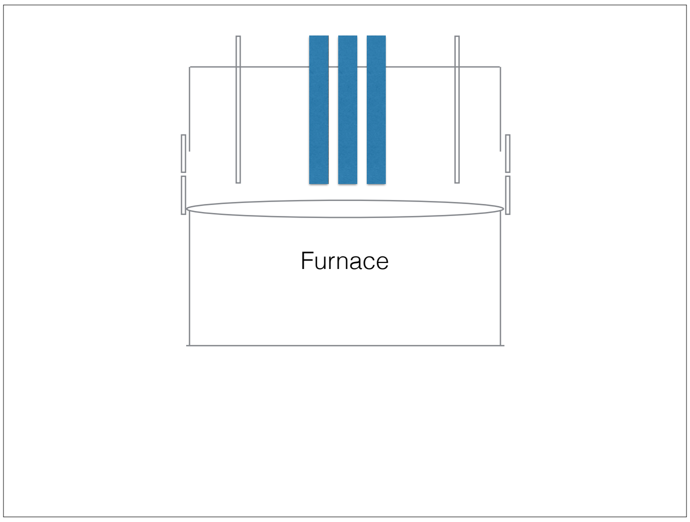

Figure 3.3: Shows a principle sketch side view of the furnace with components

In figure 3.3 The electrodes are indicated by the blue coloring, the feeder tubes for material are the long white colored rectangles, and the smaller white colored rectangles are the heat shields.

3.3 Outside the main factory building

At furnace #3 and #2 the main pipeline is led to a small heat exchanger building located right outside the main factory building. Here the heat exchanger for the artificial grass pitch (Lakselva) is located together with the heat exchangeer for the garden center (Sisoflor). The heat exchanger for the grass pitch is able to receive hot water from both furnace #3 and #2, while the Sisoflor heat exchanger is in addition receiving hot water from the exhaust pipe on furnace #2. This is a separate water heating system which utilizes the heat in the exhaust gases. The hot water heated by the exhaust gases is then led to two tanks located in the basement where it is pumped to the smaller heat exchanger building. This building is marked in figure 3.5 [16].

From the basement of the main factory building 500mm ST38.8.1 (steel) insulated distant heating pipes for furnace #1 ,400mm cast iron pipes for furnace #2 and 500mm cast iron pipes for furnace #3 [16] transports the hot cooling water to the heat exchanger building. This building is located in the eastern part of the plant site, in close proximity to the sea and can be seen in figure 3.5 Inside this building there are in total ten heat exchangers. Seven of these are heat exchanging the hot water with colder sea water, and three are heat exchanging hot water from the factory with cold water from a fish farm (Sisomar). In figure 3.4 we can see the three Sisomar heat exchangers marked as 1,2 and 3. For the seven remaining heat exchangers which are dealing with the main stream of hot cooling water, three of these are connected to furnace #3, two heat exchangers are connected with furnace #2 two and the last two heat exchangers are connected to furnace #1.

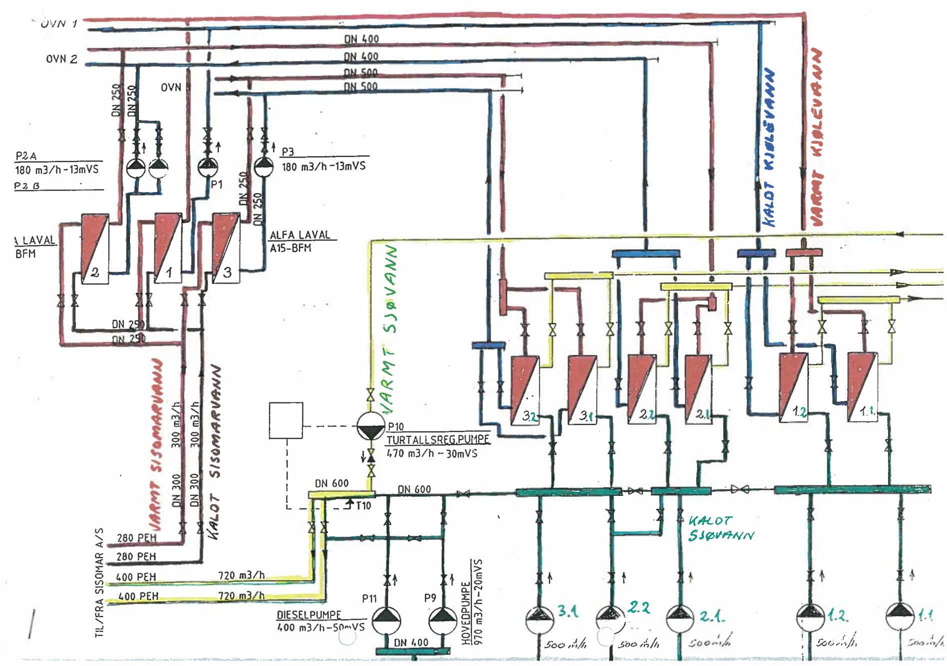

Figure 3.4: Shows a detailed overview of the layout for the heat exchanger building[16].

Figure 3.4 is a picture of the pipeline layout of the heat exchanger building.[16] Figure 3.4 indicates the various pipelines in different coloring: Red indicates the pipes containing hot fresh water, blue indicates pipes containing cold fresh water, the yellow pipes indicate the hot seawater after the heat exchangers, and the green pipes indicate cold seawater.

Figure 3.4 also indicates the various pumps and their capacities. The pumps for the sea water side have capacities of two times $900m^3/h$ for each furnace, but this sea water is divided over the amount of heat exchangers for each furnace. The same is done for the hot fresh water from the furnaces, which is being pumped at two times $567m^3/h$ for each furnace. As an example for furnace three, hot water is being pumped to three heat exchangers at $1134m^3/h$, providing approximately $378m^3/h$ for each heat exchanger. From the cold side of the exchanger $1800m^3/h$ is being pumped. So for each heat exchanger approximately $600m^3/h$ is being pumped through it.

3.3.1 Pipelines

The outer pipelines are running from the main factory building to the heat exchanger building. These pipes are in close proximity to both the mechanical garage and the wardrobe and office building (Miljøbygget). A overview of the plant site and the pipeline layout can be seen in figure 3.5.

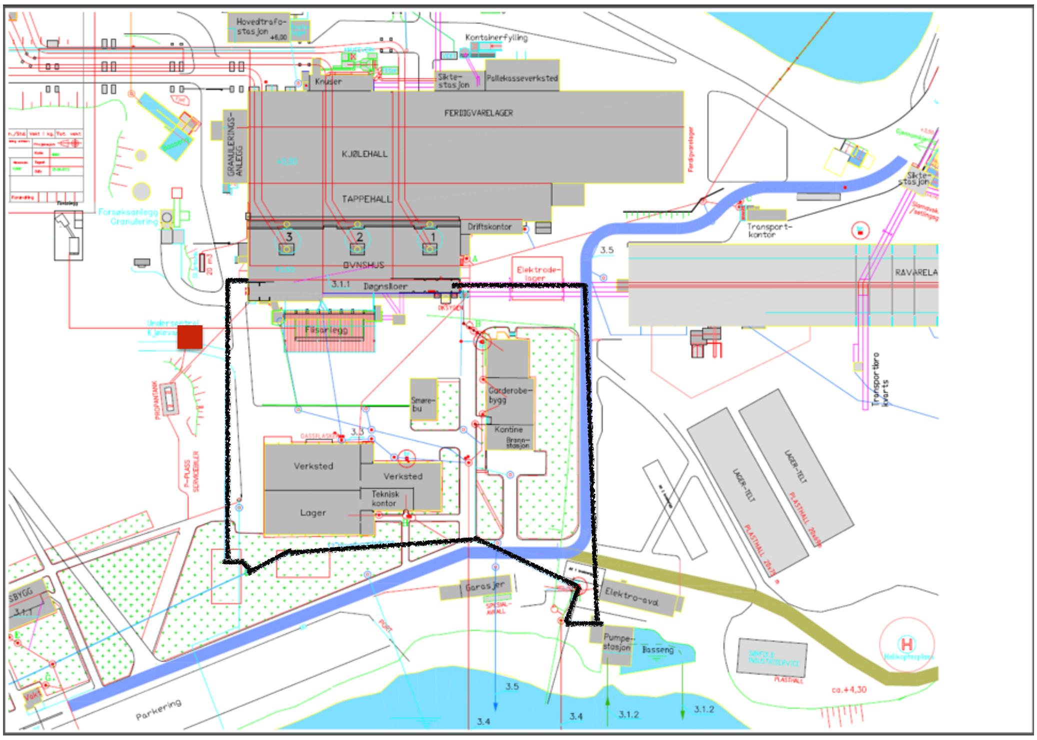

Figure 3.5: A figure of the main water line from the main factory building to the heat exchanger building(marked with the black fuzzy lines). The smaller heat exchanger building marked as the red square [16].

3.4 Heat recycling

3.4.1 Lakselva artifical grass pitch.

The Lakselva artificial grass pitch has pipelines underneath the artificial grass math in order to keep the pitch snow free during winter time. The total amount of thermal power, utilizing the hot cooling water from the plant, for heating the pitch is estimated to be 0,628 MW [19].

3.4.2 Sisoflor

Sisoflor was a company that started in 1993. They produced roses and cucumbers in their garden center located in close proximity to Elkem Salten. Sisoflor also utilized the hot cooling water from Elkem Salten. Here the hot cooling water was utilized for heating water inside their garden center. In October 2011 Sisoflor went bankrupt.[18] The total amount of thermal power utilized at Sisoflor from Elkem Salten is estimated to have been 0,295 MW [19].

3.4.3 Sisomar

Sisomar is a smolt producer established in 1995 as a cooperative venture between Elkem(50%) and various edible fish producers in Nordland county(50%). The business was established in order to utilize hot cooling water from Elkem Salten together with a high demand for smolt production.[25] The total amount of thermal power from Elkem Salten to Sisomar is estimated to be 4,514 MW [19].

3.5 The environment building (Miljøbygget)

The environment building (milkøbygget) is a combined office and wardrobe building consisting of two floors. The basement of the building holds showers, a training room, some conference rooms and the headquarter for the plants fire department. The second floor holds the main lunchroom, some small conference rooms, a large wardrobe and offices. Since this building has the largest wardrobe compartments it also has the largest supply of hot water[16].

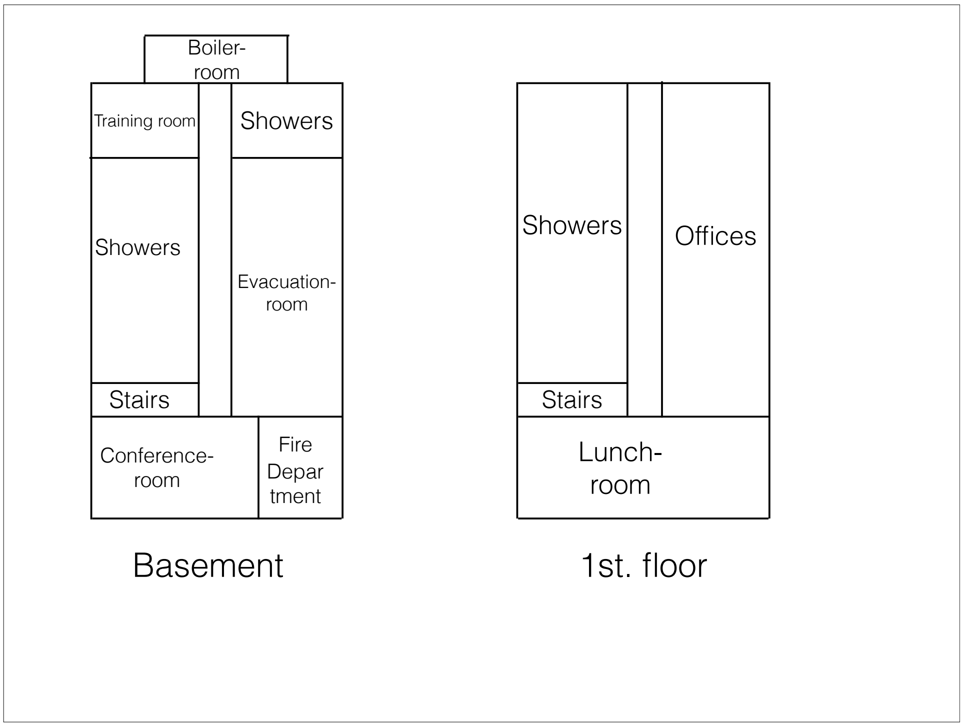

Figure 3.6: A sketch of the floors in Miljøbygget [16].

Heating is sectioned as base load heating, which is the dimensioned heating power, needed for a building for the general heating season of a year. A peak/top load is the extra heating power needed on extreme days. Extreme days are days that are abnormal to the dimensioned outdoor temperature for the area [29].

The environment building has a base load heating system consisting of electrical panel radiators together with a ventilation heating system, which altogether is the main heating system for the building.

The wardrobe compartments holds a higher temperature then the rest of the building. This is to ensure drying of wet work clothing and towels used by the plants workers.

The total electrical consumption for this building is $1437,52 MWh$ per year. This consumption includes the warm water heaters, lighting, computers, printers, other electricity-consuming appliances and the above mentioned radiators and ventilation system.

Chapter 4

Methodology

In this chapter I have crated a plot for the temperature gradient of the pipeline between the furnace housing and the heat exchanger housing. I have then calculated the rate of energy loss for this system, the rate of energy gained and the rate of energy for the heat exchanger. For the temperature and heat flux plots the partial differential equation (PDE) toolbox ®, which is a application in the MATLAB ®software was utilized. The PDE application is a graphical tool used to draw figures and solving the partial differential equation for different environments for these figures. The different environments are in addition to heat transfer: electrostatics, magnetostatics, structural mechanical strain and stress, generic scalar and system, AC power electromagnetics, conductive media DC, and diffusion. These figures are then given numerical values for density, heat capacity, coefficient of heat conduction, heat source, convective heat transfer coefficient and external temperature. In the PDE application one can choose solve for heat transfer, and specify wether one desires to use a parabolic differential equation or an elliptic differential equation to solve. The parabolic equation in the PDE application is:

$$\rho C \frac{\partial T}{\partial t} - \nabla \cdot (\kappa \nabla T) = Q + h(T_{ext} - T) \tag{4.1}$$

Here ρ is the density, C is the heat capacity κ is the coefficient of heat conduction, Q is heat source, h is the convective heat transfer coefficient, T_{ext} is the external temperature, and T is the temperature. Equation 4.1 is the derivative of Fourier's law. For my plots the steady- state version of this equation was chosen. For the steady- state version or the elliptic differential equation the partial derivative of T by lower case t is zero, the same applies for Q, since there is no heat generated. So the final expression is:

$$-\nabla \cdot (\kappa \nabla T) = h(T_{ext} - T) \tag{4.2}$$

Since the elliptic differential equation considers the steady state it simplifies the simulation since for the steady state there is no heat transfer by convection only conduction, due to there is no movement in the medium [13]. This is also the base for the first assumption in the simulation, which is that the materials are homogenous with no change in entropy. So the heat gained distributes evenly throughout the material.

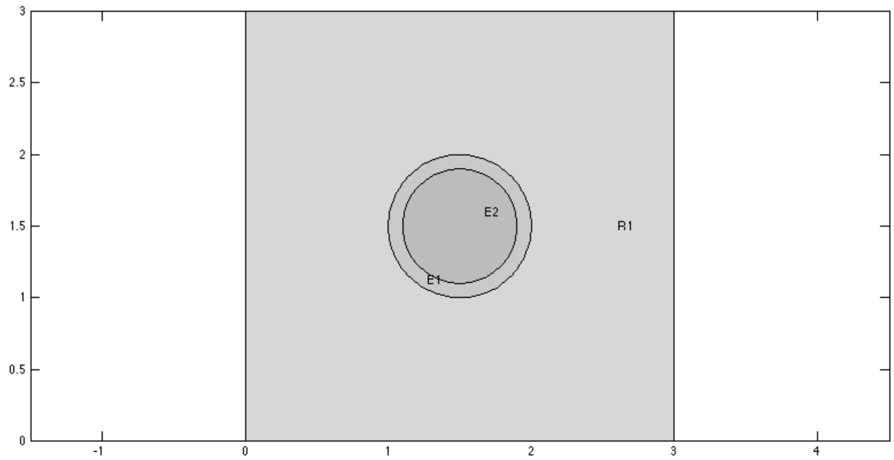

Figure 4.1: The figures used in the PDE application

In figure 4.1 we see the figures I used in the PDE application where E2 is represents the inside of the pipe, and has a size of a 1,4 diameter circle. E1 represents the pipe itself, and has a size of a 1,5 diameter circle. R1 is the surrounding soil and has a size of 3x3 box.

Values	E1	E2	R1
κ	80,2 $\frac{W}{mK}$	0,631 $\frac{W}{mK}$	0,3 $\frac{W}{mK}$
h	0	0,57$\frac{W}{m^2K}$	0,24 $\frac{W}{m^2K}$
T_{ext}	0	313,15 K	283,15 K

Table 4.1: The different values used in the PDE plot.

The numerical values presented in table 4 are gathered from various tables. The heat transfer coefficients (h) are from [1]. The thermal conductivities for: E1 the numerical value is from Table A-3 [12]. E2, the numerical value is from Table A-9 [12]. R1 the numerical value is a combination of the thermal conductivity for soil, with organics content and soil, saturated, from [3].

The second assumption I have done is that the soil is in uniform contact with the pipeline. This assumption is necessary for the temperature gradient flux to be evenly distributed in the normal direction to the pipes length.

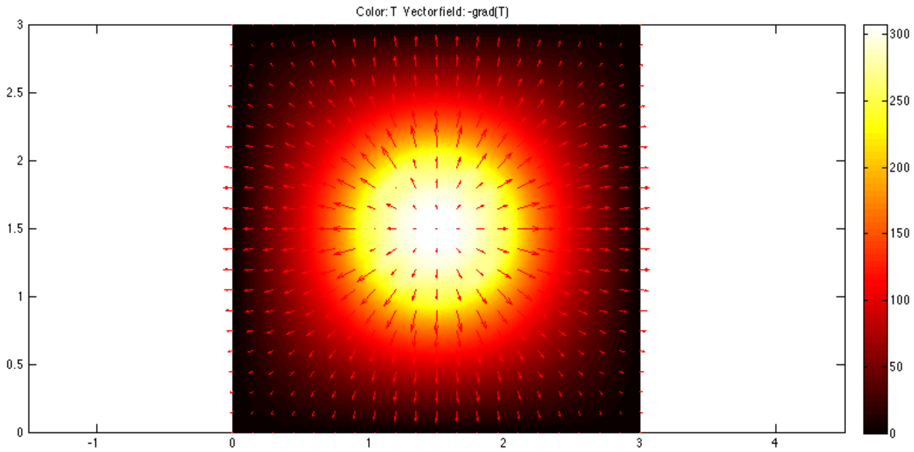

Figure 4.2: The PDE vector field plot with external temperature of zero.

Figure 4.2 shows the temperature gradient normal to the pipes length as the external temperature is zero. As long as the external temperature is equal to zero then there will be no change in the temperature gradient. But as soon as the time is set to a different value than zero the temperature of the soil surrounding the pipe will increase up to a stable temperature. And the temperature gradient out of the pipe will change due to the increased temperature of the soil, following the first law of thermodynamics. This also applies for the pipe itself.

4.1 Calculations of energy balance

For my calculations of the energy balance for pipes and heat exchanger. I have used tables in [13] to find the various numbers for density (ρ), heat capacities ($c_p water$), thermal conductivity (κ), heat transfer coefficient (h) and overall heat transfer coefficients (U).

4.1.1 Assumptions

- A steady- state system and steady operating conditions meaning that the system is not subject to changes over time in this scenario.
- The heat exchanger is well isolated so that the heat loss from this is negligible.
- No heat loss inside the furnace housing and inside the heat exchanger housing. So the temperatures around the gathering tank for cooling water are the same as the cooling water.

- The changes in potential and kinetic energies of the fluid streams are negligible.
- There is no fouling factor for the system.
- The fluid properties are constant.
- Heat transfer for the pipe is one dimensional since there is symmetry from the center.
- Assume same pipe dimensions everywhere.
- Thermal conductivity is constant.
- Pumps are preforming at maximum capacity.
- Temperature inn and out of furnace are $28°C$ inn and $40°C$ out.
- Temperature into heat exchanger (warm side) are $39°C$, and has the same c_p as water at $40°C$.
- Temperature into heat exchanger (cool side) are $5°C$.
- The pipeline's length from the furnace housing to the heat exchanger housing is 200m.
- The temperature gradient is 1 degree loss per 200m of pipe. $\Delta T_{insidethemainpipeline} = 1$.
- The temperature difference between the inner surface of the pipe and the outer surface is $0{,}1°C$.

	Furnace	Heat exchanger	Pipeline
Pump capacities($\frac{m^3}{h}$)	236 (for one small part) + 1135	1800	1135

Table 4.2: Numerical values provided by Elkem Salten.

The pump capacities in the furnace is $236\frac{m^3}{h}$ for two pieces of equipment mounted high up in the chimney, I have therefore neglected the trykkringer and spjeld equipment in my calculations. Giving me a overall pump capacity of $1135\frac{m^3}{h}$ for the whole system.

Since the area of the heat exchanger is unknown in these calculations it is not possible to calculate the numerical value for U, this is said to range between 850 - 1700 ($\frac{W}{m^2 \cdot °C}$) for water to water heat exchanger [13]. And I have set the value of U to be 1700 ($\frac{W}{m^2 \cdot °C}$).

Medium / Material	$\rho(\frac{kg}{m^3})$	$c_p(\frac{KJ}{kg \cdot K})$	$\kappa(\frac{W}{m \cdot K})$	Prandtl number (Pr)
Water liquid ($5°C$)	999,9	4,205	0,571	11,2
Water liquid ($10°C$)	999,7	4,194	0,580	9,45
Water liquid ($15°C$)	999,1	4,185	0,589	8,09
Water liquid ($20°C$)	998,0	4,182	0,598	7,01
Water liquid ($25°C$)	997,0	4,180	0,607	6,14
Water liquid ($30°C$)	996,0	4,178	0,615	5,42
Water liquid ($35°C$)	994,0	4,178	0,623	4,83
Water liquid ($40°C$)	992,1	4,179	0,631	4,32
Iron($T = 300K$)	7870	447	80,2	

Table 4.3: Numerical values from [13] tables for some selected mediums and temperatures.

4.1.2 Calculations

First I find the flow rates for the system:

$$\dot{m} = pumpcapacity \cdot \rho \tag{4.3}$$

$$\dot{m}_{pipe} = \frac{1136\frac{m^3}{h} \cdot 992,1\frac{kg}{m^3}}{3600s} \tag{4.4}$$

$$\dot{m}_{pipe} = 312,81\frac{kg}{s} \tag{4.5}$$

For the heat exchanger I used the same mass flow as for the pipeline since this is the same as the mass flow rate into the heat exchanger on the warm side. Therefore the mass flow out of the heat exchanger on the cold side must be the same as the mass flow in inn order to satisfy assumption #2. The conservation of power relation:

$$\dot{Q}_{in} = \dot{Q}_{out} + \dot{Q}_{loss} \tag{4.6}$$

Where $\dot{Q}_{in}$ is the thermal power gained by the system in the equipment by radiation, convection and conduction. $\dot{Q}_{out}$ is the thermal power which the heat exchanger disposes by conduction, radiation and convection. $\dot{Q}_{loss}$ is the thermal power lost in the pipe by conduction, convection and radiation. Here:

$$\dot{Q}_{loss} = \dot{Q}_{conduction} + \dot{Q}_{convection} + \dot{Q}_{radiation} \tag{4.7}$$

$$\dot{Q}_{conduction} = \frac{1}{r}\frac{\partial}{\partial r}(\kappa r\frac{\partial T}{\partial r}) + \dot{e}_{generated} = \rho c\frac{\partial T}{\partial t} \tag{4.8}$$

$$\dot{Q}_{convection} + \dot{Q}_{radiation} = \dot{Q}_{combined} \tag{4.9}$$

$$\dot{Q}_{combined} = h_{combined}A_s(T_s - T_\infty) \tag{4.10}$$

$\dot{Q}_{in}$ can be expressed as:

$$\dot{Q}_{in} = \dot{m} \cdot c_{p_{Waterliquid(40^\circ C)}} \cdot \Delta T_{intofurncae} \tag{4.11}$$

$$\dot{Q}_{in} = 312,81 \frac{kg}{s} \cdot 4,179 \frac{KJ}{kg \cdot K} \cdot (313,15K - 301,15K) \tag{4.12}$$

$$\dot{Q}_{in} = 15,687 MW \tag{4.13}$$

$\dot{Q}_{loss}$ can also be expressed in a similar manner to $\dot{Q}_{in}$ only difference being the ΔT element. So:

$$\dot{Q}_{loss} = \dot{m} \cdot c_{p_{Waterliquid(40^\circ C)}} \cdot \Delta T_{temperaturegradientinthemainpipeline} \tag{4.14}$$

$$\dot{Q}_{loss} = 312,81 \frac{kg}{s} \cdot 4,179 \frac{KJ}{kg \cdot K} \cdot (1K) \tag{4.15}$$

$$\dot{Q}_{loss} = 1,307 MW \tag{4.16}$$

From the $\dot{Q}_{loss}$ the different types of losses from the system can be calculated.$\dot{Q}_{conduction}$ will here be:

$$\dot{Q}_{conduction} = \frac{\partial}{\partial r}(r \frac{\partial T}{\partial r}) \tag{4.17}$$

$\dot{e}_{generated}$ is zero because there is no heat generated, the $\rho c \frac{\partial T}{\partial t}$ is zero due to the steady state conditions, no change over time, and the heat transfer coefficient κ is constant. Expression 4.17 is the mathematical expression used in the calculation of the rate of conduction:

$$\frac{\partial}{\partial r}(r \frac{\partial T}{\partial r}) = 0 \tag{4.18}$$

With boundary conditions:

$$T(r_1) = T_1 = 40^\circ C \tag{4.19}$$

$$T(r_2) = T_2 = 39,9^\circ C \tag{4.20}$$

These values are assumptions I have made where T_1 is the temperature of the liquid, and T_2 is the temperature outside the iron pipe. Integrating expression 4.18 one time with $\frac{\partial}{\partial r}$ gives:

$$r \frac{\partial T}{\partial r} = C_1 \tag{4.21}$$

Here C_1 is a arbitrary constant. Then by rearranging:

$$\frac{\partial T}{\partial r} = \frac{C_1}{r} \tag{4.22}$$

$$Then \quad \partial T = C_1 \frac{\partial r}{r} \tag{4.23}$$

Integration gives:

$$\int_0^r \partial T = C_1 \int_0^r \frac{\partial r}{r} \tag{4.24}$$

Gives:

$$T(r) = C_1 \ln r + C_2 \tag{4.25}$$

With boundary conditions:

$$T(r_1) = C_1 \ln r_1 + C_2 = T_1 \tag{4.26}$$
$$T(r_2) = C_1 \ln r_2 + C_2 = T_2 \tag{4.27}$$

This is two equations with two unknowns C_1 and C_2, solving for these:

$$C_2 = T_1 - C_1 \ln r_1 \tag{4.28}$$
$$C_1 \ln r_2 T_1 - C_1 \ln r_1 = T_2 \tag{4.29}$$
$$T_2 - T_1 = C_1 \ln \frac{r_2}{r_1} \tag{4.30}$$
$$C_1 = \frac{T_2 - T_1}{\ln \frac{r_2}{r_1}} \tag{4.31}$$

So:

$$\frac{T_2 - T_1}{\ln \frac{r_2}{r_1}} \ln r_1 + C_2 = T_1 \tag{4.32}$$
$$C_2 = T_1 - \frac{T_2 - T_1}{\ln \frac{r_2}{r_1}} \tag{4.33}$$

I now have the values for C_1 and C_2. By inserting these values in $T(r_2)$ from expression 4.27 and rearranging, I now have an expression for the variation of temperature in the pipe. Which is:

$$T(r) = \frac{\ln \frac{r}{r_1}}{\ln \frac{r_2}{r_1}}(T_2 - T_1) + T_1 \tag{4.34}$$

In order to find the rate of loss by conduction I will have to use the Fourier's law of conduction 2.5 for a cylinder which is:

$$\dot{Q}_{cylinder} = -\kappa A \frac{\partial T}{\partial r} \tag{4.35}$$
$$= -\kappa(2\pi r L)\frac{C_1}{r} \tag{4.36}$$
$$= -2\pi\kappa L C_1 \tag{4.37}$$
$$= 2\pi\kappa L \frac{T_1 - T_2}{\ln \frac{r_2}{r_1}} \tag{4.38}$$

Where: $\kappa = 80,2(\frac{W}{m \cdot K})$(iron), $r_1 = 0,245m$(Diameter $= 0,49m$), $r_2 = 0,25m$(Diameter$=$ $0,5m$), $L = 200m$,$T_1 = 40°C$ and $T_2 = 39,9°C$. The inner diameter of this pipe is known, but the outer diameter is a assumption I have made. By inserting these values I get:

$$\dot{Q}_{conduction} = 2\pi(80,2(\frac{W}{m \cdot K}))(200m)\frac{40 - 39,9}{\ln \frac{0,25}{0,245}} \tag{4.39}$$

$$\dot{Q}_{conduction} = 0,498MW \tag{4.40}$$

Expression 4.38 could here be expressed in a more generalized form such as:

$$\dot{Q} = Sk(T_1 - T_2) \tag{4.41}$$

Where:

$$S = \frac{2\pi L}{\ln \frac{r_2}{r_1}} \tag{4.42}$$

Is the shape factor for a cylinder. This means that this part of the conduction loss is between the inner radius and the outer radius of the pipe. In order to find the total loss by conduction I will have to treat this section which I have just calculated as a isothermal cylinder buried in soil at a depth of $Z = 1m$,with a surface temperature of$T_{surface} = 5°C$ and a shape factor of:

$$S = \frac{2\pi L}{\ln \frac{4Z}{D_2}} \tag{4.43}$$

By adding these two losses together I now have the total loss by conduction from the cooling water:

$$\dot{Q}_{conduction} = 0,498MW + \frac{2\pi\kappa L(T_2 - T_{surface,soil})}{\ln \frac{4 \cdot Z}{D}} \tag{4.44}$$

$$\dot{Q}_{conduction} = 0,498MW + \frac{2 \cdot \pi \cdot \kappa \cdot L(39°C) - 5°C}{\ln \frac{4 \cdot 1m}{0,5m}} \tag{4.45}$$

$$\dot{Q}_{conduction} = 0,498MW + 4,93kW = 0,503MW \tag{4.46}$$

Since the result for the loss by conduction to the surface is $\frac{1}{100}$ of the loss to the pipe I have chosen to neglect this result from my calculation. By neglecting this small part I have also simplified my calculations a bit. I could now find the loss by convection and radiation by using expression 4.9, and 4.11 which gives after a small rearranging:

$$\dot{Q}_{combined} = \dot{Q}_{loss} - \dot{Q}_{conduction} \tag{4.47}$$

$$\dot{Q}_{combined} = 1,307MW - 0,498MW \tag{4.48}$$

$$\dot{Q}_{combined} = 0,809MW \tag{4.49}$$

Then by using expression 4.8 and rearranging this I find the rate for energy out of the system, which also is the effect of the heat exchanger:

$$\dot{Q}_{out} = \dot{Q}_{in} - \dot{Q}_{loss} \tag{4.50}$$
$$\dot{Q}_{out} = 15,687MW - 1,307MW \tag{4.51}$$
$$\dot{Q}_{out} = 14,380MW \tag{4.52}$$

Since the temperature in to the heat exchanger is known it is now possible to calculate the heat exchangers efficiency, maximum effect and area:

$$\dot{Q}_{warm} = \dot{m}c_{p,warm}(T_{in,warm} - T_{out,warm}) \tag{4.53}$$
$$\dot{Q}_{cold} = \dot{m}c_{p,cold}(T_{in,cold} - T_{out,cold}) \tag{4.54}$$

From these expressions with some rearranging, I find the expressions for the temperature out of the heat exchanger on both the warm and cold side:

$$T_{out,cold} = T_{in,cold} + \frac{\dot{Q}_{cold}}{C_c} \tag{4.55}$$
$$T_{out,warm} = T_{in,warm} - \frac{\dot{Q}_{warm}}{C_h} \tag{4.56}$$

Where:

$$C_h = \dot{m}c_{p,warm} \tag{4.57}$$
$$C_c = \dot{m}c_{p,cold} \tag{4.58}$$
$$C_c = 312,81\frac{kg}{s} \cdot 4,205\frac{kJ}{kg \cdot K}(fromtable.4.3) \tag{4.59}$$
$$C_c = 1315,366\frac{kJ}{K \cdot s} \tag{4.60}$$
$$C_h = 312,81\frac{kg}{s} \cdot 4,179\frac{kJ}{kg \cdot K}(fromtable.4.3) \tag{4.61}$$
$$C_h = 1307,232\frac{kJ}{K \cdot s} \tag{4.62}$$

I have set $T_{in,warm}$ to be $39°C$ and $T_{in,cold}$ to be $5°C$, since $\dot{Q}_{cold} = \dot{Q}_{warm} = \dot{Q}_{out}$, then by inserting into 4.56 I get:

$$T_{out,cold} = 5°C + \frac{14,380MW}{1315,366\frac{kJ}{K \cdot s}} \tag{4.63}$$
$$T_{out,warm} = 39°C - \frac{14,380MW}{1307,232\frac{kJ}{K \cdot s}} \tag{4.64}$$
$$T_{out,cold} = 15,932°C \tag{4.65}$$
$$T_{out,warm} = 27,999°C \tag{4.66}$$

By using equation 2.25 and 2.26 I find the area for the heat exchanger assuming U is $1700\frac{W}{m^2 \cdot ^\circ C}$. first I find the logarithmic temperature difference in the heat exchanger:

$$\Delta T_1 = T_{in,warm} - T_{out,cold} = 39^\circ C - 15,932^\circ C = 23,068^\circ C \tag{4.67}$$

$$\Delta T_2 = T_{out,warm} - T_{in,cold} = 27,999^\circ C - 5^\circ C = 22,999^\circ C \tag{4.68}$$

Then by inserting into eq. 2.26 I find the ΔT_{lm}:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} = \frac{23,068 - 22,999}{\ln \frac{23,068}{22,999}} = 23,033^\circ C \tag{4.69}$$

By rearranging equation 2.25 I find the area to be:

$$A = \frac{\dot{Q}}{U \cdot \Delta T_{lm}} = \frac{14,380MW}{1700\frac{W}{m^2 \cdot ^\circ C} \cdot 23,033^\circ C} = 367,248m^2 \tag{4.70}$$

The heat transfer effectiveness is given as:

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{4.71}$$

$$\dot{Q}_{max} = C_{min}(T_{in,warm} - T_{in,cold}) \tag{4.72}$$

C_{min} is the smallest of C_h and C_c, here $C_{min} = C_h$. So:

$$\dot{Q}_{max} = 1307,232\frac{kJ}{K \cdot s} \cdot (39^\circ C - 5^\circ C) = 44,445MW \tag{4.73}$$

$$\epsilon = \frac{14,380MW}{44,445MW} = 0,323 \tag{4.74}$$

4.2 Power consumption for Miljøbygget

For this section there are two approaches: First is to establish the heat loss from the building by only taking into account the overall heat transfer coefficient for walls, roof, floors and windows/doors for the building. The other approach is to start from the electrical bill for the building, and then try to estimate the amount of electrical energy which are used for heating of the building. For both these approaches there are some major uncertainties with establishing the overall heat transfer coefficient for the windows/doors, floors, walls, and roof. This is due to the fact that this building has been built over three separate stages, and now one knows exactly what materials each section has been built with. And for the second approach the uncertainties are mostly that each electrical heating ordinance consumes more electricity than it is capable of delivering, this results in a false reading of the electrical bill making the necessary heat needed lower then what it should be from the electrical bill. As a result I have created two different calculations of the energy need for the building.

A assumption I have made for this section of the energy balance at Elkem Salten, is the dimensioned outdoor temperature. This temperature is a measurement of the most extreme temperature which can be expected. The dimensioned outdoor temperature (DUT) (in norwegian: dimensjonerende ute- temperatur) is defined as the lowest registered average temperature for an area over a 72 hours period for the last thirty years. This is from [29] page 4. Tønnesen et.al [27] found that for the region where Elkem Salten Verk is located the dimensioning outdoor temperature is $-13°C$.

To calculate the power input for Miljøbygget I have a basis in the electrical bill for the building as an assumption. In this the total power consumed electricity is together with the electricity used for lighting, computers and water heaters. From Elkem Salten I have the total amount of lighting and the power of each lighting installation for the building:

- Total power consumption for the building pr. year: $1437,53MWh$.
- Lighting power for the first floor: $7002W$ pr. year: $61,33MWh$ assuming always on throughout the year.
- Lighting power for the basement: $6606W$ pr. year: $57,86MWh$ assuming always on throughout the year.
- Outdoor dimensioned temperature is $-13°C$.
- The indoor temperature is $25°C$.

Elements	Measures in meters
Windows (small)	0,72m · 0,68m
Windows (single frame)	1,65m · 1,00m
Windows (double frame)	1,65m · 2,15m
Windows (triple frame)	1,65m · 2,92m
Doors	2,4m · 2,4m
Floor (first section)	18,66m · 17,35m
Floor (second section)	20,77m · 21,16m
Floor (third section)	11,20m · 21,16m
Ceiling hight (first section)	5,6m (2· 2,8m)
Ceiling hight (second section)	5,6m (2· 2,8m)
Ceiling hight (third section)	3,5m + 2,8m
Roof	Sum of floor areas
Walls (first section)	18,66m · 5,6m + 2·(17,35m · 5,6m)
Walls (second section)	2,5m · 5,6m + 2·(20,77m · 5,6m)
Walls (third section)	21,16m · 6,3m + 2·(11,20m · 5,6m)

Table 4.4: Measures for the environmental building.

The area of the windows are measured by me, with a laser measuring device. The rest of the data presented in table 4.4 is provided by Elkem Salten. Due to the uncertainty of the laser measuring device ($\pm 5, 0 \cdot 10^{-4} m$), I have rounded my calculations to two decimal points.

The area of walls per section without windows are calculated as:

$$5,6m \cdot (18,66m + 2(17,36m)) \quad (4.75)$$
$$+5,6m \cdot (2,5m + 2(20,77m)) \quad (4.76)$$
$$+6,3m \cdot (21,16m + 2(11,20m)) \quad (4.77)$$
$$-\text{total area of windows} \quad (4.78)$$

Elements	Area (m^2)	Number of elements
Windows	112,70 (m^2)	63
Doors	17,28 (m^2)	3
Floor	1000,24 (m^2)	1
Walls	707,16 (m^2)	4
Roof	1000,24 (m^2)	1

Table 4.5: This table shows the total overall area- values for Miljøbygget.

By totalize of table 4.2, I find the total area for each surface which has thermal loss from the building:

$$112,70(m^2) \quad (4.79)$$
$$+ 17,28(m^2) \quad (4.80)$$
$$+ 1000,24(m^2) \quad (4.81)$$
$$+ 707,16(m^2) \quad (4.82)$$
$$+ 1000,24(m^2) \quad (4.83)$$
$$A_{total} = \underline{2837,62(m^2)} \quad (4.84)$$

The total electrical energy consumption for this building without the lighting is:

$$1.437,53(MWh) - (61,33(MWh) + 57,86(MWh)) = \quad (4.85)$$
$$1.437,53(MWh) - 119,20(MWh) = \quad (4.86)$$
$$= \underline{1.318,32(MWh)} \quad (4.87)$$

From this I now have to make an assumption of how much additional electrical energy goes to computers, hot water heaters, etc. Here I have set this to be 200 MWh, so by assumption the power used for heating is 1.118 ,32 MWh. This gives a energy rate of:

$$\dot{Q} = \frac{\text{Total energy}}{\text{Heating hours per year}} = \frac{1.118.320(kWh)}{365,25 \cdot 24} = 127,57kW \quad (4.88)$$

Then by using a combination of 2.6 and the thermal resistance meaning that the heat transfer coefficient h is the same as the overall heat transfer coefficient U. I can now calculate the average overall heat transfer coefficient for the entire building:

$$127,57kW = U_{average} \cdot A_{total}(T_{indoor} - T_{dimensioned}) \tag{4.89}$$

By rearranging 4.89 and inserting the numerical values for: Total area for the building, indoor temperature and dimensioned outdoor temperature i have:

$$U_{average} = \frac{127,57kW}{2837,62(m^2) \cdot 38(^{\circ}C)} = 1,18(\frac{W}{m^2 \cdot^{\circ} C}) \tag{4.90}$$

By comparison the current average overall heat transfer coefficient for buildings given by the TEK- 10 Norwegian building standard from [11].

Elements	U ($\frac{W}{m^2 \cdot ^{\circ}C}$)
Windows	1,6 ($\frac{W}{m^2 \cdot ^{\circ}C}$)
Floor	0,18 ($\frac{W}{m^2 \cdot ^{\circ}C}$)
Walls	0,22 ($\frac{W}{m^2 \cdot ^{\circ}C}$)
Roof	0,18 ($\frac{W}{m^2 \cdot ^{\circ}C}$)

Table 4.6: Minimum values for the overall heat transfer coefficient from [11].

The Norwegian building standards from 1997 (TEK-97), has the same minimum demands regarding overall heat transfer coefficients as the numerical values presented in table 4.6 [5]. This gives an average overall heat transfer coefficient for the building of:

$$U_{average(TEK-10)} = \frac{1,6 + 0,22 + 0,18 + 0,18}{4} = 0,54(\frac{W}{m^2 \cdot^{\circ} C}) \tag{4.91}$$

This gives a difference in percent of:

$$\frac{1,18}{0,54} \cdot 100\% = 218,5\% \tag{4.92}$$

So the overall heat transfer coefficient i have calculated from my assumptions is $218,5\%$ higher then the TEK-10 standard.

Chapter 5

Results

This chapter of my thesis will describe the different parts of my suggested heating system with calculations for each section.

5.1 The cooling systems heat central

The heat central for the cooling system is a heat exchanger, which is connected to each of the furnaces cooling system with a branched connection and valves. From this side the heat exchanger receives hot cooling water from one or all furnaces after the cooling water has been collected in the sink. The other side of the heat exchanger is connected to a closed loop with a pump. From the loop the thermal energy is transferred to all components that currently are utilizing the hot water from the cooling system as their energy source. These components are the Lakselva soccer pitch, the garden centre and the fish farm. For the heat central system to obtain a high efficiency of thermal transfer the pumps capacity must be eqaual to the maximum obtainable heat transfer rate, in order to deliver the desired amount of heat.

5.1.1 Assumptions for the heat central:

- Stable operation conditions and the consumption of water is the same throughout the year.
- The heat central is well insulated and the heat exchanger effectiveness is $\eta = 1$.
- There is no loss of thermal energy in the pipes before the heat central.
- The energy removed from the system is only by thermal transfer inside the heat central.
- The change in c_p from 25 to $30°C$ is linear.

5.1.2 Calculations for the heat central:

The sum of branch currents is equal to the total current:

$$\dot{m}_{\text{branch furnace one}} + \dot{m}_{\text{branch furnace two}} + \dot{m}_{\text{branch furnace three}} = \dot{m}_{\text{Total into heat central}} \tag{5.1}$$

So:

$$\Sigma_{i=1}^{3}\dot{m}_{\text{branch, i}} = \dot{m}_{\text{Total}} \tag{5.2}$$

And:

$$\frac{\dot{m}_{\text{Total}}}{3} = \dot{m}_{\text{branch}} \tag{5.3}$$

By equation 5.3 if $\dot{m}_{\text{Total}} = 312,81(\frac{kg}{s})$ then:

$$\dot{m}_{\text{branch}} = \frac{312,81}{3} = 104,27(\frac{kg}{s}) \tag{5.4}$$

Now by adding the rates of energy taken from the main cooling system of the furnaces I get:

$$\text{Lakselva artificial gras pitch} + \text{Sisomar} + \text{Miljøbygget} = \text{The total energy recycled} \tag{5.5}$$

$$628kW + 4514kW + 127,57kW = 5269,57kW \tag{5.6}$$

So:

$$\dot{Q}_{hc} = 5269,57kW \tag{5.7}$$

Then by using 2.4 and using the branch current for one furnace on the heat central loop side I will find the temperature difference.

$$(\Delta T) = \frac{\dot{Q}_{hc}}{\dot{m} \cdot c_p} \tag{5.8}$$

$$(\Delta T) = \frac{5269,57}{104,27(\frac{kg}{s}) \cdot 4,179(\frac{kJ}{kg \cdot K})} \tag{5.9}$$

$$(\Delta T) = 12,09K \tag{5.10}$$

Where:

$$\Delta T = T_{\text{out of heat exchanger}} - T_{\text{Into heat exchanger}} \tag{5.11}$$

In order to determine $T_{\text{out of heat exchanger}}$ I will have to find the temperatures of the heat exchanger on the furnaces side. If the heat central is placed in close proximity to the main factory building I assume no loss from the furnaces to the heat central. So the temperature of cooling water from the furnaces to the heat central is assumed to be $40°C$ Which in turn gives me from 5.11:

$$40°C - 12,09°C = 27,91°C \tag{5.12}$$

For the "warm" side of the heat exchanger or the furnace side of the heat exchanger. By assuming that the recycled energy is equal to the energy removed from the heat central system. I now have the same temperature out on the "warm" side as in to the "cold" side of the heat centrals heat exchanger. This gives the same ΔT for the cold side. So the temperature drop over the "cold" loop will be the same as for the "warm" side:

$$40°C - 12,09°C = 27,91°C \tag{5.13}$$

By assuming that the change in c_p between $25°C$ and $30°C$ from table 4.1.1 is linear then the change per $1°C$ is:

$$\text{per}1°C = \frac{c_p(25°C) - c_p(30°C)}{5} \tag{5.14}$$

And per $0,1°C$ is:

$$0,1°C = \frac{c_p(25°C) - c_p(30°C)}{50} \tag{5.15}$$

$$0,1°C = \frac{4,180 - 4,178}{50} = 4,0 \cdot 10^{-5} \tag{5.16}$$

Multiplied by the difference between $27,91°C$ and $25°C$. Which is $2.91°C$:

$$4,0 \cdot 10^{-5} \cdot 29,1 = 1,164 \cdot 10^{-3} \tag{5.17}$$

Since the difference is given per $0,1°C$. Then the $c_p(27,91°C)$ is given as:

$$c_p(27,91) = c_p(25) - 1,164 \cdot 10^{-3} = 4,17883(\frac{kJ}{kg \cdot K}) \tag{5.18}$$

I use equation 4.62 to find the values for C_c and C_h:

$$C_c = \dot{m}_{branch} \cdot c_p(27,91) = 435,72(\frac{kJ}{s \cdot K}) \tag{5.19}$$

$$C_h = \dot{m}_{branch} \cdot c_p(40) = 435,74(\frac{kJ}{s \cdot K}) \tag{5.20}$$

So:

$$C_{min} = C_c \tag{5.21}$$

$$T_{out,cold} = 27,91°C + \frac{5269,57(kW)}{435,72(\frac{kJ}{s \cdot K})} = 39,9°C \tag{5.22}$$

$$T_{out,hot} = 40°C - \frac{5269,57kW}{435,74(\frac{kJ}{s \cdot K})} = 27,90 \tag{5.23}$$

This gives the values for ΔT_1 and ΔT_2 :

$$\Delta T_1 = 40°C - 39,9°C = 0,01 \tag{5.24}$$

$$\Delta T_2 = 27,91°C - 27,90°C = 0,01 \tag{5.25}$$

ΔT_{lm} is then:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \tag{5.26}$$

$$= \frac{0,01 - 0,01}{\ln \frac{0,01}{0,01}} \tag{5.27}$$

$$= 0,01 \tag{5.28}$$

Here $\Delta T_{lm} = \frac{0}{0}$ an expression which is indetermine. In this case, it can be shown by l'hôpital's rule that $\Delta T_{lm} = \Delta T_1 = \Delta T_2$ [12] (page 644). The area for the heat centrals heat exchangeer is then found to be:

$$A_{hc} = \frac{\dot{Q}_{hc}}{U_{hc} \cdot \Delta T_{lm}} \tag{5.29}$$

$$= \frac{5269.57(kW)}{1700(\frac{W}{m^2 \cdot K}) \cdot 0,01°C} \tag{5.30}$$

$$= 309,97m^2 \tag{5.31}$$

$$\dot{Q}_{max} = C_{min} \cdot (T_{in,hot} - T_{in,cold}) \tag{5.32}$$

$$= 435,72(\frac{kJ}{s \cdot K}) \cdot (12,09(°C)) \tag{5.33}$$

$$= 5267,85(kW) \tag{5.34}$$

So the heat exchanger effectiveness for the heat central (η_{hc}) is:

$$\eta_{hc} = \frac{\dot{Q}_{hc}}{\dot{Q}_{max,hc}} \tag{5.35}$$

$$= \frac{5269,57(kW)}{5267,85(kW)} \tag{5.36}$$

$$= 1 \tag{5.37}$$

5.2 The heating system for Miljøbygget

The heating system for Miljøbygget is divided into three separate systems, which all use hot water as their energy source. The first system is a water based surface heating system, the second is a spot heating system, and the third is a ventilation heating system. As a fourth system and an addition or possible combination to these systems a water heating system is described and calculated. The water beaded surface heating system I have chose to calculate is a floor heating system, which utilizes hot water to provide heating inside the building by pipeline casted in the concrete floor.

5.2.1 Assumptions for the water based heating system:

- The temperature of hot cooling water in to the building is $40°C$.
- The temperature of hot cooling water out of the building is $35°C$.
- The rate of energy needed for heating is: 127,57 kW.
- The flow is fully developed.
- The system is in steady state.
- The diameter of the water heating system is 20mm, and is separated with 0,3m.
- Air is an ideal gas.
- The physical properties of air at $24,5°C$ at 1 atm. are equal to the physical properties of air at $25°C$ at 1 atm.

5.2.2 Calculations for the water based heating system:

This section contains the calculations for the heating system. In order to determine the heat flux through the floor I need to determine the convective and radiative heat transfer coefficients for the floor.

First I use equation 2.12 to determine the characteristic length for the floor inside miljøbygget, this is neccesary for the calculation of the heat flux through the floor for the building. I have therefore in my calculations used one large surface, which is the sum of the areas for both floors:

$$L_c = \frac{40m \cdot 50m}{2 \cdot 40m + 2 \cdot 50m} = \frac{2000m^2}{180m} = 11,11m \tag{5.38}$$

Then the film temperature is determined (T_f):

$$T_f = \frac{T_s + T_\infty}{2} = \frac{27°C + 22°C}{2} = 24,5°C = 297,65K \tag{5.39}$$

The Rayleigh number is defined as:

$$Ra_{L_c} = \frac{g \cdot \beta(T_s - T_\infty)L_c^3}{v^2} \cdot Pr \tag{5.40}$$

	κ	v	Pr	ρ	c_p
Air (20°C)	0,02514 ($\frac{W}{m \cdot K}$)	$1.516 \cdot 10^{-5}$ ($\frac{m^2}{s}$)	0.7309 (Pr)	1,204 ($\frac{kg}{m^3}$)	1,007 ($\frac{kJ}{kg \cdot K}$)
Air (25°C)	0,02551 ($\frac{W}{m \cdot K}$)	$1.562 \cdot 10^{-5}$ ($\frac{m^2}{s}$)	0,7296 (Pr)	1,184 ($\frac{kg}{m^3}$)	1,007 ($\frac{kJ}{kg \cdot K}$)
Air(30°C)	0,02588 ($\frac{W}{m \cdot K}$)	$1.608 \cdot 10^{-5}$ ($\frac{m^2}{s}$)	0,7282 (Pr)	1,164 ($\frac{kg}{m^3}$)	1,007 ($\frac{kJ}{kg \cdot K}$)

Table 5.1: Numerical values from Table A-15 in [12].

By inserting the numerical values from 5.1(Air 25°C) , 5.38 and 5.39 into equation 5.40 I find the Rayleigh number (Ra_L):

$$Ra_{L_c} = \frac{9,81(\frac{m}{s^2}) \cdot \frac{1}{297,65K} \cdot (27°C - 22°C) \cdot 11,11m^3}{(1,562 \cdot 10^{-5})^2} \cdot 0,7296 = 5,22 \cdot 10^{10} \tag{5.41}$$

Then the Nusselt number is according to equation 2.13 in the upwards direction from the topside of the floor. Where $C = 0,54$ and $^n = 1/4$ from [12] for the upwards direction. For the downwards direction $C = 0,27$ and $^n = 1/4$. So:

$$Nu_{up} = 0,54 \cdot Ra_{L_c}^{1/4} = 0,54 \cdot (5,22 \cdot 10^{10})^{1/4} = 258,11 \tag{5.42}$$

$$Nu_{down} = 0,27 \cdot Ra_{L_c}^{1/4} = 0,27 \cdot (5,22 \cdot 10^{10})^{1/4} = 129,05 \tag{5.43}$$

Then by using the other relation for for the Nusselt number from 2.11 The convective heat transfer coefficient h_{conv} is calculated:

$$\begin{aligned} h_{conv} &= \frac{\kappa_{air(25°C)}}{L_C} \cdot Nu_{up} + \frac{\kappa_{air(25°C)}}{L_C} \cdot Nu_{down} && (5.44) \\ &= \frac{0,02551(\frac{W}{m \cdot K})}{11,11m} \cdot (258,11) + \frac{0,02551(\frac{W}{m \cdot K})}{11,11m} \cdot (129,05) && (5.45) \\ &= \frac{0,02551(\frac{W}{m \cdot K})}{11,11m} \cdot (258,11 + 129,05) && (5.46) \\ &= 0,88(\frac{W}{m^2 \cdot K}) && (5.47) \end{aligned}$$

This is the heat transfer coefficient for the contribution of $h_{combined}$ by convection. The contribution from radiation to $h_{combined}$ is found by using equation 2.10:

$$h_{rad} = \epsilon\sigma(T_s + T_\infty)(T_s^2 + T_\infty^2) \tag{5.48}$$

Here the numerical values are inserted into the equation, and ϵ is set to be 1 by assuming that the floor surface is black, resulting in the maximum contribution from radiation:

$$h_{rad} = 1 \cdot (5,67 \cdot 10^{-8} \frac{W}{m^2 \cdot K^4}) \cdot (300,15K + 295,15K) \cdot ((300,15K)^2 + (295,15K)^2) = 5,98(\frac{W}{m^2 \cdot K}) \tag{5.49}$$

$h_{combined}$ is then given as:

$$h_{combined} = 0,88(\frac{W}{m^2 \cdot K}) + 5,98(\frac{W}{m^2 \cdot K}) = 6,86\frac{W}{m^2 \cdot K} \tag{5.50}$$

Layer (from bottom)	Thickness s (mm)	thermal resistance κ ($\frac{W}{m \cdot K}$)
1	500 (mm)	Concrete $\kappa = 1{,}7$
2	100 (mm)	Polystrene $\kappa = 0{,}035$
3	30 (mm)	Concrete $\kappa = 1{,}7$
Pipes		
4	50 (mm)	Concrete $\kappa = 1{,}7$
5	10 (mm)	linoleum / PVC $\kappa = 0{,}2$

Table 5.2: Suggestion for a possible build of floor. Based on example 2.1 page 32 in [29].

In order to find the heat transfer in the upwards and downwards direction I first need to calculate the numerical value for $h_{combined}$ and the overall heat transfer coefficient in both directions by using the values presented in table 5.2:

The ΔT presented in 2.30 and 2.31 is given as:

$$\Delta T = T_m - T_a \tag{5.51}$$

Where T_m is the average temperature in the pipeline, and T_a is the set air temperature inside the building.

When the set air temperature is $22°C$ and the average temperature in the pipeline is set to be $37,5°C$ from:

$$T_m = \frac{40 + 35}{2} = 37,5°C \tag{5.52}$$

Then the ΔT from 5.51 is:

$$\Delta T = 37,5 - 22 = 15,5°C \tag{5.53}$$

I now find the overall thermal resistance for the upwards and downwards directions by using equations 2.32 and 2.33:

$$R_{up} = \frac{1}{6,86\frac{W}{m^2 \cdot K}} + \frac{0,01m}{0,2\frac{W}{m \cdot K}} + \frac{0,05m}{1,7\frac{W}{m \cdot K}} = 0,22(\frac{m^2 \cdot K}{W}) \tag{5.54}$$

And:

$$R_{down} = \frac{1}{6,86\frac{W}{m^2\cdot K}} + \frac{0,03m}{1,7\frac{W}{m\cdot K}} + \frac{0,1m}{0,035\frac{W}{m\cdot K}} + \frac{0,5m}{1,7\frac{W}{m\cdot K}} = 3,31(\frac{m^2\cdot K}{W}) \tag{5.55}$$

The thermal transfer in the upwards direction given in equation 2.30:

$$\dot{q}_{up} = \frac{15,5}{\frac{1,15\cdot 0,3\cdot(3,31+0,22)\cdot\log\frac{0,3}{0,02\cdot\pi}}{\pi\cdot 1,7\cdot 3,31} + 0,22} = 47,29(\frac{W}{m^2}) \tag{5.56}$$

There thermal transfer in the downwards direction from equation 2.31:

$$\dot{q}_{down} = \frac{15,5}{\frac{1,15\cdot 0,3\cdot(3,29+0,20)\cdot\log\frac{0,3}{0,02\cdot\pi}}{\pi\cdot 1,7\cdot 0,20} + 3,29} = 3,14(\frac{W}{m^2}) \tag{5.57}$$

So for this system The total thermal energy per square meter of floor is:

$$\dot{q}_{up} + \dot{q}_{down} = \text{Total thermal energy needed} \tag{5.58}$$

$$47,29\frac{W}{m^2} + 3,14\frac{W}{m^2} = 50,43\frac{W}{m^2} \tag{5.59}$$

For this building the office area is about 50 percent of the total area and the wardrobes makes the last 50 percent of area. The building has in total about $2000m^2$ area divided over two floors. The wardrobe area needs to hold a higher temperature due to air drying of clothing, towels and increased comfort due to exposed skin. The comfort temperature for wardrobes is between $25°C$ and $29°C$ Table 2.1 page 31 [29]. This system does then need a total rate of thermal energy:

$$\dot{Q} = 50,43(\frac{W}{m^2})\cdot 2000m^2\cdot\frac{1}{1000} = 100,86kW \tag{5.60}$$

This leaves from the buildings energy balance of:

$$127,57kW - 100,86kW = 26,71kW \tag{5.61}$$

This surplus could be used as a starting point for spot heating sources for the increase in air temperature in this half of the building. If this surplus is inserted in equation 5.56 and solved with ΔT set as unknown I get:

$$26,71(\frac{W}{m^2}) = \frac{\Delta T}{\frac{1,15\cdot 0,3\cdot(3,29+0,20)\cdot\log\frac{0,3}{0,02\cdot\pi}}{\pi\cdot 1,7\cdot 3,29} + 0,22} \tag{5.62}$$

Then:

$$26,71 = 3,0157\Delta T \tag{5.63}$$

$$\Delta T = 8,75 \tag{5.64}$$

By adding this ΔT to the set air temperature for the building this should give me the air temperature for the wardrobe section of the building:

$$22°C + \Delta T = 22 + 8,75 = 30,75°C \tag{5.65}$$

The mass flow rate through this system will be with one loop for the floor heating, and one loop for the spot heating sources given as:

$$\dot{m} = \frac{\dot{Q}}{c_p \cdot \Delta T} \tag{5.66}$$

which is a rewritten version of equation 2.4 :

$$\dot{m}_{floor} = \frac{\dot{Q}_{floors}}{c_p(40°C) \cdot (40°C - 35°C)} = \frac{100,86kW}{4,179\frac{kJ}{kg \cdot K} \cdot 5K} = 4,82\frac{kg}{s} \tag{5.67}$$

And:

$$\dot{m}_{spot} = \frac{\dot{Q}_{spot}}{c_p(40°C) \cdot (40°C - 35°C)} = \frac{26,71kW}{4,179\frac{kJ}{kg \cdot K} \cdot 5K} = 1,27\frac{kg}{s} \tag{5.68}$$

So the total mass flow rate of water for this system is:

$$1,27\frac{kg}{s} + 4,82\frac{kg}{s} = 6,09\frac{kg}{s} \tag{5.69}$$

5.2.3 Spot heating system

This system is a example of a spot heating system, where radiation heat strips without wafers are used for the calculations. The objective her is to calculate the length necessary for the radiation heat strip. In order to deliver 127,57 kW of thermal energy to the building.

Assumptions:

- The temperature of hot cooling water in to the building is $40°C$.
- The temperature of hot cooling water out of the building is $35°C$.
- Steady operating conditions.
- Air is an ideal gas.
- Assume no loss in pipes.
- Assume room temperature of walls.
- Air pressure is 1 atm.

- Assume surface temperature for the radiation heat strip is $30°C$.
- Assume room temperature $20°C$.
- Outer diameter of the radiating heat strip is 0,08m.
- Assume the pipe surface is black oxidized copper with emissivity ($\epsilon = 0,78$) from table A- 18 [12], and thermal conductivity ($commercialbronze = 52(\frac{W}{m \cdot K})$) table A-3 [12].
- Assume the pipe is 0,01m thick.
- Assume the needed energy for heating is $\dot{Q} = 127,57kW$.

Calculations:

The film temperature around the radiation heat strip is with temperatures room ($20°C = 293,17K$) and surface ($30°C = 303,15K$) from equation 5.39:

$$T_f = \frac{30°C + 20°C}{2} = 25°C = 298,15K \tag{5.70}$$

Physical values for air at $25°C$ are taken from table 5.1. The characteristic length of the radiating heat strip is found in table 9-1 page 528 in [12] which states:

$$L_c = D \tag{5.71}$$

Following assumptions:

$$L_c = 0,08m \tag{5.72}$$

. The rayleigh number is determined to be:

$$Ra_{L_c} = \frac{9,81(\frac{m}{s^2}) \cdot \frac{1}{298,15K} \cdot (30°C - 20°C) \cdot 0,08m^3}{(1,562 \cdot 10^{-5})^2} \cdot 0,7296 = 5,037 \cdot 10^5 \tag{5.73}$$

From equation 5.40 The equation for the Nusselt number is determined by geometry. It is found in table 9-1 page 528 in [12] and is in this case given as:

$$Nu = (0,6 + \frac{0,387 \cdot Ra_{L_c}^{1/6}}{(1 + (0,559/Pr)^{9/16})^{8/27}})^2 \tag{5.74}$$

So by inserting numerical values the Nusselt number is calculated to be:

$$Nu = (0,6 + \frac{0,387 \cdot (5,037 \cdot 10^5)^{1/6}}{(1 + (0,559/0,7298)^{9/16})^{8/27}})^2 = 12,05 \tag{5.75}$$

The heat transfer coefficient is calculated by equation 2.11. The convection heat transfer coefficient is calculated to be:

$$h = \frac{0,02551(\frac{W}{m \cdot K})}{0,08m} \cdot 12,05 = 3,48\frac{W}{m^2 \cdot K} \tag{5.76}$$

The Radiation heat transfer coefficient is calculated by using equation 2.10:

$$h_{rad} = \epsilon \cdot \sigma \cdot (T_s + T_{surr}) \cdot (T_s^2 + T_{surr}^2) \tag{5.77}$$

Inserting numerical values, and calculating the radiation heat transfer coefficient:

$$0,78 \cdot (5,67 \cdot 10^{-8}\frac{W}{m^2 \cdot K^4}) \cdot (303,15K + 293,15K) \cdot (303,15K^2 + 293,15K^2) = 4,69\frac{W}{m^2 \cdot K} \tag{5.78}$$

The combined heat transfer coefficient is then:

$$h_{convection} + h_{radiation} = 3,48\frac{W}{m^2 \cdot K} + 4,69\frac{W}{m^2 \cdot K} = 8,17\frac{W}{m^2 \cdot K} \tag{5.79}$$

The loss from the pipe by conduction is given as:

$$\dot{Q}_{conduction} = 2\pi\kappa L\frac{T_1 - T_2}{\ln\frac{r_2}{r_1}} \tag{5.80}$$

From equation 4.38. Following assumptions and equation 2.2, the expression for the total thermal energy needed for heating is:

$$\dot{Q} = \dot{Q}_{combined} + \dot{Q}_{conduction} \tag{5.81}$$

With numerical values:

$$127,57kW = 8,17(\frac{W}{m^2 \cdot K}) \cdot \pi \cdot 0,08m \cdot L \cdot (10K) + 2 \cdot \pi \cdot 52(\frac{W}{m \cdot K}) \cdot L \cdot \frac{10K}{\ln\frac{0,04m}{0,035m}} \tag{5.82}$$

Since L is unknown I solve equation 5.82 with regards to L:

$$\begin{align} 127,57kW &= 20,53(W/m) \cdot L + 24.468,07(W/m) \cdot L \tag{5.83} \\ &= 24.488,60(W/m) \cdot L \tag{5.84} \end{align}$$

So:

$$L = \frac{127,57kW}{24,48(kW/M)} = 5,21m \tag{5.85}$$

The mass flow rate for such a system is given by equation 5.67:

$$\dot{m} = \frac{127,57}{4,179\frac{kJ}{kg \cdot K} \cdot 5K} = 6,10\frac{kg}{s} \tag{5.86}$$

5.2.4 Ventilation heating system

The ventilation heating system is a system which uses hot cooling water to heat incoming ventilation air up to a desired temperature. For the ventilation system there are some demands provided by the norwegian government, for a building used for office section and for the wardrobe section. These demands involves the minimum rate of air being substituted inside the building. I will here try to determine the rate of air substitution necessary for this building, and the rate of thermal energy needed for heating.

Assumptions for the ventilation system:

- The temperature of hot cooling water in to the building is $40°C$.
- The temperature of hot cooling water out of the building is $35°C$.
- 50% of the building is wardrobe.
- 50% of the building is offices.
- The total floor area for this building is 2000 m^2.
- T_i is set at $20°C$ for the office section of the building.
- T_{iw} is set at $30°C$ for the wardrobe section of the building.
- T_{DUT} is set at $-13°C$.
- Assume the buildings ventilation is designed for the presence of 100 persons.
- The heat recyclers efficiency is: 0,7.

Calculations for the ventilation system:

Type of demand	Rate of air substituted
The structure itself	1 $(\frac{l}{s \cdot m^2})$
Rate of substituted air per person	7 $(\frac{l}{s})$
Wardrobe ventilation	1,8 $\frac{m^3}{h \cdot m^2}$
Calculation factor	1 $\frac{l}{s} = 3,6 \frac{m^3}{h}$

Table 5.3: Table of demands for Ventilation from [7].

The Formula for calculating the energy needed for the ventilation system is presented in equation 2.34:

$$\dot{Q} = \dot{v} \cdot C_p \cdot \rho \cdot (T_i - T_{DUT}) \tag{5.87}$$

From table 5.1 the physical properties for density (ρ) and specific heat (c_p) are found, and since the temperatures T_i, T_{iw} and T_{DUT} are assumed to be $20°C$ for T_i, $30°C$ for T_{iw} and $-13°C$ for T_{DUT}, the only variable i need in order to determine the rates of energy needed is $\dot{v}$. Which is found by summarizing the rates of substituted air for each demand:

$$\dot{v}_{building} = 3.6\frac{m^3}{h \cdot m^2} \cdot 2000m^2 = 7200\frac{m^3}{h} \tag{5.88}$$

$$\dot{v}_{perperson} = 700\frac{l}{s} \cdot 3,6\frac{m^3}{h} = 2520\frac{m^3}{h} \tag{5.89}$$

$$\dot{v}_{wardrobe} = 1000m^2 \cdot 1,8\frac{m^3}{h \cdot m^2} = 1800\frac{m^3}{h} \tag{5.90}$$

These are the total rates of substituted air for both the wardrobe section and the office section of the building. By dividing the substitution rate for the building ($\dot{v}_{building}$) and the rate per person ($\dot{v}_{perperson}$) by a factor of two, I will then have the substitution rates for each section of the building. So the total rate of air substitution for the office section is:

$$\frac{7200(\frac{m^3}{h})}{2} + \frac{2520(\frac{m^3}{h})}{2} = 4860(\frac{m^3}{h}) \cdot \frac{1}{60minutes \cdot 60seconds} = 1,35\frac{m^3}{s} \tag{5.91}$$

And for the wardrobe section:

$$\frac{7200(\frac{m^3}{h})}{2} + \frac{2520(\frac{m^3}{h})}{2} + 1800\frac{m^3}{h} = 5660(\frac{m^3}{h}) \cdot \frac{1}{60minutes \cdot 60seconds} = 1,57\frac{m^3}{s} \tag{5.92}$$

Then by inserting the numerical values into equation 2.34 I can determine the rates of energy. First I find the rate of thermal energy for the office section of the building:

$$\dot{Q}_{office} = 1,35(\frac{m^3}{s}) \cdot 1,007(\frac{kJ}{kg \cdot K}) \cdot 1,204(\frac{kg}{m^3}) \cdot (20°C - (-13°C)) \tag{5.93}$$

$$= 54,01kW \tag{5.94}$$

And the Wardrobe section:

$$\dot{Q}_{wardrobe} = 1,57(\frac{m^3}{s}) \cdot 1,007(\frac{kJ}{kg \cdot K}) \cdot 1,164(\frac{kg}{m^3}) \cdot (30°C - (-13°C)) \tag{5.95}$$

$$= 79,13kW \tag{5.96}$$

The total rate of thermal energy needed for the ventilation system is the sum of energy rates for the office section and the wardrobe section:

$$\dot{Q}_{office} + \dot{Q}_{wardrobe} = 54,01kW + 79,13kW = 133,14kW \tag{5.97}$$

For this system to be utilized for heating a exchange of thermal energy from the furnace cooling water to the ventilation system is needed. The flow rate of water through this exchanger is:

$$\dot{m} = \frac{\dot{Q}}{c_p \cdot (T_{in} - T_{out})} \tag{5.98}$$

Where T_{in} and T_{out} is the set temperature difference through the heat exchanger. By setting T_{in} equal to $40°C$, T_{out} equal to $35°C$ and c_p equal to the specific heat of water at $40°C$ found in table 4.1.1. The rate of circulated water is:

$$\dot{m} = \frac{133,14kW}{4,179\frac{kJ}{kg\cdot K} \cdot 5K} = 6,37\frac{kg}{s} \tag{5.99}$$

By using my assumption with a heat recycler which has the efficiency of 0,7 the thermal heat transfer rate of a ventilation system with a heat recycler installed is given as:

$$\dot{Q} = \dot{v} \cdot C_p \cdot \rho \cdot (T_i - T_{DUT}) \cdot (1 - \eta) \tag{5.100}$$

For the office section:

$$\dot{Q}_{recycle,office} = 1,35(\frac{m^3}{s}) \cdot 1,007(\frac{kJ}{kg \cdot K}) \cdot 1,204(\frac{kg}{m^3}) \cdot (20°C - (-13°C)) \cdot (1 - 0,7) \tag{5.101}$$

$$= 16,20kW \tag{5.102}$$

For the wardrobe section:

$$\dot{Q}_{recycle,wardrobe} = 1,57(\frac{m^3}{s}) \cdot 1,007(\frac{kJ}{kg \cdot K}) \cdot 1,164(\frac{kg}{m^3}) \cdot (30°C - (-13°C)) \cdot (1 - 0,7) \tag{5.103}$$

$$= 23,74kW \tag{5.104}$$

The total rate of theremal energy with a heat recycler is:

$$= \dot{Q}_{recycle,office} + \dot{Q}_{wardrobe} = 16,20kW + 23,74kW = 39,94kW \tag{5.105}$$

Which then gives a mass flow rate of circulated water:

$$\dot{m} = \frac{39,94kW}{4,179\frac{kJ}{kg\cdot K} \cdot 5K} = 1,91\frac{kg}{s} \tag{5.106}$$

5.2.5 Wastewater heat recycling with heat pump

Although all the above mentioned thermal recycling methods are in fact wastewater heat recycling, the system I now will describe uses a heat pump to produce warm water. In this context this system will produce and cover the buildings hot freshwater consumption by lifting the temperature of the cooling water to a higher temperature in the region of $65°C$. This temperature is considered to be safe with regards to the legionella pneumophila bacteria, which is toxic for humans if airborne. The bacteria is found naturally in freshwater, but propagates in hot water if the temperatures are between $20°C$ and $45°C$, with a peak in propagation in the temperature interval between $30°C$ and $43°C$ [29]. In order to calculate the thermal energy rates for this system i have chosen to use R134a as working fluid for the heat pump. Wastewater heat recycling is in this context mean as recycling the thermal energy from the cooling system and utilizing this energy in combination with a heat pump to produce warm water for the showers. The amount of warm water from water heaters is unknown, but from my earlier estimations the power used is in the region of 150 MWh pr. year. To find the specific enthalpies for the heat pump cycle a software called Coolpack was used. This software is free, and can be downloaded from the developers internet site [22].

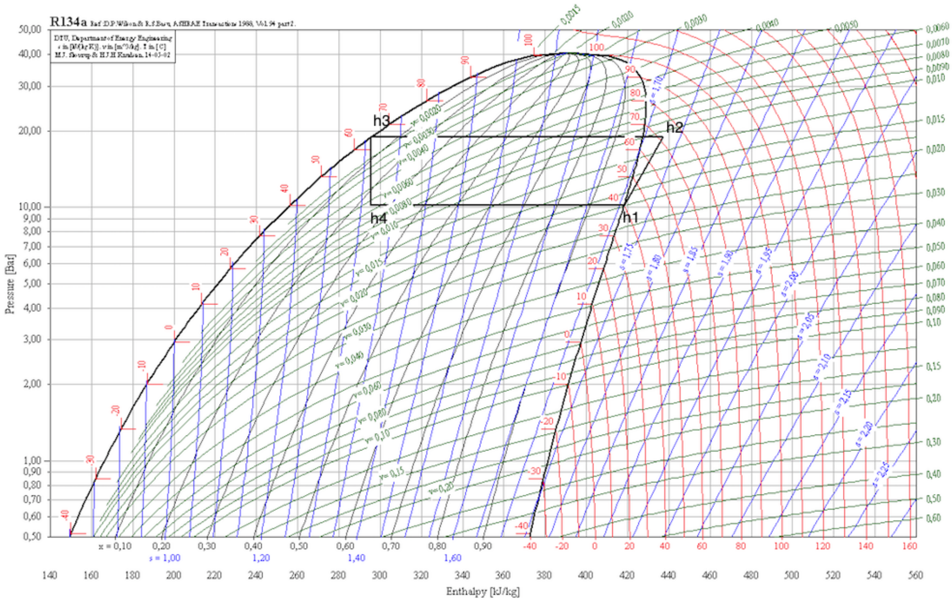

Figure 5.1: A screen dump from cool pack with the logarithmic pressure enthalpy diagram for the medium R-134a [22].

From figure 5.1 which is the same figure as 2.6 where the heat cycle is drawn into the logarithmic pressure enthalpy diagram, and the following values are found by interpolation:

Type of enthalpy	Specific Value ($\frac{kJ}{kg}$)
Enthalpy of saturated vapor (h_1)	419
Enthalpy of Isentropic compression (h_2)	433
Enthalpy of saturated liquid (h_3)	295
Enthalpy after throttle (h_4)	295

Table 5.4: Numerical values from the logarithmic pressure enthalpy diagram.

Assumptions for the wastewater heat recycling with heat pump:

- The energy needed is 150 MWh per year.
- The working fluid is R134a.
- The temperature into the condenser is $67,5°C = 340,65K$.
- The temperature out of the condenser is $65°C = 338,15K$.
- The temperature into the evaporator is $37,5°C = 310,65K$.
- The temperature out of the evaporator is $40°C = 313,15K$.
- The evaporator and condensers heat transfer efficiency is 1 ($\eta = 1$).
- The isentropic efficiency of the compressor is $\eta_{is} = 0,64$.

The reasoning for the assumption of energy needed for hot water equal to 150 MWh per year is: From [29] table 7.2 page 329 provides a numerical value of 3 kWh pr shower. So if I assume there are 150 persons using the showers per day, the total electrical consumption per day is:

$$150\text{persons per day} \cdot 3\text{kWh per shower} = 4500\text{kWh per day}. \tag{5.107}$$

From [24] a man- year worked is 1750 hours holidays not excluded. If a week of work is defined as 37,5 hours for seven week days [8](should not exceed 38 hours per week). The total amount of days of work is:

$$\text{Work days per year} = \frac{1750hperyear}{\frac{37{,}5hperweek}{7daysperweek}} = \frac{1750hperyear}{5,36hperday} = 326,5daysperyear \tag{5.108}$$

In turn the total electrical consumption used for showers is:

$$\text{Total electrical consumption used for showers per year} = 450\text{kWh per day} \cdot 326,5daysperyear \tag{5.109}$$

$$= 146925\text{kWh per year} \tag{5.110}$$

The remaining electrical consumption in order to achieve 150.000 kWh per year, I assume to be from other hot water consuming apliances such as sinks, dishwashers and laundry machines.

Calculations for the wastewater heat recycling with heat pump:

The thermal energy needed from the condenser to the cold freshwater is:

$$\text{150 MWh per year} = \frac{150000kWh}{365,25 \cdot 24} = 17,11kW \tag{5.111}$$

So the rate of thermal energy needed from the heat pump is:

$$\dot{Q}_{cond} = 17,11kW \tag{5.112}$$

By using values from table 5.4 and equation 2.38 with some rearranging I find the mass flow to be:

$$\dot{m} = \frac{\dot{Q}_{cond}}{(h_2 - h_3)} = \frac{17,11\frac{kJ}{s}}{433\frac{kJ}{kg} - 295\frac{kJ}{kg}} = 0,12\frac{kg}{s} \tag{5.113}$$

This process requires additional energy for the compressor in the cycle, I will now focus on calculating the rate of mechanical energy necessary to run this system with the working fluid R134a, and the potential losses in the process. The minimum work done by the compressor is:

$$\dot{W}_{min} = \frac{\dot{Q}_{cond}(T_{min,cond} - T_{max,evap})}{T_{min,cond}} = \frac{17,11kW(338,15K - 313,15K)}{338,15K} = 1,26kW \tag{5.114}$$

$\dot{W}_{min}$ is the work done in an ideal Carnot-process, this does not account for the over compression due to the loss in the condenser and evaporator. To find this loss I will have to calculate the work with the maximum condenser temperature, and the minimum evaporator temperature:

$$\dot{W}_{over} = \frac{\dot{Q}_{cond}(T_{max,cond} - T_{min,evap})}{T_{max,cond}} = \frac{17,11kW(340,65K - 310,65K)}{340,65K} = 1,50kW \tag{5.115}$$

The loss from the evaporator and condenser is then:

$$\text{Exchanger loss} = \dot{W}_{over} - \dot{W}_{min} = 1,50kW - 1,26kW = 0,24kW \tag{5.116}$$

The theoretical work done in this process is found by using the numerical values from table 5.4 for h_2 and h_1 in equation 2.40:

$$\dot{W}_{teo} = \dot{m}(h_2 - h_1) = 0,12\frac{kg}{s}(433\frac{kJ}{kg} - 419\frac{kJ}{kg}) = 1,68kW \tag{5.117}$$

The total loss is then:

$$\text{Loss total} = \dot{W}_{teo} - \dot{W}_{min} = 1,68kW - 1,26kW = 0,42kW \tag{5.118}$$

I now have both the total loss and the condenser/ evaporator loss, and their difference is then the process loss:

$$\text{Process loss} = \text{Loss total} - \text{Exchanger loss} = 0,42kW - 0,24kW = 0,18kW \tag{5.119}$$

The actual work done to the system by the compressor is:

$$\eta_{is} = \frac{\dot{W}_{teo}}{\dot{W}_{actual}} \tag{5.120}$$

When $\eta_{is} = 0,64$ the actual work done by the compressor is:

$$\dot{W}_{actual} = \frac{\dot{W}_{teo}}{\eta_{is}} = \frac{1,68kW}{0,64} = 2,625kW \approx 2,63kW \tag{5.121}$$

This gives a coefficient of performance of:

$$COP = \frac{\dot{Q}_{cond}}{\dot{W}_{actual}} = \frac{17,11kW}{2,63kW} = 6,50 \tag{5.122}$$

5.3 A summary of the results

Heat transfer rates	Numerical values
Heat central	5269,57 kW
Necessary energy for the building	127,57 kW
Water based heating system	127,57 kW
- Floors	100,86 kW
- Spot heating sources	26,71 kW
Spot heating system	127,57 kW
Ventilation heating system	133,14 kW
- With heat recycle	39,94 kW
Waste water heat recycling	17,11 kW
- Compressor	2,63 kW

Mass flow rates	Numerical values
Heat central	104,27 kg/s
Water based heating system	6,09 kg/s
- Floors	4,82 kg/s
- Spot heating sources	1,27 kg/s
Spot heating system	6,10kg/s
Ventilation heating system	6,37 kg/s
- With heat recycling	1,91 kg/s
Working fluid for heat pump	0,12 kg/s

Table 5.5: Numerical values for my calculations

Table 5.5 is the numerical values from my calculations for the heating system, and the mass flow rates for these systems.

5.4 Economical analysis

This section is the economical analysis for some of the systems and combinations, for the heating systems calculated and previously described in this chapter.

Assumptions

- Maximum prices are assumed.
- Prices include pipes.
- There is no change in maintenance cost.
- Assume consumption of electricity is 1.118.323.96 kWh for heating.

- Assume consumption of electricity is 150.000kWh. for water heating.
- Assume linear downpayment.

Elkem Salten verk has a electrical consumption of 1437,52MWh for Miljøbygget per year, and from this 1.118.323,96 kWh is used for heating. Equation 5.123 is the total cost of electricity used for heating today. I choose to set these costs as the maximum NOK saved in electricity per year, and as the maximum downpayment of investments per. year. The price of $0,60(\frac{NOK}{kWh})$ includes the network tariff of $0,30(\frac{NOK}{kWh})$ [28], and the constant set price over a year of $0,30(\frac{NOK}{kWh})$[4].

$$1.118.323,96(kWh) \cdot 0,60(\frac{NOK}{kWh}) = 670.994,4(NOK) \approx 671.000(NOK) \tag{5.123}$$

The downpayment time is then:

$$\text{years of downpayment} = \frac{\text{Investment cost}}{\text{Total price of electricity}} \tag{5.124}$$

Equation 5.124 is from [17] page 117. The water heating system is not part of the original electrical consumption used for heating, and the cost is calculated with equation 5.123 to be:

$$150.000(kWh) \cdot 0,60(\frac{NOK}{kWh}) = 90.000(NOK) \tag{5.125}$$

Product	Price range (NOK)
Heat central (200 kW)	150.000,-
Heat central (1000 kW)	250.000,-
Heat pumps	10.000,- - 20.000,- pr. kW
Control systems	10.000,- - 20.000 ,- pr. system
Water based floor heating	325,- per m^2
Ventilation system (Offices)	960,- per m^2
Ventilation system (Wardrobe)	1020,- per m^2
Spot Heat sources	4000,- per kW

Table 5.6: The numerical values for costs are from [23], [2], [9] and [21].

The maximum heat transfer rate is 1000 kW per heat exchanger, so here I would need 6 heat exchangers. For the heat central, the expected investment cost based on my assumptions is:

$$\text{Heat central (1000 kW)} \cdot 6 = 250.000NOK \cdot 6 = 1.500.000NOK \tag{5.126}$$

This is the estimated cost of the heat central without any connections for heat recycling. The downpayment for the heat central investment is:

$$\text{Years of downpayment} = \frac{1.500.000NOK}{671.000\frac{NOK}{Year}} = 2,23\text{Years} \tag{5.127}$$

For the water based heating system the total investment cost is:

$$\begin{aligned}
\text{investment} &= \text{Heat central (200 kW)} + \text{Spot Heat sources} \cdot 27kW && (5.128)\\
&+ \text{Water based floor heating} \cdot 2000m^2 && (5.129)\\
&= 150.000NOK + 4000\frac{NOK}{kW} \cdot 27kW && (5.130)\\
&+ 325NOK \cdot 2000m^2 && (5.131)\\
&= 908.000NOK && (5.132)
\end{aligned}$$

This is the estimated cost for the water based floor heating system without the heat central.

With the heat central the total investment is:

$$1.500.000NOK + 908.000NOK = 2.408.000NOK \tag{5.133}$$

For this heating system including the heat central the downpayment time is:

$$\text{Years of downpayment} = \frac{2.408.000NOK}{671.000\frac{NOK}{\text{Year}}} = 3,58\text{Years} \tag{5.134}$$

or seven years and roughly two months. Without the heat central the downpayment time is:

$$\text{Years of downpayment} = \frac{908.000NOK}{671.000\frac{NOK}{\text{Year}}} = 1,35\text{Years} \tag{5.135}$$

For the spot heating source system the investment cost is:

$$\begin{aligned}
\text{Investment} &= \text{Spot heat sources} \cdot 128kW + \text{Heat central (200kW)} && (5.136)\\
&= 128kW \cdot 4.000\frac{NOK}{kW} + 150.000NOK = 662.000\text{NOK} && (5.137)
\end{aligned}$$

The estimated downpayment time is then:

$$\frac{662.000NOK}{671.000\frac{NOK}{Year}} = 0,98\text{Years} \tag{5.138}$$

The estimated downpayment time with the heat central is:

$$\frac{662.000NOK + 1.500.000NOK}{671.000\frac{NOK}{Year}} = 3,22\text{Years} \tag{5.139}$$

For the ventilation system:

$$\text{Investment} = \text{heat central (200 kW)} + \text{Ventilation (office)} \cdot 1000m^2 \tag{5.140}$$
$$+ \text{Ventilation (Wardrobe)} \cdot 1000m^2 \tag{5.141}$$
$$= 150.000NOK + 960NOK \cdot 1000m^2 \tag{5.142}$$
$$+ 1020NOK \cdot 1000m^2 \tag{5.143}$$
$$= 150.000 + 960.000 + 1.020.000 = 2.130.000NOK \tag{5.144}$$

This is the estimated investment cost without the heat central. The downpayment time for this system with the heat central is:

$$\text{Years of downpayment} = \frac{2.130.000\text{NOK} + 1.500.000\text{NOK}}{671.000\frac{NOK}{Year}} = 5,40\text{Years} \tag{5.145}$$

Without the heat central the downpayment time is:

$$\text{Years of downpayment} = \frac{2.130.000NOK}{671.000\frac{NOK}{Year}} = 3,17\text{Years} \tag{5.146}$$

For the water heating system the investment cost is calculated to be:

$$\text{Investment} = \text{Heat pump} \cdot 18kW + \text{heat central (200kW)} \tag{5.147}$$
$$= 20.000\frac{NOK}{kW} \cdot 18kW + 150.000NOK \tag{5.148}$$
$$= 510.000NOK \tag{5.149}$$

The estimated downpayment time for the water heating system is:

$$\text{Years of downpayment} = \frac{510.000NOK}{90.000\frac{NOK}{Year}} = 5,67\text{Years} \tag{5.150}$$

The water heating system is only calculated as a system without the heat central, because the idea is for this system to be added to the other systems as an additional system. The downpayment time for water based surface heating system with the water heating system is:

$$\text{Years of downpayment} = \frac{908.000NOK + 510.000NOK}{671.000\frac{NOK}{Year} + 90.000\frac{NOK}{Year}} = 1,86\text{Years} \tag{5.151}$$

With the heat central:

$$\text{Years of downpayment} = \frac{1.500.000 NOK + 908.000 NOK + 510.000 NOK}{671.000 \frac{NOK}{Year} + 90.000 \frac{NOK}{Year}} = 3,83\text{Years} \tag{5.152}$$

The estimated downpayment time for the combined spot and water heating system without the heat central is:

$$\text{Years of downpayment} = \frac{512.000 NOK + 510.000 NOK}{671.000 \frac{NOK}{Year} + 45.000 \frac{NOK}{Year}} = 1,34\text{Years} \tag{5.153}$$

With the heat central

$$\text{Years of downpayment} = \frac{1.500.000 NOK + 512.000 NOK + 510.000 NOK}{671.000 \frac{NOK}{Year} + 90.000 \frac{NOK}{Year}} = 3,31\text{Years} \tag{5.154}$$

The estimated downpayment time for the combined ventilation and water heating system without the heat central is:

$$\text{Years of downpayment} = \frac{2.130.000 NOK + 510.000 NOK}{671.000 \frac{NOK}{Year} + 90.000 \frac{NOK}{Year}} = 3,46\text{Years} \tag{5.155}$$

With the heat central:

$$\text{Years of downpayment} = \frac{1.500.000 NOK + 2.130.000 NOK + 510.000 NOK}{671.000 \frac{NOK}{Year} + 90.000 \frac{NOK}{Year}} = 5,44\text{Years} \tag{5.156}$$

Chapter 6

Discussion

In this chapter I will discuss my results from the simulations and analysis chapters.

The initial energy balance is done with a basis in typical values provided by Elkem Salten. My assumptions builds upon these numerical values. All of the heat recycling systems are calculated with the assumption to be separate systems where one alternative is considered to provide the necessary thermal energy for heating. It is however a possibility to combine several different systems were each system is dimensioned for a specific environment.

Due to the changes in physical properties for air with changes in temperature. I use different temperatures in my calculations, which influences on the results of the calculations for the systems.

6.1 The initial energy balance

Due to lack of sufficient information about several parameters, I have had to make assumptions in my calculations. This creates some uncertainties for my results. I assume same pipe dimensions for the pipes inside and outside the factory building. This is done as a simplification for the calculation, and is a source of error for the initial energy balance.

For the heat exchangers the assumption for the overall heat transfer coefficient which I have set to be 1700 $\frac{W}{m^2 \cdot K}$ is the theoretical maximum value from table 11-1 [13], and could only be applicable when the heat exchangers are recently cleaned, and the water used is chemically cleaned. This is due to a rise in the fouling factor (R_f) from biological degradation for the heat exchanger with time, which will decrease the overall heat transfer coefficient. [13] The consequence of these assumptions is that in my simplified calculation the thermal energy obtained by the cooling system is larger then it should be, which gives a over dimensioned system. For the heat exchangers the consequence of using a larger overall heat transfer coefficient is that the area of the heat exchangers becomes smaller than it actually needs to be.

The energy balance for Miljøbygget is done with a basis in measured electrical consumption for this building. The energy balance has uncertainties from the total electrical consumption which and the building material used. The uncertainty from the electrical consumption is that portions of the total consumption with exception from lighting is unknown, and is therefore assumed values. The assumption regarding building materials is made, because the building has been built in three different building extensions over three separate construction-periods. The material used for this construction was infeasible to determine from construction sketches. In turn this makes the overall heat transfer coefficient for the building entirely based on my assumptions. When compared to the current building standard my overall heat transfer coefficient is 218,5% higher. Which contributes to the uncertainty for the heating system, and makes my simulations too unreliable to applicate directly. The risk is that the power necessary for heating for this building is calculated too high.

6.2 The heat central

The basis for the heat central is the initial energy balance. Here the rate of energy available is found to be 1,5 giga watts for one furnace, and the thermal energy currently utilized for energy recycling is about one third of this thermal energy rate (equation 5.4). I therefore try to make a heat central system which extracts one third of the energy available plus the additional energy needed for heating of Miljøbygget. The system must also be able to provide enough energy from each furnace independently. The heat central is connected to the cooling system through branch-pipes with separate valves into the heat centrals heat exchanger. This is done in order to provide continuous delivery of thermal energy when one or two furnaces are out of work, which increases the reliability of the system. In the calculations for the logarithmic temperature difference (equations 5.22- 5.28), the answer is incorrect in equation (5.22). My reason for using a incorrect answer is to create a difference in ΔT_1, which would be zero if the correct answer is used, and the logarithmic temperature difference would be undetermined due to $\ln(\frac{0}{0{,}01})$. My calculations for the heat central has one major unrealistic assumption where the heat exchanger efficiency is set to $(\eta = 1)$. This is unrealistic because a heat exchanger effectiveness of one, or 100% thermal energy transfer is the theoretical maximum value and is impossible to achieve. Based on this a loss of thermal energy from the heat exchanger is expected, and an increase in the temperature difference or flow rate is necessary to counteract some of the thermal energy loss in the heat exchanger.

6.3 Water based surface heating system

The idea for this system is to have two loops of hot water, one is the main heating loop which is to provide 100,86 kW of thermal energy and the other loop is for the spot heating sources in the wardrobe section of the building.The water based floor heating system is calculated with a basis in the energy balance for Miljøbygget, where the calculated rate of thermal energy needed for heating is 127,57 kW (4.88).The result from the calculations of the radiative contribution for the heat transfer coefficient above the floor, is calculated with an assumption of the linoleum on the floor to have the same emissivity as a black surface. I expect the error of this value to be in the region of ten to fifteen percent compared to similar materials such as rubber were the emissivity varies between 0,93 (hard) and 0,86 (soft) [12] table A-18. The spot heating loops have not been calculated any further, because the convectors, radiators, and radiating heat strips have individual differences regarding the ratio between convective and radiative thermal energy transfer. This ratio does also vary between manufacturers [29]. By assuming a maximum value for the rate of thermal energy needed. The excess energy which in my calculations are used for the spot heating sources, are set as a result of the set air temperature and the hot water temperature. This difference will decrease if one sets a higher air temperature resulting in a lower loss and therefore a lower heat flux. A source of error for this system is the maximum heat transfer set by the energy balance for this building, which results in a higher temperature in the wardrobe section then the comfort temperature. The higher temperature indicate that the energy balance could be over-dimensioned. Another source of error is found in the thermal resistance calculation, which is calculated with the heat transfer coefficient for linoleum, but is not calculated again with the heat transfer coefficient for tiles. The tiles is expected to be used in the wardrobe area, and has a slightly higher heat transfer coefficient than linoleum. The consequence of disregarding these changes is that the excess energy which forms the basis for the spot heating system is higher then it would be by taking these changes into account.

6.4 Spot heating source system

The spot heating source system is based on the initial energy balance for Miljøbygget. From my calculations I have found that the length of such a heating source to be 5,21m. This is surprisingly short. The main contributing factor for this length is the temperature difference between the inside of the pipe and the pipes outside temperature. If this temperature difference is decreased the loss for the pipe by conduction is reduced, which in turn increases the length of the spot heating source. In my simulation the pipe was assumed to have no wafers, the wafers will increase the surface of the pipe, and therefore increase the loss and decrease the length [12].

6.5 Ventilation heating system

The ventilations system is calculated as two separate systems where one is for the office section of the building, and the second system is for the wardrobe section of the building. This approach was necessary due to the different indoor temperatures for the two sections of the building due to different comfort temperatures for each section. The ventilation systems are dimensioned in my calculations for one hundred people which in this case might be a low number. The ventilations system is not based on the energy balance for Miljøbygget, but is calculated as a separate system. This results in a higher rate of thermal energy needed for heating when compared to the energy balance for Miljøbygget.

6.6 Wastewater heating

The estimated rate of thermal energy for the water-heating system is based on the remaining electrical consumption from the electrical bill. I have assumed that the energy needed for water-heating is 150MWh, which leaves 50 MWh for all the other electrically driven devices in the building (4.87). This assumption is unreliable due to a unknown electrical energy demand for the devices. The simulation is therefore a model of how to calculate such a system. Another source of error is the interpolation of enthalpies in the logarithmic pressure enthalpy chart. The interpolation are done by hand and are therefore estimations because of the scale of the chart. The isentropic efficiency is assumed to be 0,64, is made so the coefficient of performance is possible to calculate. According to [10] (figure 20), the value for the isentropic efficiency is found by interpolation of the pressure ratio between the condenser pressure and the evaporator pressure. The numerical value I have chosen for the isentropic efficiency is not of great importance because it would vary with the type of compressor chosen, and somewhat individual variations is expected.

6.7 Economics analysis

I have suggested several different solutions for utilization of thermal energy in Miljøbygget. In my economics analysis I have tried to calculate the costs of these investments, and the downpayment time for each of them. However labor is not included in my calculations. The consequences of this additional expense might be sever, and could impose a large contribution to my calculated downpayment time. The prices which this analysis builds upon is gathered from my sources and are not based on tenders, this is an uncertainty for the calculations. The network tariff used in the calculations is a average value for Nordland county, but is here an uncertainty due to Elkem Salten Verks private power grid. If Elkem Salten Verk is connected to the main power grid through their privately owned power grid, then the estimated network tariff might change, which in turn affects my calculations.

The cost of the heat central will contribute to the investment budget and, is not necessary for such a small expansion of the heat recycling system as Miljøbygget. For a future expansion of the heat recycling system, a investment in a heat central might be considered necessary. This of course is determined on where the expansions of systems should be and the energy rate needed. If the depot and mechanical garage are to be heated by thermal energy in addition to e.g. ice free roads, or to keep the work areas ice free. For these scenarios a heat central which extracts thermal energy from the cooling system will probably decrease the investment-cost due to savings in heat exchangers and pipes. The additional heat exchangers will also add to the lost thermal energy, so the investment might not be favorable. One could also argue that the downpayment time is not considerable, and the heat central is only unfavorable as an added investment cost when included in the spot heating system and the water heating system, which has the longest downpayment time. As I indicate in my calculations that investing in such systems is favorable for Elkem Salten.

Chapter 7

Conclusion

The Norwegian governments climate politics and goals to reduce the overall electricity consumption utilized for heating, are positive regarding a reduction of electricity consumption on a national scale. The smelting process produces vast amounts of excess heat. Some of this heat is currently being recycled and used for further economic enhancing purposes, while still vast amounts are released to the environment. The goal for my thesis was to create some suggestions to what this remaining thermal energy could be used for.

The initial energy balance is simplified and is affected by my limiting assumptions to where the thermal energy losses are located. I still mean it is a reliable energy balance as to where the major losses are, and is demonstrating an approach to calculate these.

The Heat central is calculated as an example of a heat central which is supplying the rest of the heating systems with thermal energy. I include a heat central into this system, in order to provide one point of energy extraction without physically removing water from the furnace cooling system. The heat centrals three connection-points is contributing to the reliability and stability with regard to the systems temperature changes. The reliability and stability is increased with the connection of all three furnaces versus connecting to two furnaces.

The surface heating system is calculated as a floor heating system utilizing thermal energy form the cooling system to provide indoor heating for Miljøbygget. This system gives the best temperature profile with regards to comfort [29], and does not have too long downpayment time.

The spot heating system was found to be a sufficient heating source, although its calculations are based on the initial energy balance, which limits the maximum heat transfer. This system is the heating system which have the shortest downpayment time of all heating systems. The temperature profile for this type of heating sources are not as good as the surface heating source, which makes this system less favorable.

The ventilation heating system is already installed in Miljøbygget, which possibly contributes to decrease the investment cost, if no upgrades except the heating elements are needed. This is a system which is vital for the indoor climate, and should therefore also be considered as a favorable heating source.

Water heating system I include a water heating system partially because such a system will further reduce the electrical consumption for Miljøbygget, and partially as an control system for the other heating systems. This system seems to have a short downpayment period, which makes it a favorable system to invest in. It is however expected to have a higher maintenance cost due to the compressor. This system is also the only system which needs energy added to the system in order to drive the process.

It is beneficial to install one, or a combination of the heating systems investigated in this thesis. The benefits and advantages of installing this system is in addition to a healthier economy, a positive climate effect is also expected.

The system I will recommend, is the ventilation heating system. Despite the ventilation heating system does not have the shortest downpayment time and the cheapest system to invest in. This system is vital for the indoor climate, and due to the already existing system the investment cost is expected to decrease, assuming some of the system elements could be kept as they are today. If this is not the case, a future rehabilitation of the ventilation system might be a opportunity for such an investment.

7.1 Future work

For future work the most critical part will be to reduce the uncertainties of all the calculations, and simulations. The use of a powerful simulation software might be recommended. The advantage is suspected to be further identifing important variables, and the calculation of these will be easier.

The thermal energy produced at Elkem Salten Verk holds possibilities for further utilization, and a larger expansion then presented in this thesis might be an option, such as providing heating for the public of Straumen.

Bibliography

[1] Numerical table of heat transfer coefficients. URL http://apollo.lsc.vsc.edu/classes/met455/notes/section6/2.html. Las sharked for availability 21-05-2014.

[2] Boligvarmes internett site. URL http://www.boligvarme.no/hva-er-vannb-ren-varme-/sp-rsm-l-og-svar.html. Last checked for availability 14-05-2014.

[3] Table with numerical values for thermal conductivity. URL http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html. Las sharked for availability 21-05-2014.

[4] Konkuransetilsynets internett site. URL http://www.konkurransetilsynet.no/no/kraftpriser/sjekk-kraftpriser/. Last checked for availability 14-05-2014.

[5] Forskrift om krav til byggverk (tek97). URL http://lovdata.no/dokument/SF/forskrift/1997-01-22-33?q=teknisk+byggeforskrift+10#KAPITTEL_8. Last checked for availability 14-05-2014.

[6] meld. st. 21 melding til stortinget: Norsk klimapolitikk, (2011 - 2012). URL http://www.regjeringen.no/nb/dep/kld/dok/regpubl/stmeld/2011-2012/meld-st-21-2011-2012.html?id=679374. Last checked for availability 14-05-2014.

[7] Arbeidstilsynet. Veiledning om klima og luft påarbeidsplassen, . URL http://www.arbeidstilsynet.no/binfil/download2.php?tid=79437. Last checked for availability 14-05-2014.

[8] Arbeidstilsynet. Arbeidstilsynets internet fact page about works hours and time., . URL http://www.arbeidstilsynet.no/fakta.html?tid=78151. Last checked for availability 14-05-2014.

[9] Ventinor AS. Pers. inf. Tom Erik Karlsen, 2014.

[10] Arne M Bredesen, Ola M Magnussen, Jostein Pettersen, Kåre Aflekt, and Munan Elgsæter. Fag tep 4255 varmepumpende prosesser og systemer termodynamisk grunnlag. Technical report, Norges Teknisk Naturvitenskapelige Universitet Institutt for energi og prosessteknikk, 2010.

[11] Statens byggtekniske état (BE). Veiledning om teknisk krav til byggverk kapittel 14 energi. URL `http://lovdata.no/dokument/SF/forskrift/2010-03-26-489`. Last checked for availability 14-05-2014.

[12] Afshin J. Çengel, Yanus A. Ghajar. *Heat and Mass Transfer: Fundamentals & Application*. McGraw Hill, fourth edition si units edition, 2011.

[13] Yanus A. Çengel. *Heat and mass transfer: a practical approach*. McGraw Hill, third si units edition, 2006.

[14] Elkem. Elkem.com. URL `http://www.elkem.com/en/news/item/New-NOK-250-million-furnace-at-Elkem-Salten-makes-operation-more-environme`. Last checked for availability 14-05-2014.

[15] Elkem, 2013. URL `http://www.elkem.com/en/Silicon-materials/About-us/Salten/`. Last checked for availability 14-05-2014.

[16] Elkem. Pers. inf. Nils Eivind Kamfjord, 2013.

[17] Kjell Gunnar Hoff. *Bedriftens Økonomi*. Universitetsforlaget, seventh edition edition, 2012.

[18] Jan-Olav Ingvaldsen. Sisoflor konkurs. *Avisa Nordland*, 2011. URL `http://www.an.no/nyheter/article5790479.ece`. Last checked for availability 14-05-2014.

[19] Nils E. Kamfjord, Edin H. Myrhaug, Halvard Tveit, and Bernd Wittgens. Energy balance of a 45mw (ferro-) silicon submerged arc furnace. pages 1–10, 2010.

[20] Norbert. Kilicarslan, Ali. Müller. A comparative study of water as a refrigerant with some current refrigerants. *Wiley InterScience*, pages 947–959, 2005. doi: 10.1002/er.1084.

[21] VVS Kupp. Price of a radiator, 2014. URL `http://www.vvskupp.no/Avdelinger/Produkter/Varmesystemer/Radiator/Purmo/Purmo_Hygiene/Purmo-Hygiene-H10-Radiator_-Enkel-Hyde-500-mm_-Lengde-2000-mm_-1092-W-328650120-p0000044724`. Last checked for availability 14-05-2014.

[22] Technical University of Denmark Department of Mechanical Engineering. Coolpack. URL `http://en.ipu.dk/Indhold/refrigeration-and-energy-technology/coolpack.aspx`.

[23] Mats Rosenberg. Kostnader for fjernvarmeutbygging. Technical report, Bioen AS, 2010.

[24] Statistisk sentralbyrå. Definition of man- years worked. URL `http://www.ssb.no/a/metadata/conceptvariable/vardok/2744/en`. Last checked for availability 14-05-2014.

[25] Sisomar. History of sisomar, 2013. URL `http://www.sisomar.no/page5.php`. Last checked for availability 14-05-2014.

[26] Jørn Stene and Ole Øiene Smedgård. Hensiktsmessige varme- og kjøleløsninger i bygninger. 2013.

[27] Trude Tokle and Jens Tønnesen. Inndeling av norge i klimasoner. Technical report, SINTEF energiforskning, 1999. The document is part of the series TR A4856.

[28] Norges vassdrags-og energidirektorat. Network tariff statistics nordland county. URL `http://www.nve.no/no/Kraftmarked/Nettleie1/Nettleiestatistikk/Nettleiestatistikk-naringskunder---2005-og-senere/`. Last checked for availability 14-05-2014.

[29] David Zijdemans. *Vannbaserte oppvarmings- og kjølesystemer*. Skarland Press AS, first edition, 2012.