Optimization of a District Heating Network
with the Focus on Heat Loss

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Challenge the future

Optimization of a District Heating Network

with the Focus on Heat Loss

by

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Abstract

In the Netherlands most houses and buildings are connected to a gas network since the nineteen seventies to heat our houses and cook. In march of this year the new Dutch government announced that to meet the climate accord of Paris all dutch household should be of the gas net in 2030. To do this multiple alternatives to heat homes are available and one of those which has proven to be feasible over the last years is district heat. In a district heating network hot water is pumped round a city through a network of pipe lines to houses and other buildings. The network is split into a primary and a secondary network. In the primary network the heat is transported from the source, and in the secondary network the heat is pumped from a substation to the customers. The heat usually is rest heat of a STEG- or waste incineration power plant. With the increasing demand for heat due to the disconnection of the gas network, and at the same time the search for a more sustainable heat source for the district heating network like geothermal heat the prices and costs rise rapidly. This makes the heat loss in the system a more important issue also from a financial perspective. The current installed networks are all series 1 single pipe lines of ST/PUR/PE, this means the inner pipe through which the water flows is steal, followed by an insulating PUR layer which is covered by a protective PE layer. To decrease the heat loss from the pipe lines Nuon is looking at two possible options. Series 2 single piping which has a thicker insulating PUR layer and second the twin pipeline where both the steal supply and return lines are embedded into one insulating PUR layer and a protective PE shell. The heat loss calculations for the buried pipelines in the ground are done using empirical formulas found in literature. These are checked numerical using a pde-solver. Comparing both results it was the concluded the empirical formulas give a good approximation of the heat loss. It also showed the heat loss is for a great deal depended on the temperature in the pipe and the ground temperature. In current heat loss calculations done by Nuon the temperature gradient of the water entering the network and leaving the network in not taken into consideration. By using the ambient temperature and a formula formulated in literature an estimation of the ground temperature at the buried depth of the pipelines can be made. By dividing the network into a number of pieces with length dx and an individual temperature T the temperature gradient over the entire network is taken into account. Collected data by Nuon concerning the user demand and the ambient temperature makes it possible to simulate what is the mass flow through the system at any given time and to analyze how the system responds to a demand fluctuation. By analyzing what happens in the networks in term of heat loss during a diurnal heat curve the results show that the heat loss stays nearly constant during a day. There are however big differences between different days in different times of the year. Analyzing what happens over the course of a year especially in the summer during times of very low demand the system under performs. A main reason for this is caused by the minimum required temperature of 70 degrees Celsius at the customers due to salmonella regulations. This causes a lot of extra mass flow of hot water pumped around the system which heats up the return flow. This is also clearly notable from the efficiency of the network which drops tremendously in the summer. Comparing the difference in heat loss for the series 1, series 2 and twin system the results where as expected an decrease in the amount of heat lost, by 14.6% for series 2 and 39.70% for twin compared to series 1. The overall image of what happens actually stays the same with high return temperature and relative high heat loss and low efficiency in summer. There are a few options to further improve the performance of the system with a few percent by changing the inlet temperature or the location of the bypass valve. Financially speaking the twin system is also the better choice of the three. Compared to series 1 for series 2 the investment costs will increase because the materials and instalment costs will increase. For the twin system the prices of the materials will increase and the placements of the welds will become more expansive however only half the number of pipe lines and joints is needed, so in total the prices will stay nearly the same compared to series 1. There is a lot of discussion on whether or not the maintenance costs will increase for the twin system, arguments for both cases are given. However the maintenance costs are so small compared to the costs of the heat loss that in every case the twin system is clearly the better choice.

Preface

The presented report is written to fulfill the final master thesis assignment that is part of the process and energy track of the master mechanical Engineering at the TU Delft. In this report you will find my findings on the research I did on Optimization of a District heating Network with the Focus on Heat Loss.

Over the last few months I was able to work on my master thesis project at Nuon Energy N.V. they assigned me with the question of what the advantages and disadvantages for using the series 2 and twin pipeline system would be as compared to the current used series 1. It was for me to determine from what perspective I would approach this problem, where I would lay the focus on and what methods I would use to answer the questions. I'm really thank full for Nuon they gave me the opportunity and that they gave me this freedom for me to do useful research for them and also ensure I could do work on my master thesis. I would like to say special thanks to Stephan Bunnik from Nuon for being my daily supervisor during the time I was working at Nuon.

I would like to thank Prof. dr. ir. Bendiks Jan Boersma for being my supervisor at the TU delft, and the rest of the committee for being part of this.

T.J. de Boer
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Nomenclature

δT	Temperature change
$\dot{m}$	Mass flow
ν	Viscosity
c	Distance between return and supply flow pipes
Cp	Heat capacity
d_1	Diameter of steel pipe
d_2	Diameter of PUR layer
d_3	Diameter of PE layer
dt	Time step
dx	Length delta x of a section of the pipe
H	Depth at which the pipe is buried
h	Heat transmission
k_1	Conductivity of steal pipe
k_2	Conductivity of PUR layer
k_3	Conductivity of PE layer
k_g	Conductivity of steal pipe
Q_n	Heat in the system
Q_{in}	Heat flow in section
q_{loss}	Heat loss
Q_{out}	Heat flow out section
R	Heat resistance
Re	Reynolds number
T	Temperature
T_0	Temperature
T_1	Temperature
T_2	Temperature
T_a	Ambient temperature
T_g	Ground temperature
T_r	Return temperature
T_s	Supply temperature
T_{in}	Inlet temperature
T_{out}	Outlet temperature
v	Flow speed

1
Introduction

A district heating network is a network pipelines that delivers hot water to homes and businesses get from a centralized heat source, instead of using an individual heat source such as a boiler or a heat pump. Residual heat from a power plant or a garbage disposal plant or any other type of heat source is pumped around the city to heat up houses and building or heat tap water. The district heat network is divided into two sections: the primary or transport network and the secondary or distribution network. In the primary network the heat is transported from the heat source to the substations or heat transfer station(HTS). In the HTS the heat is transferred to the secondary network through a set of heat exchangers and pumps. From the HTS the hot water is transported to the customers. The heat source usually is rest heat from the industry such as power plants and garbage disposal plants. These heat sources are arguably unsustainable heat sources. This is why Nuon Energy N.V. is also looking for more sustainable sources of heat like cooling heat from data centres or geothermal heat. Because the heat sources are mostly located far from the customers, household and office buildings, the heat is first pumped in the primary network of pipes at a temperature in the range of 100 - 130 degrees Celsius. In the heat transfer stations (HTS) the heat is transferred to a lower temperature heat net, usually this secondary net is a 70 - 40 net. This means the supply temperature to the users is 70 degrees Celsius and the return temperature is 40 degrees Celsius. The heat is always delivered by heat exchange, either through the customers central heating net (radiators or underfloor heating) or for tap-water through a heat exchanger. Assuming there are no leakages this means there is always a return flow with the same mass flow as the supply flow, this also means there is always a supply and return pipeline installed.

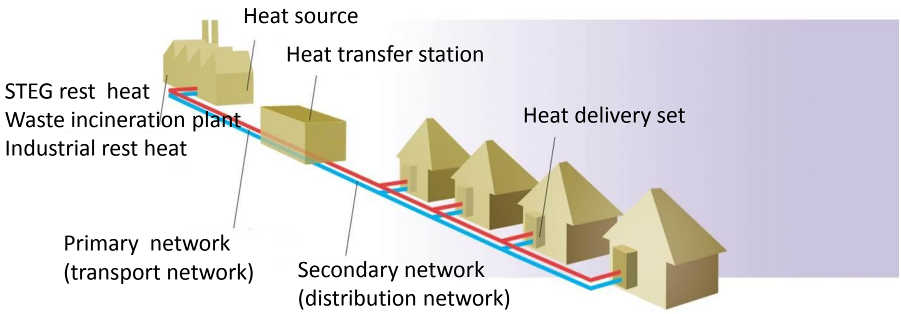

Figure 1.1: Main components in a district heating network

Since heat loss is becoming more of an issue for sustainability reasons and because of the increasing cost for the heat that feeds the system it is important to minimize the heat loss. A possible way to do this is to replace the currently used Series 1 pipelines by the series 2, which has thicker insulation, or the twin pipeline system where the supply and return pipe are embedded into one insulating layer and protective PE layer. However current calculations on the heat loss is the system are all based on ideal conditions. An analysis of what happens in a system in operation is never done by Nuon. Before the question of how much heat loss can be saved by using the series 2 or twin system can be answered a detailed analysis of the performance of a district heating system in operation mode is done.

1.1. Design of a secondary district heating network

The secondary district heating network can be split into three parts. First the HTS where the heat is transferred from the primary net to the distribution network. Second the distribution net which consists of a pipeline network through which the hot water is pumped towards the customers. And lastly a supply set at the customers where the heat is delivered to either the central heating or the tap-water system.

1.1.1. piping

The piping comes in a range of predefined diameters produced by Logstor. These are referred to as the nominal diameters or DN size. The secondary network is build up containing a number pipelines in different sizes ranging from DN125 to DN25, starting with the widest one DN125 at the HTS and endings with the smallest DN25 at the users. All the DN sizes are ST/PUR/PE insulated. This means the inner pipe, containing the supply or return water is made of steal. The second layer is a insulating PUR foam layer. The last layer is a plastic cover layer to protect the PUR. The current networks are all installed using series 1, this means the piping always comes in pairs 1.2. One hot water for the supply and one cold water for the return meaning quite a lot of space is needed in the ground for the heat net.

Description

A preinsulated pipe consists of:

Pos.	Part	Material
1	Service pipe	Steel
2	Insulation	Polyurethane foam
3	Outer casing	Polyethylene, PE-HD
4	Two 1.5 mm² copper wires for surveillance	One wire is tinned
5	Pipe label	

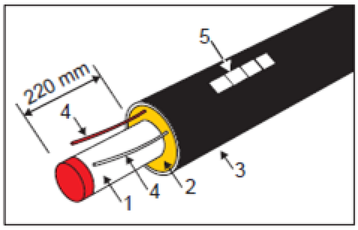

Figure 1.2: Single pipe

The single piping series 2 system has a thicker insulating PUR layer so automatically the outer diameter of the PE layer increases. The steal sizes are identical to those of series 1. For the twin piping the steal dimensions are also the same however both the supply and return temperature are within a common PE and PUR shell 2.2. The dimensions of the single piping series 1 and series 2 and the twin are found in appendix A.

Description

A preinsulated TwinPipe consists of:

Pos.	Part	Material
1	Service pipe	Steel
2	Insulation	Polyurethane foam
3	Outer casing	Polyethylene, PE-HD
4	Alarm wire for surveillance	Copper (one is tinned)
5	Pipe label	

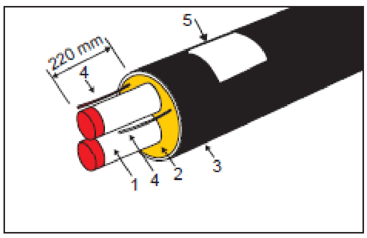

Figure 1.3: Twin pipe

1.1.2. Heat transfer station

The heat transfer station (HTS) is a room located in or near the residential area. In the HTS the heat is transferred from the primary to the secondary network. The main components in a HTS are a flat plate heat exchanger, pumps and some control valves, an overview is shown in appendix: B. The secondary network usually is a branched out piping network to transport the hot water to all the users in the net. The primary and secondary networks are both independent closed systems meaning there is always a return flow of water. In the HTS the heat is transferred from the primary to the secondary network through a flat plate heat exchanger. The water is pumped round in the secondary network by the two pumps in the HTS. The sizes of the heat exchanger and pumps are dependent on the amount and capacity of the users in the secondary net. Usually this is somewhere between the 200 and 600 users with an installed capacity of 8 to 22 kW depending on the size and demand of the user.

1.1.3. Heat delivery set

The users all have a supply set in their homes. The water is delivered to the users with a minimum temperature of 70 degrees Celsius from here the hot water goes either directly into the central heating system of the user or through a heat exchanger to heat the tap-water. Depending on the capacity there is an maximum mass flow of water that can flow through the set. To heat the tap water the hot water always goes through a heat exchanger this is also the reason the temperature delivered to consumers should always be 70 degrees Celsius, this is because of salmonella regulations. In times of no, or very low, heat demand the temperature of the water will cool below that 70 degrees. In order to keep the system at temperature a bypass is installed in the form of a temperature regulated valve, this is to ensure the water keeps flowing and stays at temperature.

1.2. Scope of the thesis

The scope of this project will be the heat losses in the secondary pipeline network under different conditions and settings. The heat loss in the HTS, the supply set, and the performance of the heat exchangers and pumps are not taken into consideration. Also during operation other types of heat losses can accrue such as heat losses caused by leakages or illegal tapping, these will all not be taken into consideration.

2

Steady state heat loss calculations

At first a solution to find the heat loss in the form of a steady state calculation is presented. This is done with solutions based on explicit formulas found in literature which give an approximation for the heat loss of a buried district heating pipe in W/m. Secondly the explicit formulae are checked for accuracy by a numerical evaluation. These calculation are done with a numerical partial differential equation (pde) solution using the pde-solver in MATLAB.

2.1. Empirical heat loss calculations

The steady state heat loss of a district heating pipe system in the ground is analyzed. The empirical formulas for heat loss calculation in the pipes are based on the literature by Petter Wallentén [1] and Benny Böhm [2].

2.1.1. Steady state heat flux

The system is assumed to run continuous over time. So all the heat will settle in the ground over time, assuming the only heat loss is caused by the convection and radiation from the ground to the air. This is why the heat loss can be assumed to be steady state. The temperature fluctuation is not high enough to take the transient heat loss in consideration. Also the temperature in the pipe is assumed uniform in a cross section: no laminar heat flux profile will form in the water. This can also be concluded from the fact that turbulent flow is assumed which follows from the calculations on the Reynolds number from pipe flow.

$$Re = \frac{v * D}{\nu} \tag{2.1}$$

The Reynold number is dependent on the flow velocity of the water(v), the diameter of the pipe(D) and the viscosity of the water(ν), equation: 2.1. The diameter is depending on the DN-size of the pipe. The viscosity is depended on the temperature and pressure of the water, which varies throughout the entire network. The flow speed is depending on the demand of the individual users. This means many unique Reynolds numbers are found in the system. The flow is assumed laminar when the Reynolds number is below 2100 [3]. When the total demand is assumed at max capacity of the system and is distributed equally over all the users and the temperature in the entire system is set at 70 degrees the Reynolds number is above 2100 everywhere. At 40% of maximum mass flow the first pipe section drops below 2100, a DN32 section. At 10% all of the DN25 and still only the one DN32 section have a Reynolds number below 2100. However in reality the demand won't be distributed equally over all the users. This will mean that in some sections the flow will be zero while in others the flow will be higher at any given time, this is especially true for the DN25 pipes which are at the end of a string connecting the user. This means it is save to assume the flow is always turbulent.

2.1.2. Heat Capacity of Water and conduction factors of materials

Although the heat capacity of water is temperature dependent in this thesis and in the model, because of calculation time the Cp of water is assumed constant and set at a value of 4180 J/kgK. The conduction factors used are in table 2.1.

Material	Conductivity [W/m*K]	Variable
Steel (ST)	43	k_1
PUR	0.03	k_2
PE	0.33	k_3
Ground	1.6	k_g

Table 2.1: Heat conductivity of different Materials

All these values for conductivity are assumed constant so independent of the temperature. Also the conductivity for PUR increases over time [4], this is not taken in account but the live-time average is taken. The conductivity for the ground or soil is influenced a lot by all different factors depending on the type of soil and for example moist level, this will be discussed in chapter 5. The most common value in the Netherlands is $1.6W/(m*K)$.

2.2. Heat loss from a single insulated pipe

The formula for the heat loss of a single pipe in any medium is given by equation 2.2 [5].

$$q = 2*\pi*L*k*(T_1 - T_2)/ln(r_2/r_1). \tag{2.2}$$

2.2.1. Single pipe in the ground

The empirical formulae of Petter Wellenten also take in account the effect of the ground and the effect of the interaction between the supply and return heat.

The heat loss is a function of the temperature difference and the thermal conductivity of the ground. The heat loss is given by 2.3 in [W/m]. The variables are defined showed in figure **??**. The dimensions are depending on the DN-size given in appendix **??**.

$$q = 2*\pi*k_g*(T_1 - T_0)*h_1(H/r_4, \mathrm{B}). \tag{2.3}$$

For the dimensionless heat loss factor h_1 we get:

$$h_1^-1 = ln(\frac{2H}{r_3}) + \mathrm{B}. \tag{2.4}$$

The dimensionless parameter B is given as:

$$\mathrm{B} = k_g*(\frac{1}{k_1}ln(\frac{r_0}{r_1}) + \frac{1}{k_2}ln(\frac{r_1}{r_2}) + \frac{1}{k_3}ln(\frac{r_3}{r_2})). \tag{2.5}$$

2.2.2. Two pipes in the ground

For two pipes in the ground an extra term is added for the influence the two pipes have on each-other. Two formulas for the heat loss of buried district heating pipes are mentioned in literature [1] and [2]. The formula given by Wellenten:

$$q = 4*\pi*k_g*((\frac{T_1 + T_2}{2} - T_0)*h_1(H/r_4, D/r_4, \mathrm{B}). \tag{2.6}$$

T_1 and T_2 are defined as:

$$T_1 = \frac{T_s + T_r}{2}. \tag{2.7}$$

$$T_2 = \frac{T_s - T_r}{2}. \tag{2.8}$$

and

$$h_1^-1 = ln(\frac{2H}{r_3}) + \mathrm{B} + ln(\sqrt{1 + (\frac{H}{D})^2}). \tag{2.9}$$

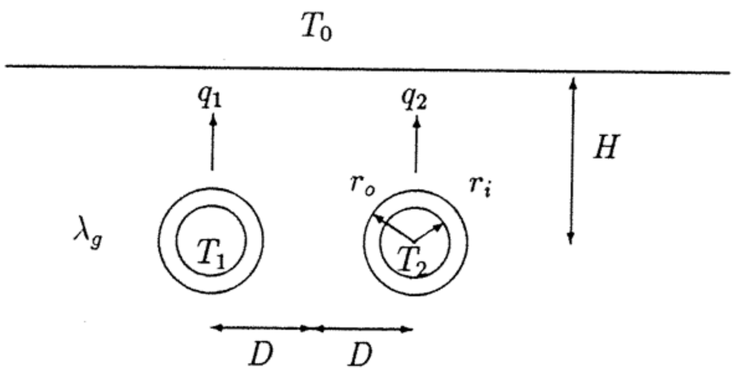

Figure 2.1: Two pipes in the ground.

Or the formula as given by Böhm:

$$q_{loss} = \Delta T/R. \tag{2.10}$$

$$q_{Loss} = \frac{(T_{supply} - T_{ground}) * (R_{gr} + R_{pipe}) - (T_{return} - T_{ground}) * R_{parallelpipes}}{(R_{groundcover} + R_{pipe}) * (R_{groundcover} + R_{pipe}) - R^2_{parallelpipes}}. \tag{2.11}$$

$$1/R = \sum_{i=1}^{n} (1/R_1 + 1/R_2 + 1/R_3 + ... + 1/R_n). \tag{2.12}$$

$$1/R_{pipe} = \frac{2\pi}{\frac{1}{k_a} ln\frac{r_2}{r_1} + \frac{1}{k_b} ln\frac{r_3}{r_2}}. \tag{2.13}$$

$$1/R_{groundcover} = \frac{2\pi k_{ground}}{ln\frac{2H}{r_4}}. \tag{2.14}$$

$$1/R_{parallelpipes} \frac{4\pi k_{ground}}{ln(1 + (\frac{2H}{C})^2)}. \tag{2.15}$$

2.2.3. Twin pipe in the ground

The empirical formula for the heat loss of the twin-pipe system in the ground is also given by Petter Wallentén.

$$q_1 = q_s + q_a \tag{2.16}$$

$$q_1 = q_s - q_a \tag{2.17}$$

$$q_s = (T_s - T_c) * 2\pi\lambda_i * h_s(r_i/r_c, D/r_c) \tag{2.18}$$

$$q_a = T_a * 2\pi\lambda_i * h_a(r_i/r_c, D/r_c) \tag{2.19}$$

$$h_s^- 1 = ln(\frac{r_c^2}{2Dr_i}) - ln(\frac{r_c^4}{r_c^4 - D^4}) \tag{2.20}$$

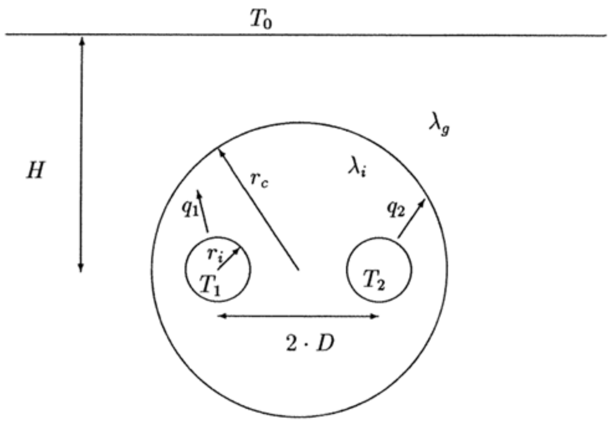

Figure 2.2: Two pipes embedded in one circular insulation.

$$h_a^- 1 = ln(\frac{2D}{r_i}) - ln(\frac{r_c^2 + D^2}{r_c^2 - D^2}) \tag{2.21}$$

The other parameters used in these equations are all size related. The can be found in the figures 2.1 and 2.2. Other dimensions like how deep the pipe buried and the distance between the pipes are given in appendix C. The difference between series 1 and series 2 piping is the size of the insulating PUR layer. Assuming a supply temperature of 70 degrees Celsius supply and 40 degrees Celsius return temperature the heat losses for the different sizes of piping used in an average district heating distribution network are given in figure 2.3. There is a decrease in heat loss as expected from series 1 to 2 and even more for twin. Also there is a small difference between the calculations as done by the formulas of Wallentén and Böhm

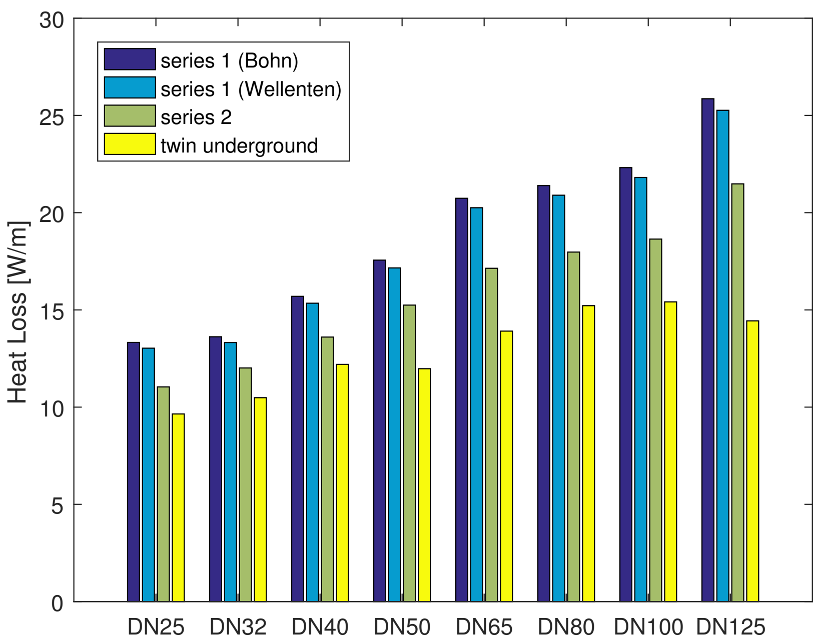

Figure 2.3: Heat loss in W/m for each DN-size.

2.3. Numerical solution

Besides an empirical solution the heat loss of the pipeline will be analyzed using a numerical model to validate the empirical solution. The results will be compared in this next section. To find a numerical solution for this problem the side view is divided in a grid, each section of the mesh is then given a temperature so a partial differential equation (pde) can be used to determine the heat flux between neighboring grid sections. There are different software packages that are used to solve problems like this. In this case the pde-solver toolbox of Matlab is used. The same assumptions can be done as in the previous part so a steady state is assumed. This means the net heat flux in the ground will approach zero and the only heat loss is from convection and radiation form the ground to the air. Also the flow in the pipe is assumed fully developed and the temperature of the temperature of the water is uniform distributed over a cross section.

2.3.1. Model and boundary conditions.

The model is a grid of 5000 mm by 2500 mm 2.4. This is to be the limit at which the ground temperature is influenced by the heat from the pipes [6]. A too large model will cost to much computing power especially since the sizes of the mesh need to be small enough so the different layers of the insulated pipes can be defined. At these boundaries the temperature of the ground is assumed to be no longer influenced by the heat from the pipe meaning that the heat flux at these boundaries is set to zero. This also means the only heat flux in the system is at the boundary between the ground and the air. This heat flux is combined heat radiation and convection. The last boundary conditions are set at the inside of the pipelines. These are set at a constant temperature.

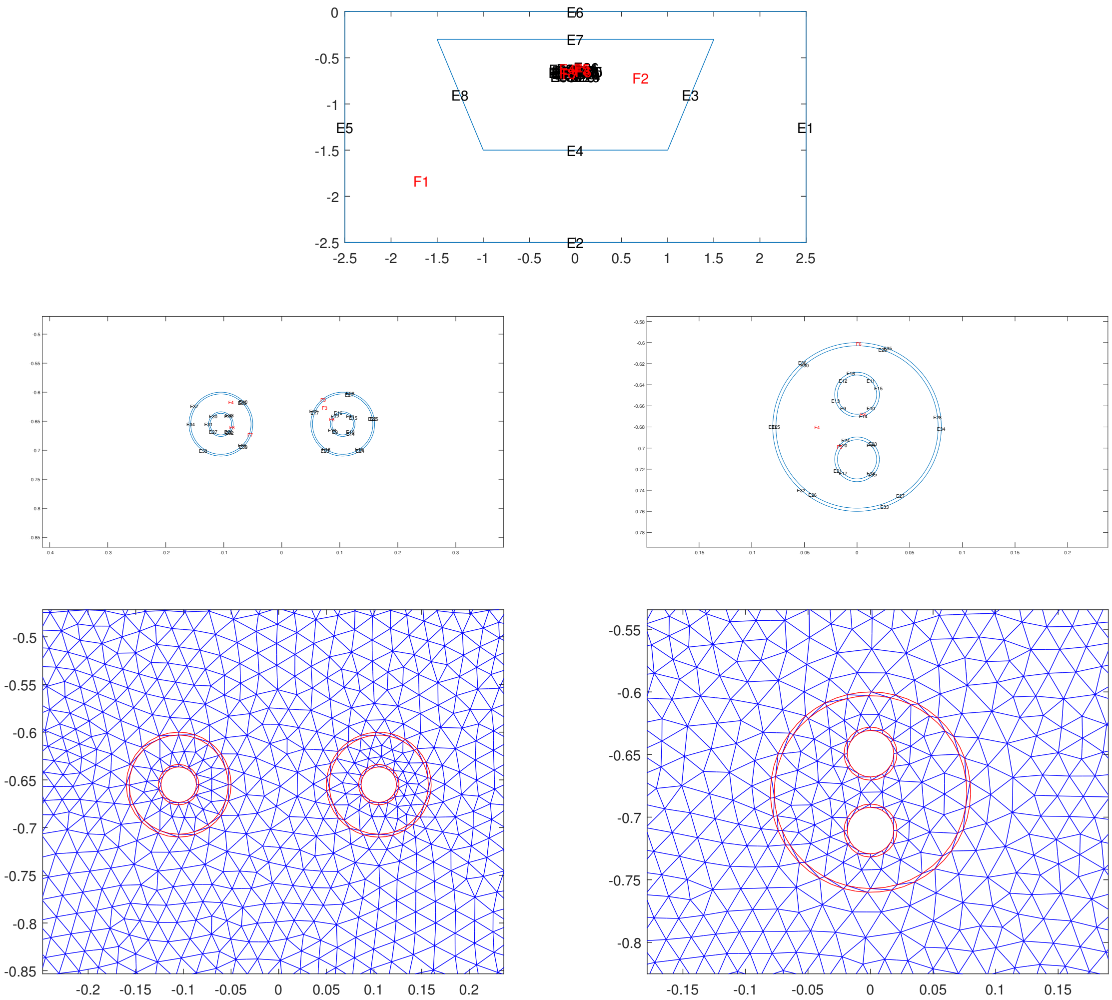

Figure 2.4: Rectangular model of the ground mesh (top) with the single piping (middle left) and the twin piping (middle right). Mesh of the single piping in the ground (left) and twin (right).

2.3.2. Heat transmission form the ground to the air

The total heat transmission form the ground to the air is combined radiation and convection. The main part of the ground covering the pipelines in the secondary network will be pavement in the form of bricks or asphalt. The total heat convection is usually divided in forced and natural convection. Both depending on the ambient and groudn temperature. Natural convection is caused by the difference in temperature between the ground and the air causing a flow of rising air. The forced convection is caused by the wind blowing over the ground. One of the two will be dominant and the other one can be neglected. This makes it very hard to predict the convection. For radiation too, there are to much variables to calculated the exact heat flux from the ground to the air because it is depended on too much variables: outside Temperature, Sun intensity, amount of shade, wind speed, type of surface(Soil, grass, paving, etc.), so it is assumed that the combined heat flux for radiation and convection of the ground to the air is 14.6 $W/(m*K)$ [7].

2.3.3. Calculations

Now that all the boundary conditions are set the pde-solver will look for a solution at which the temperature gradient of all the individual mesh elements are nearly constant, or the net heat flux through a mesh part is nearly zero. This is called the steady-state solution. A plot of the temperature distribution at steady state is shown in figure 2.5. At the steady state there will be heat flux through the top boundary, from the ground to the air. This is the total heat loss from the supply and return pipe. These calculations are done a couple of times varying the ground temperature, the supply temperature and the ground conductivity.

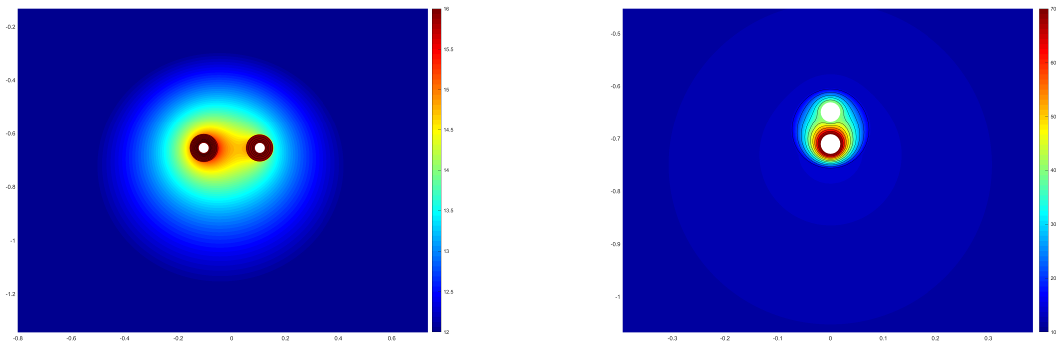

Figure 2.5: Temperature field in a single pipe (left), and twin (right). Temperature supply-return-ground: 70-40-10 C.

2.4. Results

The total heat loss of the two pipes given as the heat flux between the ground and the air are plotted against the calculated heat loss. The results are plotted in figure 2.6. Result for the series 2 and twin can be found in appendix D. The difference between the numerical and empirical results increases for the twin system. This is possibly caused by how the boundary layers in the system are defined. The temperature for both the supply and return pipe are fixed in this model while in reality, especially for the twin system where the supply pipe an return pipe are close together, a small part of the heat from the supply pipe is heating the return pipe. In [7] this is solved by setting part of the boundary condition of the return pipe as a negative heat flux calculated using the heat loss formula by Wallentén, doing this the results of the numerical and empirical solution show great resemblance.

2.5. Conclusion

In this section the results of the different approaches for heat loss are compared and will be evaluated. We can conclude the empirical formula and the partial differential solver show great resemblance. According to the numerical calcualtions done the empirical formula's given by Petter Wallentén give a pretty good approximation of what the heat loss is for buried pipes. The heat loss is greatly influenced by the environment conditions, the empirical and numerical show the same trend. The difference between the analytic and numerical solution is minimal for the series 1 and 2 piping system. In case of the twin-pipe the difference becomes a little bigger. Besides the effect of the pipes and their corresponding dimensions the amount of heat loss is depended on three more variables: the ground conductivity, the ground temperature and the temperature of the system. We can conclude that the heat loss is much heavier depending on the ground temperature and the system temperature then the heat conduction of the ground. At least for the range of heat conduction values in which the system normally operates, varying between $1 - 2W/mK$ [6], the effect on the heat loss is minimal 2.6. The ground temperature and the temperature in in the pipe of the water are of big influence on the heat loss so it should be taken into account.

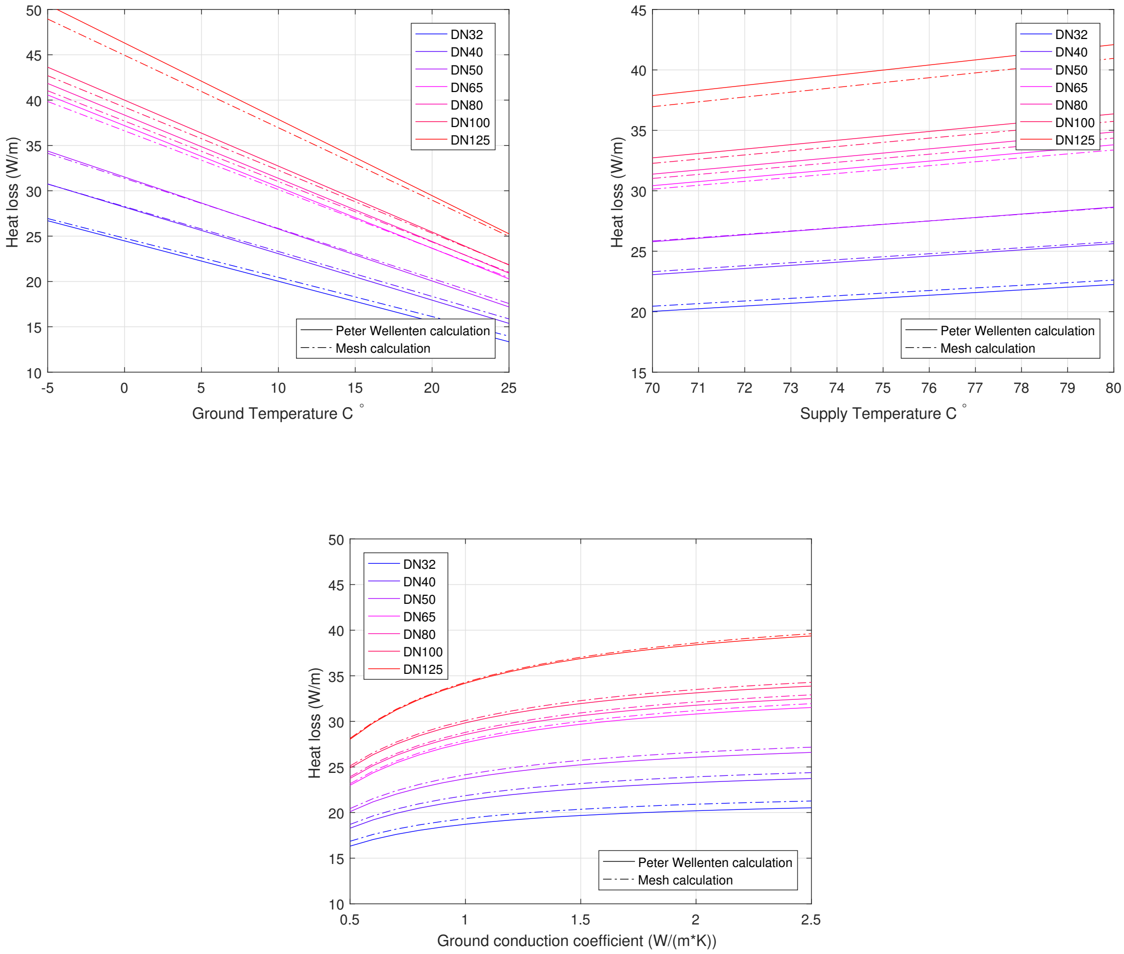

Figure 2.6: Difference between the numerical and empirical heat loss calculations varying: ground temperature (top left), supply temperature(top right) and ground conductivity (bottom).

3
Model

For the current heat loss calculations done by Nuon it is assumed the supply and return pipes are a uniform temperature, meaning the temperature at the beginning of the system is equal to that at the users. In reality there is a temperature difference beteen the beginning and end of the pipeline. For a random piece of pipeline in the system the change in temperature δT is dependent on the heat of the water flowing in, minus the heat of the water flowing out, minus the heat loss trough the wall of the pipe. Also the demand is assumed constant meaning the flow is constant. To make a more accurate guess of how much heat is lost and to be able to simulate what happens in part load, a model is introduced which solves this problem and enables us to model a distribution net over time.

3.1. Heat loss of hot water flowing through a pipe

When water flows through a pipe, due to the heat loss, a temperature profile will occur with warmer water at the inlet and colder water at the outlet. From the previous chapter we concluded that the heat loss is depended on the temperature of the water so in order to calculate the heat loss over the entire system in a precise way the pipe has to be divided in sections with length dx with a uniform temperature T. In theory this is impossible unless we take dx infinitely small or with the size of one molecule of water. In reality we take into account the calculation speed, the dimension and the accuracy of the entire network to determine the size of dx. The length of dx determines the number of sections n in which the entire network is divided. A section n of the pipe is modelled in figure 3.1.

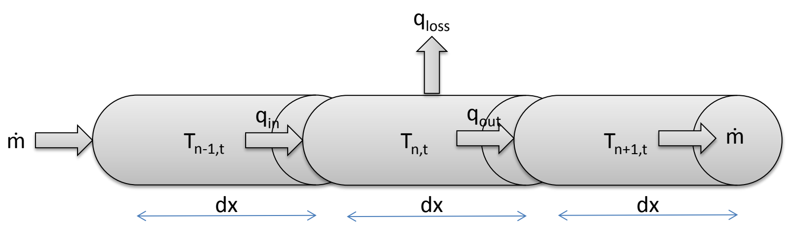

Figure 3.1: Heat loss in a pipe section with length dx.

For the first part of the analysis, which is the daily analysis, a dynamic model is used meaning the temperature in the entire system can changes over time. For a random section of the pipe n with length dx at any given t, the energy balance is given by the equation 3.1.

$$Q_{n,t} = Q_{n,t-1} + \delta Q \tag{3.1}$$

At any given time t the mass flow in and out of a section of the pipe with length dx is equal. This means the Qin and $Qout$ is only depended on the temperature and the mass that follows from the heat demand. $Qloss$ is only depended on the temperature. The change of heat in a given section n if the pipe is then given by 3.2.

$$\delta Q = Qin - Qout - QLoss \tag{3.2}$$

$$Q_{n,t+1} = Q_{n,t} + Qin_{n-1,t} - Qout_{n,t} - Qloss_{n,t} \tag{3.3}$$

$$Q_{n,t} = (T_{n,t} - T_ref) * m_{n,t} * c_p \tag{3.4}$$

The heat flow from the section $n-1$ to n and from n to $n+1$ are defined as:

$$\dot{Q}in_{n,t} = (T_{n-1,t} - T_{ref}) * \dot{m}_{n,t} * c_p \tag{3.5}$$

$$\dot{Q}out_{n,t} = (T_{n,t} - T_{ref}) * \dot{m}_{n,t} * c_p \tag{3.6}$$

The equations for the heat loss q_{loss} function given in the previous chapter has units (W/m) so it needs to be multiplied by dx. Everything added together we get:

$$Q_{n,t}(m, T_{n,t+1}) = Q_{n,t-1}(m, T_{n,t}) + Qin_{n,t}(\dot{m}, T_{n-1,t}) - Qout_{n,t}(\dot{m}, T_{n,t}) - q_{loss}(m, T_{n,t}) * dx \tag{3.7}$$

When this is done for all pieces dx in which the entire net is divided the temperature and heat loss of each specific peace of pipe at a specific time can be calculated in a much more accurate way which will give a more accurate value for the heat loss of the entire system. However to do this for every piece with length dx for an entire net will take a long time even with a computer. For this reason the state-space calculation method is introduced.

3.2. state-space calculations

The equation 3.7 can be written in the state-space form. This means the heat loss calculations for all the pieces dx of the entire system can be done at once by multiplying an array with all the temperatures $T_{n,t}$ with a state-space matrix A. The state space form is given in equation 3.9. In which Ts is the supply temperature, the heat loss (2.11) is also depended on the return temperature Tr. This is why there is also a state-space matrix B.

$$\delta T_n = Q_n / (m_n * c_p) \tag{3.8}$$

$$Q_{n,t+1} = A * Ts_t - B(n) * Tr_t \tag{3.9}$$

in which T_s is an array of all the temperatures in the system and A is the state space matrix.

$$A = \begin{bmatrix} a_{1,1} & a_{1,2} & \cdots & a_{1,n} \\ a_{2,1} & a_{2,2} & \cdots & a_{2,n} \\ \vdots & \vdots & \ddots & \vdots \\ a_{m,1} & a_{m,2} & \cdots & a_{m,n} \end{bmatrix}$$

Now the calculations for the entire system can be done at ones for a time step dt.

$$\begin{bmatrix} T_{1,t+1} \\ T_{2,t+1} \\ T_{n,t+1} \end{bmatrix} = \left(\begin{bmatrix} m_1 & - & - \\ \dot{m} * dt & m_2 - \dot{m} * dt & - \\ - & \dot{m} * dt & m_n - \dot{m} * dt \end{bmatrix} \times \begin{bmatrix} T_{1,t} \\ T_{2,t} \\ T_{n,t} \end{bmatrix} * c_p - \begin{bmatrix} qloss_{1,t} \\ qloss_{2,t} \\ qloss_{n,t} \end{bmatrix} \right) / \left(\begin{bmatrix} m_1 \\ m_2 \\ m_n \end{bmatrix} * c_p \right) \tag{3.10}$$

3.2.1. Size of dx and time step dt

The length of a pipe sections dx is chosen as 0.1 meter, the smaller the chose dx the better the accuracy because each part is simulated at constant temperature, while in reality there is a temperature slope in the length. A too small dx will decrease the speed of the calculations. With 0.1m the total system in the chosen case network contains 30277 units. The dt is depended on the mass flow through a section dx in the time dt. A too big dt will make the system numerical in stable. This instability occurs when the mass flowing through a piece of pipe with length dx in the time dt is bigger than the mass of the water in that section.

3.2.2. Model

The input of the model is the temperature array for which the first value is the inlet temperature in the HTS. Depending on the demand, the mass flow in the system and the time step is determined and the A-matrix is made. The heat loss is calculated as a function of the temperature. Then the state space calculation is done to determine the temperature array for the next time step. This is continued over time until either one of two things happen. The demand changes so the mass flow changes and a new A-matrix needs to be formed or the temperature at one of the users drops below the minimum required temperature which means the mass flow also needs to increase and a new A-matrix needs to be formed **??**. This way either a simulations of a time dependent demand curve can be done or a steady-state solution with the required total mass flow to get the minimum required temperature of 70 degrees C can be calculated. For a daily curve the first will be done while to simulate what happens over a year the second will be done because it would take to long to do a real time simulation for an entire year.

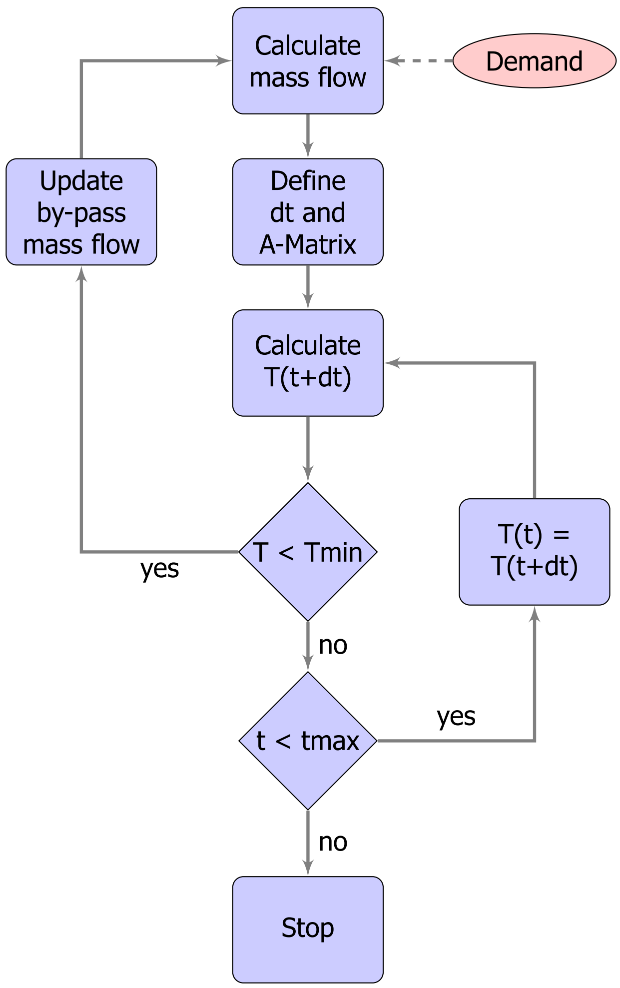

Figure 3.2: Flow chart of the Matlab model.

4

District Heating Network

The network that will be analyzed for heat loss in the heat distribution system is the AGH12 network. This is a averaged sized and designed network of Nuon Energy N.V. located in the neighborhood of Schuytgraaf in Arnhem. It covers 247 connection in the form of single family homes and some apartments. The average capacity is 12.2 kW. Near the centre of the piping network the HTS from where the hot water is pumped through the pipeline network to all the users. The layout of the network of pipes is shown in figure 4.1.

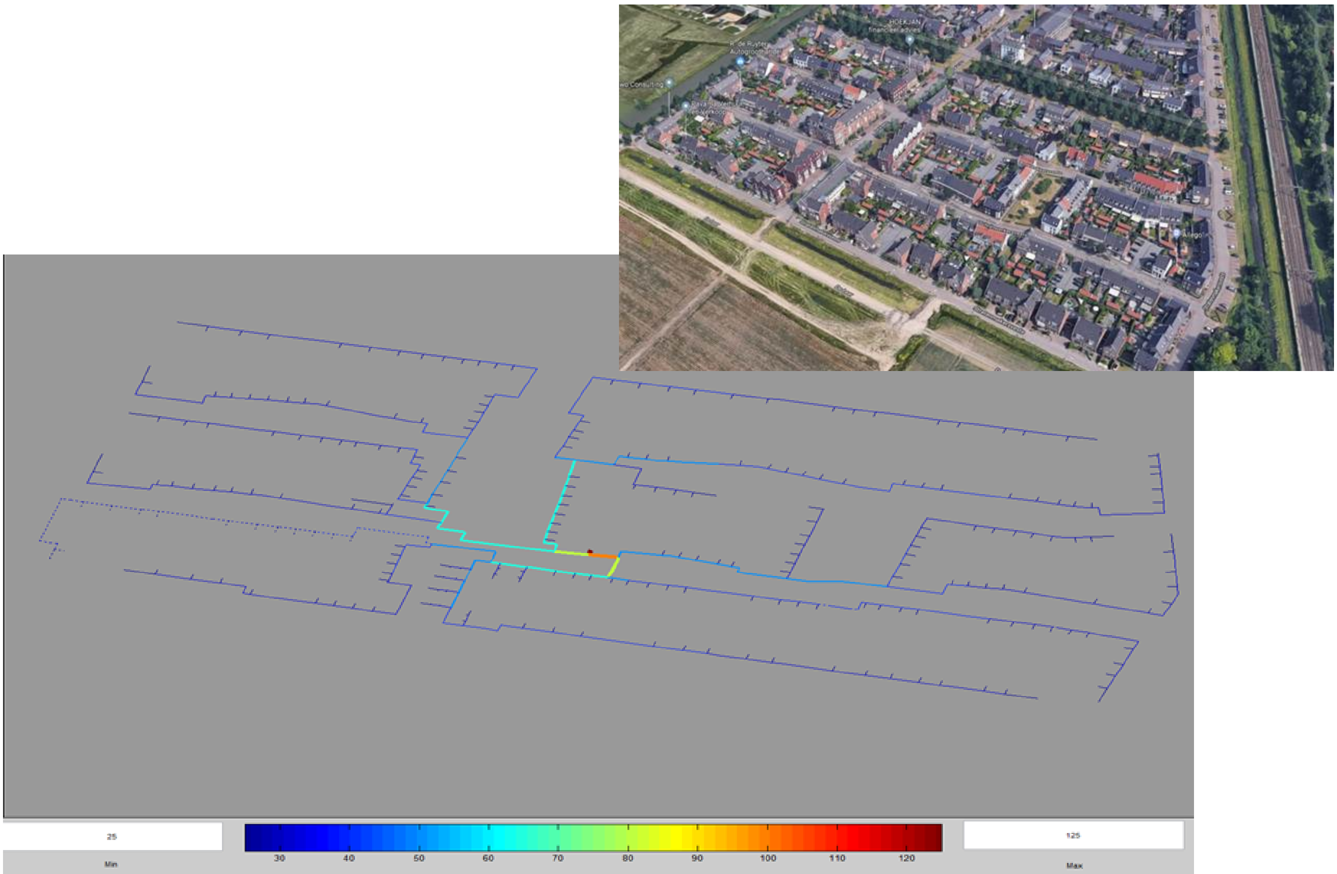

Figure 4.1: Aerial view of the district Schuytgraaf in Arnhem the Netherlands and a layout of the district heating piping network.

4.1. Installed capacity of the HTS

The installed capacity of the entire Schuytgraaf net is determined using the total installed capacity of the users in the net and the simultaneity factor. A simultaneity of max 65% is assumed in the system based on experience. This means it is assumed at any given time the maximum load is 65% of the potential heat demand in the system. Every user in the system has a heat connection that is capable of 8 - 22 kW of heat delivery. The Schuytgraaf net has 247 users connected to it with on average 12.2 kW so this would mean $12.2 * 247 = 3013.4kW$. However it is assumed that in reality this will never be the demand but that the maximum is 65% of this demand. $3013.4 * 0.65 = 1960kW$.

4.1.1. Piping network

The entire length of the AGH12 network is a Little over 3 km divided over the different diameters of piping shown in table 4.1. This is the so called trace length, the total pipeline length is twice this length since there is a supply and return pipe.

DN-size	Length [m]
DN125	2.6
DN100	8.6
DN80	21.7
DN65	150.9
DN50	236.1
DN40	646.9
DN32	1168.3
DN25	787.6
Total	3022.7

Table 4.1: Piping in the AGH12 net.

DN25

The pipes marked as DN25 are in most cases not DN25 but S25 are Steel-flex pipes. These are more flexible pipes that enter the houses. Most of the time these are not very well insulated because they need to manoeuvre into and through the building in limited space. For some part this will mean the heat loss will increase, for other parts this will mean the heat loss will decrease because they are in a warmer surrounding surrounded by air for example in the crawl space of a building could make the pipe quite good insulated, this makes it pretty unpredictable. Since the situation for each of these different pipe sections is different and unique all of them are assumed to be DN25 in the model.

4.2. Operating the system

The mass flow in the system is controlled by the pumps in the HTS and the inlet temperature by the valve ate the primary side of the heat exchanger in the HTS. The input pressure is around 22 bar and drops caused by the resistance caused by flow through the system and the heat exchanger to around 4 bar. The pressure's main function is to ensure there is a mass flow through the system. The pumps are controlled by pressure sensors at certain locations in the system. The pump is set to establish a certain pressure for each of the measuring points in the system. This way the supply side of the system will respond very quick to a sudden change in demand. When a consumer start demanding heat the flow will increase and the pressure sensors will detect a decrease in pressure in the system. This will trigger the pumps to work harder to ensure the pressure will increase pumping again. This will also mean mass flow though the system increased since the consumer has a mass flow through their heating unit. The pressure distribution over the system is better visualized via a pressure triangle, Appendix E. The system is only operated by changing the mass flow and not the input temperature. The input temperature for a district heating system for central heating and individual tap water is kept constant at 75 degrees Celsius. The input temperature is measured at the outlet of the heat exchanger in the HTS, to control the inlet temperature a valve is placed at the primary side of the heat exchanger, figure: 4.2. When a decrease or increase in the input temperature is detected the valve at the primary side increases or decreases the mass flow.

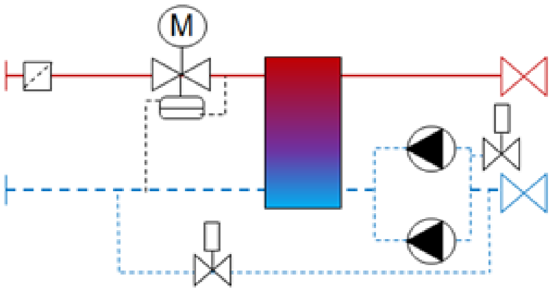

Figure 4.2: Schematic of the heat transfer station

4.2.1. Bypass flow

In the ideal scenario, there is a temperature input of Tin and the temperature at all the customers would be 70 degrees. Some customers are further away from the heat source then others meaning the ones furthest away will normally get a lower temperature. Especially with alternating environment conditions and individual heat demand it is very difficult to regulate the system. In the current situation the supply sets at the users ensure they get the minimum temperature by letting water flow through a valve even when there is no heat demanded by that user, this is the bypass flow.

4.2.2. Pump

The flow in the system is regulated with two pumps in the HTS. The pumps are parallel connected. This means they can easily be operated independent of each other. One pump should be able to produce 50 - 70 % of the design flow when the other is switched off. This will mean the rest of the time when the system is not operating in design condition the amount of pumping power is overdimensioned. This could have a big effect on the pump efficiency when a pump is not operating within it's pumping range. Every pump has an optimum operating range in the efficiency curve. When a pump is operating far form the operating curve the efficiency will drop significantly. This means it is important to match the operation conditions with the pump. The absolute minimum mass flow in the system is during summer time at $1.162kg/s$ corresponding with $3.816m^3/h$. This is on the very low side but still within the range, pump curves are in the appendix F.

5

Analysis of a system in operating conditions

5.1. Design condition

The design condition of the system is in ideal conditions, which means a few assumptions are made. The supply temperature is set at 70 degrees Celsius for the entire supply pipeline. The temperature of the ground is always at 10 degrees Celsius and the load or demand is constant, which means the mass flow is constant. The total load in this case is the design load or the maximum load condition. In this case the minimum input temperature for the system,to make sure all consumers get the required minimum temperature of 70 degrees is higher then the 70 degrees. That is why Nuon sets the input temperature for the system at 75 degrees. This will mean the users closer to the HTS will receive a water temperature of somewhere between the input temperature and the required 70 degrees while the users at the end of a string will get the minimum temperature of 70 degrees.

5.2. Load condition

In reality the system almost never operates in the design condition. The maximum heat demand is almost never met. Not even in very cold winters. The data of the AGH12 network from 2005 was monitored and is analyzed. The percentage of heat demanded per hour vs the total installed power is plotted in figure 5.1. Over an period of one hour the total demand is never higher then 60 %. The maximum heat demand over a period of 5 seconds in this system for this year was 78 %. Although the winter of that year hasn't been a very mild one this winter was also not as cold as others have been [8]. Other district heating nets which had their data logged show roughly the same numbers.

5.2.1. Diurnal Load curve

The daily load curve shows a small peak in the morning when people get up to start their day and take a shower. And a big peak in the afternoon when people get home and start heat their homes. In summer the two peaks are almost similar in size while in winter the second peak is much higher because people heat their homes. In winter months the base load is quite high because people want to keep their houses at a minimal temperature and in the summer people turn of their central heating systems and the only demand is used for tap-water. This is also the reason the daily spread is much bigger in winter. The curves in figure 5.2 clearly show not besides that the demand is much higher in winter also the daily peaks of the demand are much higher during winter, or the spread between the minimal and maximal demand.

As the data is collected with an interval of one hour the averaged load curves still show an angular pattern. To solve this and to get a smoother, more realistic load curve a polynomial fit is done over the monthly average data, figure:5.3.

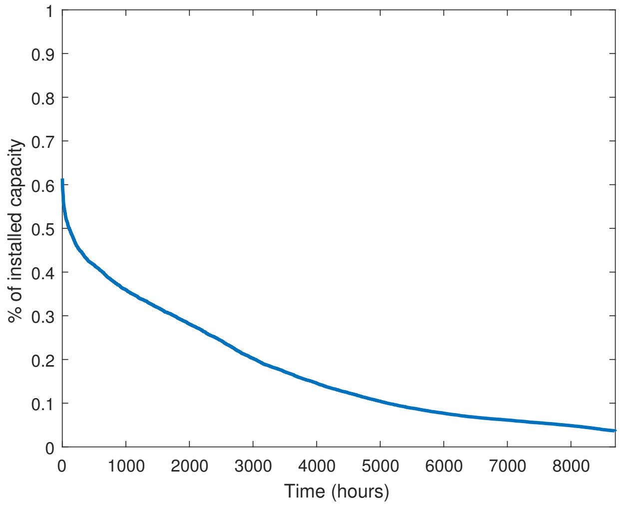

Figure 5.1: Percentage of the installed capacity demanded.

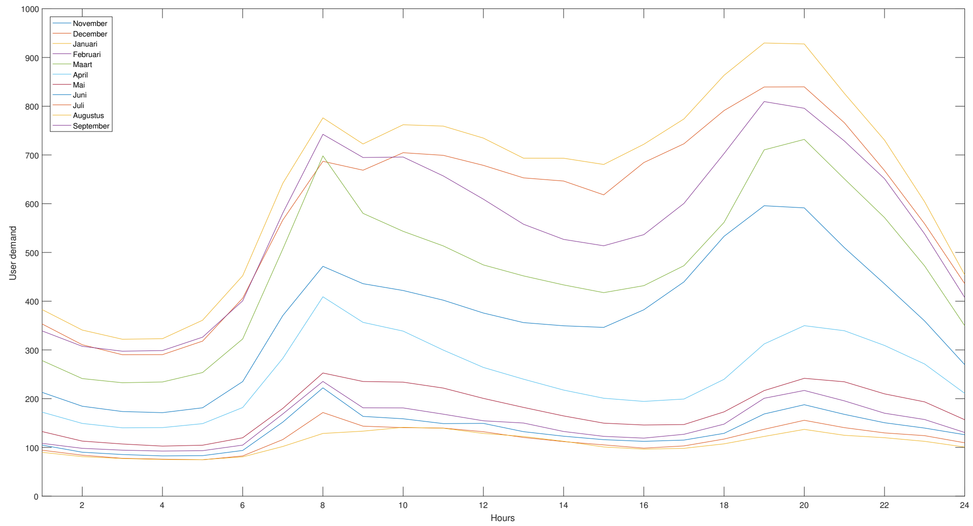

Figure 5.2: Daily demand curves, averaged for each month.

5.2.2. Annual Load Curve

Plotting all the hourly data of the entire period of 2005 in one plot result in the blue plot of figure 5.4. The annual load curve shows a peak in winter (begin and end) and valley in the summer (middle). Also the difference in of spread between winter and summer is clearly visible. Where in winter the spread can get up to a difference of 700kW in summer the difference is some days less then 60 kW between a peak and a valley. The data set of the month October rearranged from high to low so that looks a little off. To get an annual load curve a polynomial fit is done over the daily mean of the user data. This gives the red plot in figure 5.4.

5.3. Ground Temperature

In chapter 2 it was concluded that The ground temperature will be of big influence on the heat loss of the system. The depth to which the ground temperature is influenced can be divided in roughly two parts. At a depth more than 20 meters the ground temperature is almost constant year round. From roughly 1 meter till 20 meters the temperature varies throughout the year following the seasonal

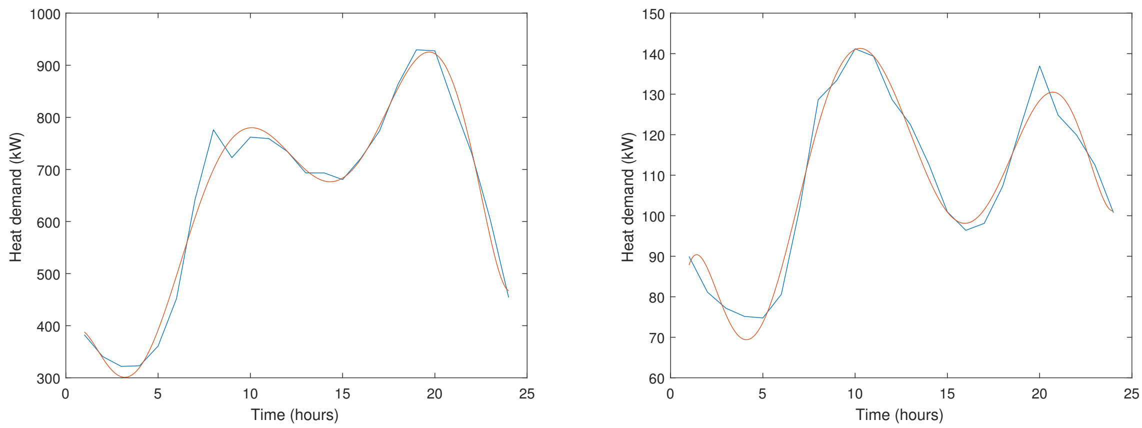

Figure 5.3: Demand curves for January(Left) and August (right)

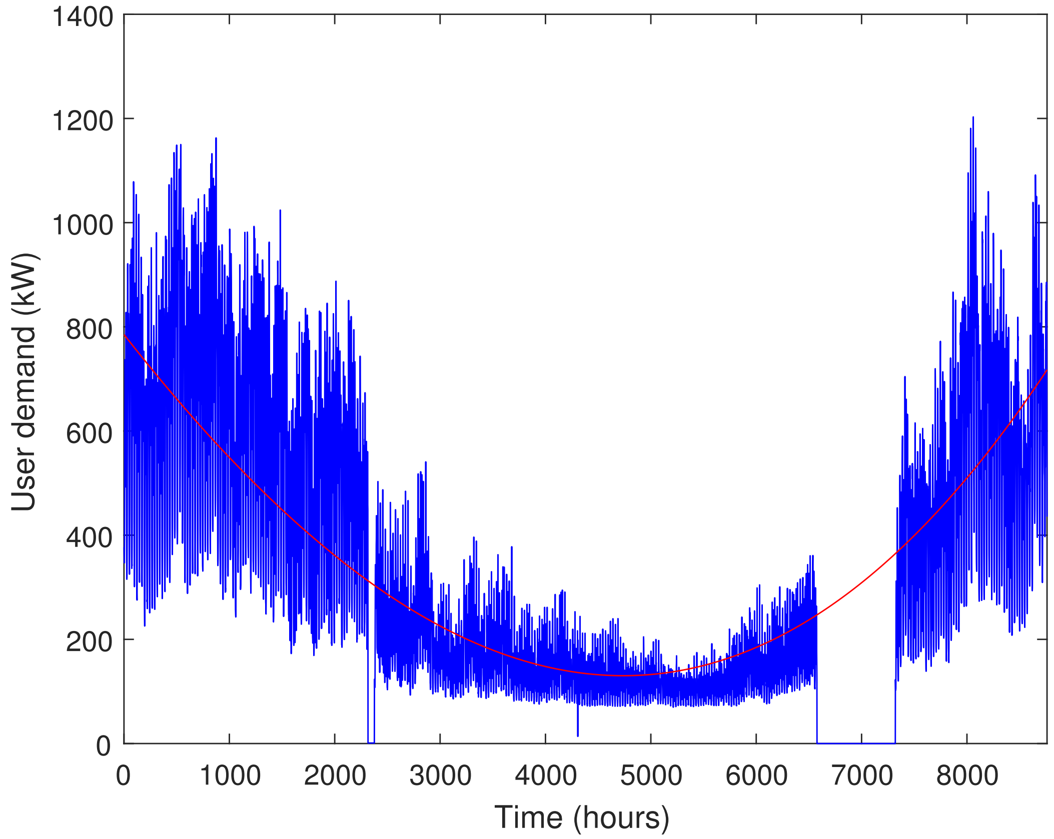

Figure 5.4: Measured hourly demand data of an entire year with polynomial fit.

change of ambient temperature. From the top surface till 1 meter depth the temperature of the ground is effected daily due to diurnal cycle, solar radiation and other weather conditions [9].

In literature two formulas can be found that describe the ground temperature at a certain depth, Baggs[10] and Kasuda[11] both are in appendix G. The formula of Kasuda will be used: G.2. This is plotted together with the ambient temperature that was measured over the same period as the user data from section 5.2.

$$T_{soil(D,tyear)} = T_{mean} - T_{amp} * exp(-D\sqrt{\frac{\pi}{365\alpha}} * cos\frac{2\pi}{365}(t_{year} - t_{shift} - D/2\sqrt{\frac{365}{\pi * \alpha}})) \quad (5.1)$$

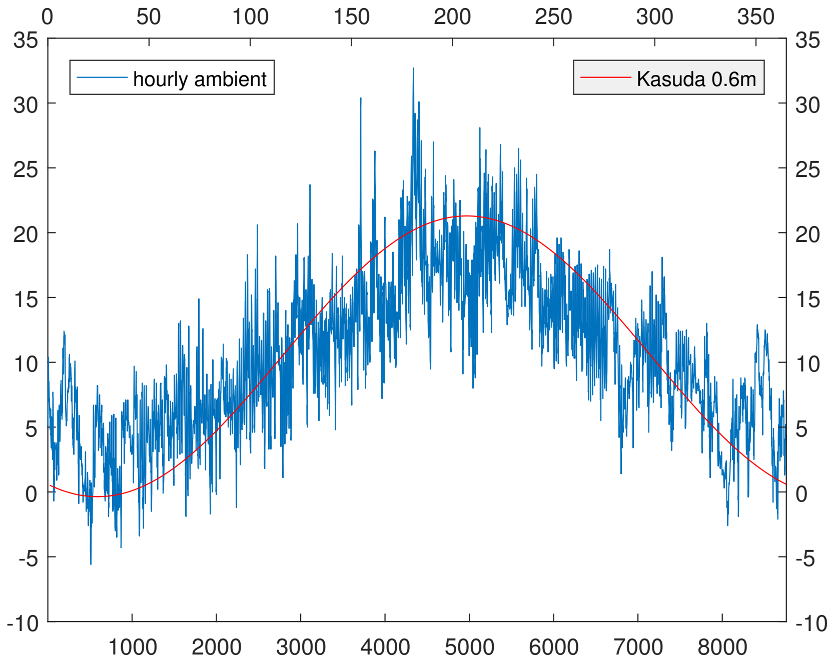

Figure 5.5: Measured ambient temperature and plot for the ground temperature at a depth of 0.6m as given by Kasuda.

5.3.1. Measured Temperature

The ambient temperature is measured for the same period in 2004 and 2005 as the user data. This is plotted in figure **??**. Besides the data of the ambient temperature both equations G.1,G.2 are plotted in the figure. From here on the formula of Kasuda is used to estimate the ground temperature at various depths which is plotted in Appendix H.

5.4. Ground conductivity

A second important ground parameter is the conductivity. The conductivity is heavily depended on the type of soil and the moist level of the soil. This will be different in every location and will even differ from season to season. During the autumn and winter season the ground usually contains a higher moist level then during summer, at day's of long dryness the ground will contain much less moist. The penetration time of the air temperature to a certain depth in the ground is very much influenced by the conductivity which is by itself a result of the type of ground eg.(clay, rocks, sand) and the moist level of the ground[6]. The depth at which a pipe is supposed to be buried is around 60 cm. The ground temperature at these depths is influenced both by an annual cycle as well as diurnal depending on the conductivity of the soil [12]. In theory the district heating system of Nuon is buried in a layer of sand however this isn't done consistent in practice. An overview of how the pipes should be buried in the ground is shown in AppendixC. It was concluded, form the analysis of the heat loss with the pde-solver in chapter 2, that the conductivity in the range of values that is common for the ground the effect on the heat loss is minimal. The influence it has on the conductivity is also minimal and will strongly depend on the local conditions in region you are and specific conditions of the ground like the moist level. It is assumed the conductivity is 1.6 $W/m*K$ which is the common/average used conductivity for sandy soil [13]. The ambient temperature and convection between the ground-air interface can change rapidly during a period of 24 hours (day and night) or even shorter due to the change in weather conditions or the difference between shadow or direct sun. As discussed in the section for ground temperature this effect will be neglected. The daily temperature fluctuation at a depth of 60 cm is very small and will not be taken into account [14],[15].

6
Heat loss

An analysis is done determine what happens in the system during operating conditions. First the heat loss is calculated for full load or ideal conditions. Then the system is evaluated for some diurnal and annual load curves as determined in the chapter 5.

6.1. Logstor calculation tool

Logstor is the supplier of the pipe sections. They provide an online calculating tool which can be used to estimate the heat losses in a system. For the Logstor heat loss calculations the conditions are assumed ideal meaning 70 degrees Celsius supply, 40 degrees Celsius return and 10 degrees Celsius ambient. The results for series 1, series 2 and twin are displayed in table 6.1.

piping system	Heat loss ideal [MWh/year]	Heat loss [GJ/year]
Series 1	484.84	1745.42
Series 2	357.38	1340.92
Twin	270.04	1017.75

Table 6.1: Heat loss calculated by the Logstos calculation tool

It is still assumed the temperature is constant over the entire system while in reality there is a heat gradient from the input of the net to the users. To get an more accurate calculation of the heat loss in the system and an better understanding of what happens in the system the heat gradient should be taken into account the Logstor calculation tool is no longer workable, so the state-space calculation method is introduced.

6.2. Full load condition

In the design condition the load is as explained at 65% simultaneity. For the considered network this means a total customer demand of 1995 kW. The input temperature is 75 degrees and the reference temperature is the ground temperature at 10 degrees Celsius. In this case the heat pumped into the system is at 75 degrees Celsius. For the calculation model we assume the total demand is equally divided over all the users independent on the size of their supply set.

6.2.1. Heat loss calculation using Matlab model

The results from the heat losses calculated using the Matlab model for a system in operation will be discussed and evaluated here and compared to the Logstor calculations. The results for the full load conditions are displayed in table 6.2. The results from Logstor were given in MWh but converted to GJ because the commodity trading for heat is done in gigajoules. Comparing the Logstor and Matlab calculations for the ideal situation already a difference is noted.

piping system	Ideal Logstor [GJ/year]	Ideal Matlab [GJ/year]
Series 1	1745.42	1939.5
Series 2	1340.92	1671.8
Twin	1017.75	1165.3

Table 6.2: Heat loss in GJ/year in the ideal situation

6.3. Operating condition

In reality the system is almost always in part load condition. There will be extra heat loss due to mass flow being pump round the system to keep the system at a minimum temperature of 70 degrees Celsius for the customers. In practice this means the heat loss in the return flow increases a lot because the return flow heats up rapidly. This is not seen in the full load conditions because the mass flow is high enough to keep the system at temperature. The daily and the annual load conditions are discussed. The ground temperature is kept constant during the course of a day because the diurnal effect of temperature change at a depth of 0.6 meter is set to be zero. However the ground temperature is assumed to be the monthly average for that month. These calculations are done real time to simulate what happens in the system.

6.4. Diurnal load curve

The analyzes have been done for a case with high demand during winter and low demand during summer. During winter time there is a wide spread between peaks and valleys in terms of demand, the month with on average the highest demand is January. During summer time the average demand is much lower and also the peaks in the demand are relatively much lower then during winter time, the demand is more constant during the course of a day. The month with the lowest average demand and peaks is August. A polynomial fit is applied as discussed in the previous chapter to smooth out the angular profile caused by the data being measured with an interval of one hour 5.3.

6.4.1. Mass flow

Figure 6.1 shows the mass flow during a day. The yellow line represents the mass flow that follows from the demand of the users, the orange represents the bypass mass flow in the system that ensures the system is kept at the minimum temperature of 70 degrees on the supply side. Also the amount of bypass flow is very high increasing the return temperature. In the winter 6.1 (left) the by-pass flow is very low meaning the user demand flow is high enough to keep the system at temperature. In summer time 6.1 (right) this is very different with the bypass mass flow at times actually being higher then the demand flow.

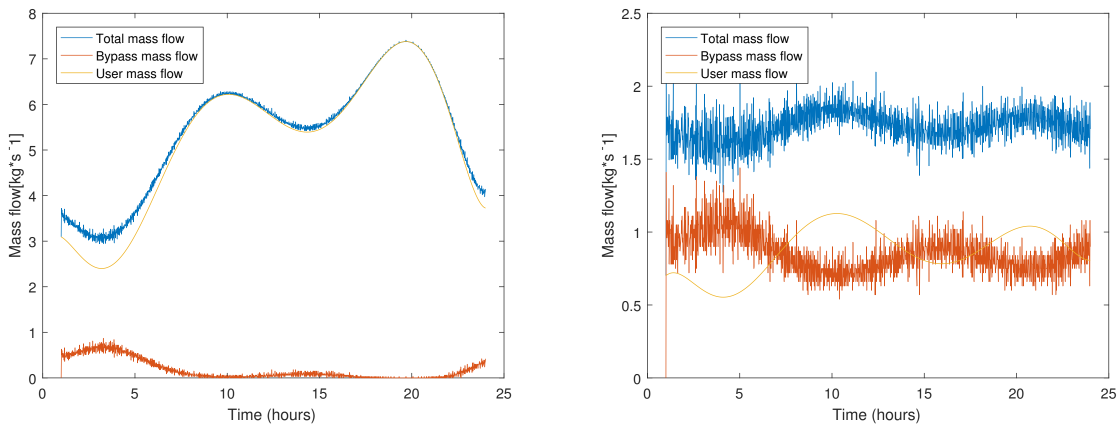

Figure 6.1: Mass flows in the system for an average day in winter(Left) and summer(right).

The effect of the bypass mass flow is seen in the return temperature 6.2 which increases as the by-pass flow increases. This is expected since extra heat is pumped around while this is not consumed by the users.

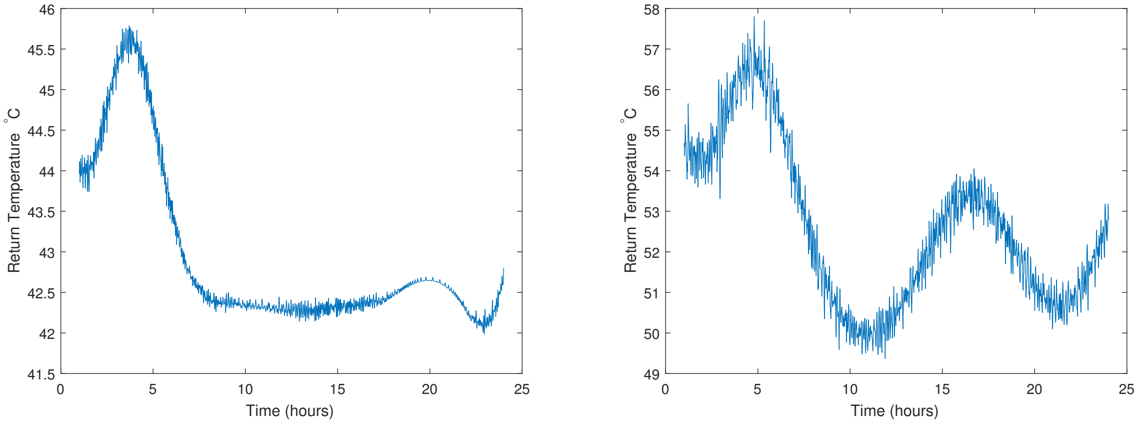

Figure 6.2: Return temperature curve for and average day in winter(Left) and summer(right).

6.4.2. Heat flux

In figure 6.3 the blue line shows the total heat input in the system. The reason it is so high frequently fluctuating is caused by the bypass mass flow being on and off. The total heat input is the sum of the heat demand and the heat loss. The orange line shows the demand curve, the yellow line shows the heat loss. The heat loss on the daily bases stays nearly constant independent of what the demand does. This is true for both the summer and winter case.

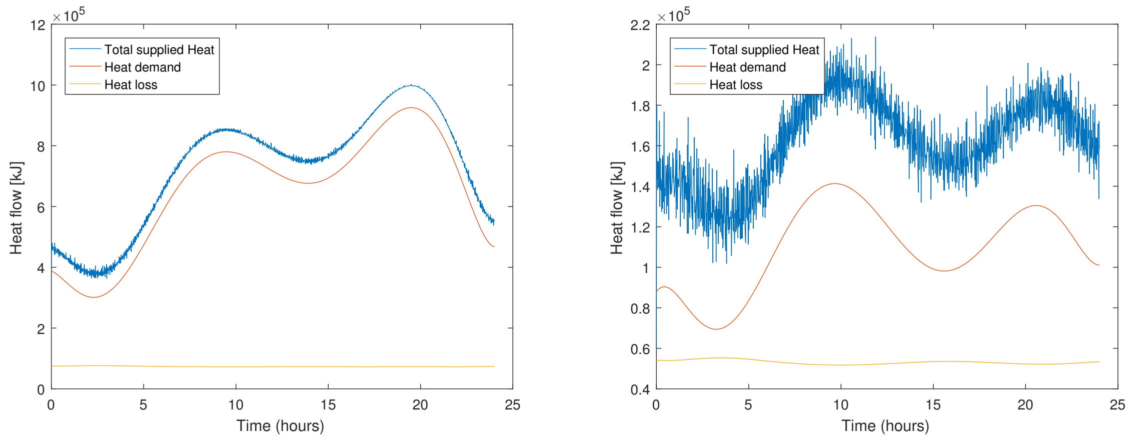

Figure 6.3: Heat flow in the system on a average day in winter(Left) and summer(right).

From here on it is assumed that for the analysis of the annual heat loss, the daily demand and the heat loss can be averaged, because the results will stay the same. Although the demand curve is fluctuating a lot the heat loss stays nearly constant. This means averaging both over the course of a day will give the same results for the total. The averaged annual heat curve is given by the polynomial fit over the demand data in figure 5.4.

6.5. Annual load curve results

When the annual situation is analyzed the daily demand is averaged as explained in chapter 5. The annual change in ground temperature at a depth of 0.6m is taken in account using the formula of Kasuda 5.5. The bypass flow should be taken into account. For a daily average demand and ground temperature the heat loss is calculated by iterating the heat loss calculation and increasing the bypass mass flow until the minimum required temperature of 70 C is met and the temperature gradient is stable. This will also reduce the calculation time because a year round real time calculation would take too much time. Besides from the data set some measurements are missing or disturbed, by using a fit over to determine demand curve all of this is immediately solved.

6.5.1. Heat loss

The total heat loss is divided over the supply and return pipe lines. This gives a interesting insight in the total heat loss especially for the return side. The supply side shows an expected curve with high heat loss in winter and lower heat loss in summer 6.4. The return side starts by following the same trend but seems to stabilize during summer.

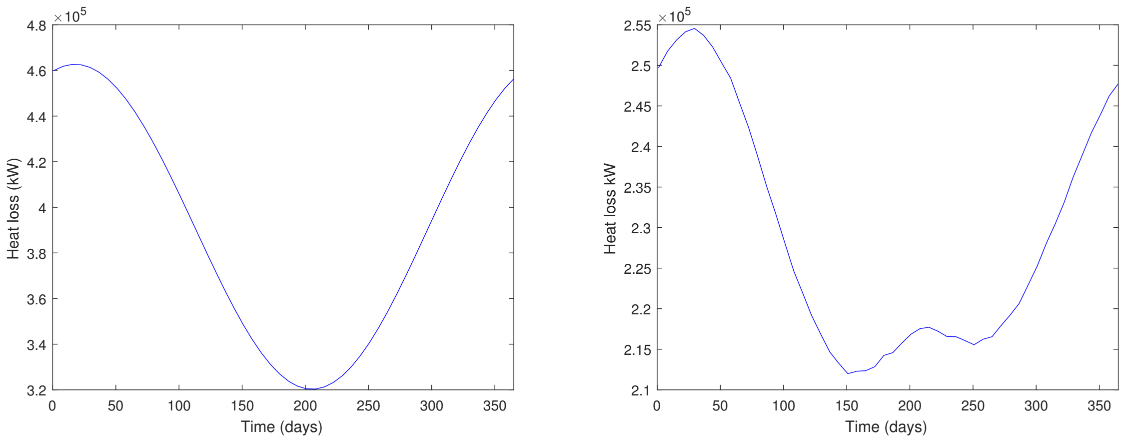

Figure 6.4: Heat loss supply pipe lines (Left) and return pipe lines (right)

6.5.2. mass flow

The plot for the the mass flow through the network shows three graphs, the customer mass flow, bypass mass flow and the sum of the two 6.5. This plot could also give an explanation for why the heat loss in the summer time won't decrease more. The first thing that is noted is that the bypass flow at no time reaches absolutely zero meaning there is always at least a little bypass flow. In winter time the by-pass flow is very low especially compared to the flow demanded from the customers. This means almost no hot water is pumped back into the return flow, meaning the return temperature at the HTS and the heat loss from the return pipeline should follows the expected trend. In summer time the bypass flow at times is larger then the user flow meaning a lot of hot water flows directly back in the return pipeline. This means the return temperature in the return pipeline will heat up, increasing the heat loss and the return temperature at the HTS. This is also what was seen in the results.

6.5.3. Efficiency

The efficiency is defined as the heat demanded by the users divided over the total heat being put in at the HTS. The efficiency of the system in the winter is over 90% however in summer during times of low demand the efficiency drops below 70 percent.

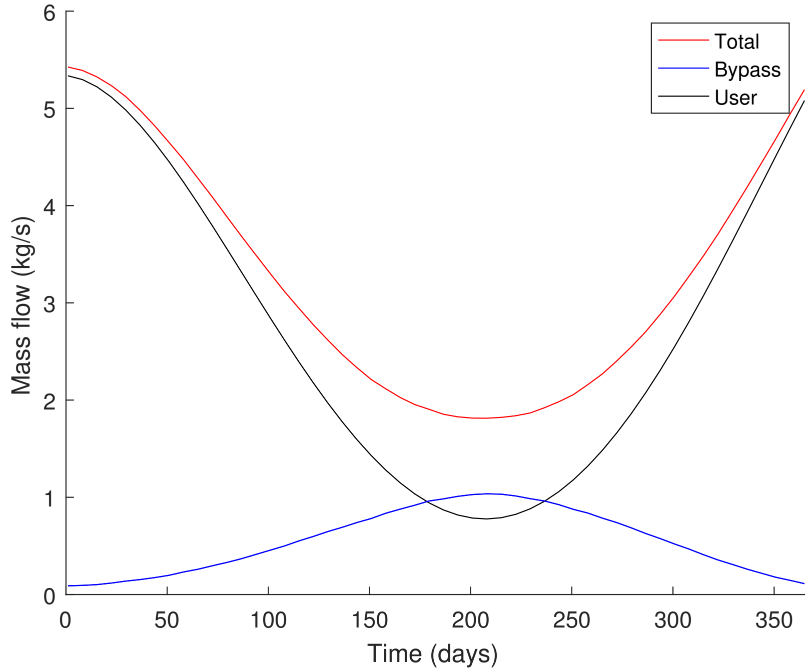

Figure 6.5: Mass flows through the system.

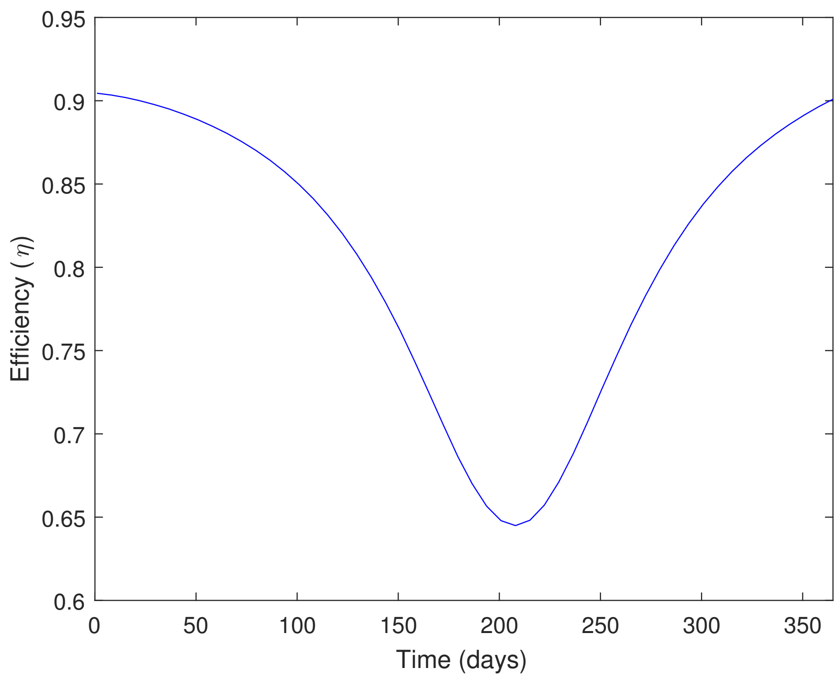

Figure 6.6: Efficiency of the system.

7

Results

When the series 2 or twin piping system is used instead of the series 1 there will be an expected decrease in heat loss based on the knowledge obtained in chapter 2. Besides this decrease in heat loss the amount of bypass flow and the return temperature also have a great impact on the performance of the system as seen in chapter 6.

7.1. Series 2 and Twin

Changing the piping will mean a change in heat loss, this will almost always mean a decrease in heat loss. When loss heat is lost the temperature delivered to the users could increase. This means either a decrease in (bypass) mass flow or the inlet temperature can be set lower.

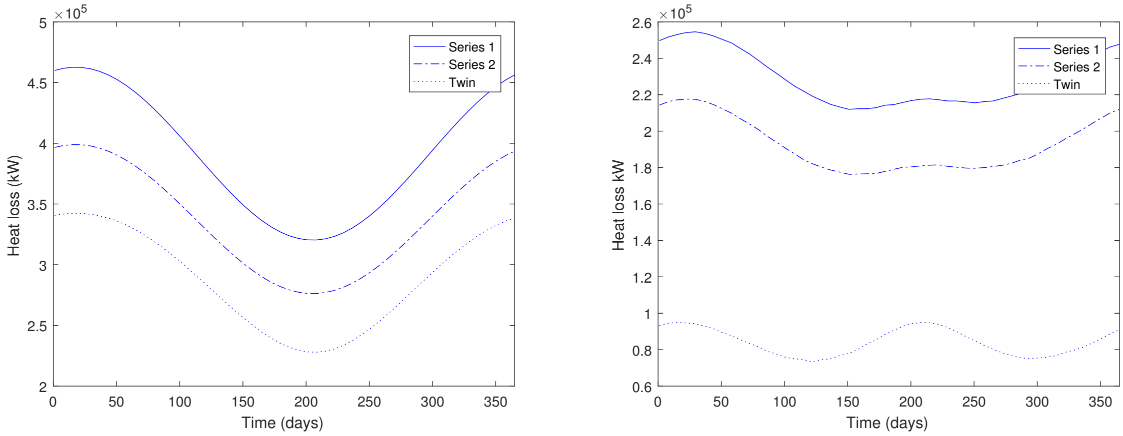

Figure 7.1: Heat loss supply (Left) and return (right) for series 1, series 2 and twin.

7.1.1. Effect on heat loss

The effect on the heat loss are quite as expected with quite an decrease in heat loss for series 2 and even more for twin 7.1. Interesting is the even clearer peak in the heat loss of the return flow in the summer for the twin system. Checking the difference between the return temperatures we see a quite interesting development with a rise of up to 10 degrees in summer time. The temperature difference for the series 2 and twin is already a little better, but still in the summer there is a high increase in temperature due to the bypass flow 7.2. This probably causes the increase in heat loss in the high summer when the by pass flow is high. Plotting the efficiency of the system 7.3, in wintertime the efficiency is above the 90%, however there is still a dip in the summer.

The amount of gigajoules lost over the entire year is given in table 7.1. In comparison with the ideal calculations done with Matlab there is only a small difference. Now there is much more insight

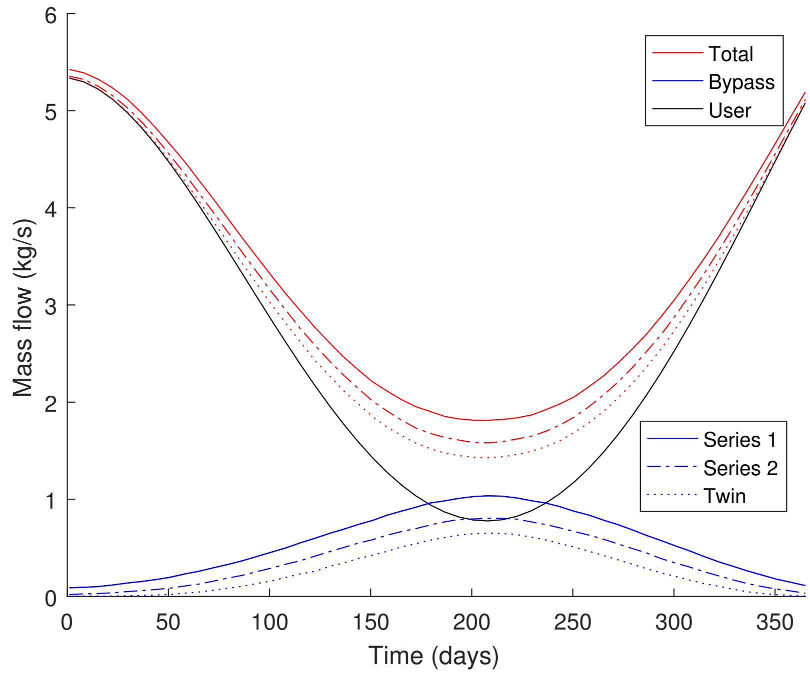

Figure 7.2: Mass flows through the system for series 1, series 2 and twin.

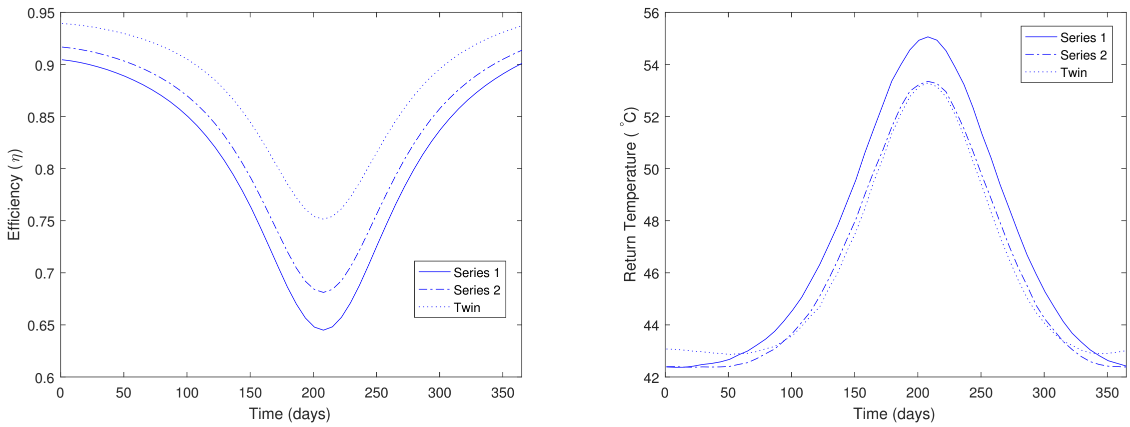

Figure 7.3: Efficiency (left) and return temperature(right) for series 1, series 2 and twin.

in what actually happens in the system. Two main issues are seen in these results: the low efficiency in the summer and the high return temperature. An effect of this is the relatively high heat loss at the return side in the summer, the amount of heat lost does not follow the expected trend seen at the supply side but it stays constant. In winter for the twin system the bypass flow is zero meaning the temperature is 70 degrees or higher in the entire system. During winter the return temperature also rises meaning there is to much heat put into the system. All of this could be solved by lowering the input temperature, decreasing the heat loss even further.

piping system	Ideal Matlab [GJ/year]	Heat loss [GJ/year]
Series 1	1939.5	1957.1
Series 2	1671.8	1671.2
Twin	1165.3	1180.1

Table 7.1: Heat loss in GJ/year.

8

Changing the operating settings

To further improve the performance of the system, especially during summer when the system has very low efficiency, some other changes in the system will be considered. This might involve changes that can be applied to existing networks and result in a decrease in heat loss. All these calculations are done for the series 1 system.

8.1. By pass flow

The bypass flow can be either increased or decreased. Increasing the by-pass flow might seem counter intuitive but [16] already showed that increasing the mass flow through a pipe will decrease the heat loss up to a certain point at which the heat loss will become nearly constant. The question is if the operational conditions are beneath this critical mass flow. Especially during summer when the mass flow in the system is very low which could explain part of the low efficiency in summer. Another advantage of increasing the bypass flow is that it will stabilize the work needed to be done by the pump. There will be some difference in head but in general the pumps will be in a more stable operating condition, so this will increase the pump performance. However increasing the bypass mass flow will mean an increase in the return temperature and so more heat loss in the return pipe, this is also seen in the result.

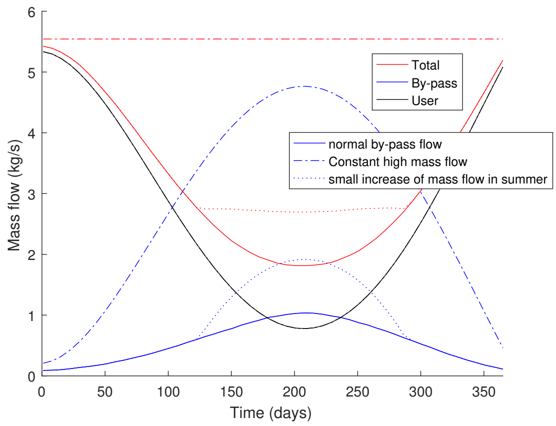

Figure 8.1: Increased mass flow.

The bypass mass flow is increased for two different cases. A constant high mass flow the entire year round and a small increase in mass flow only in summer when the mass flow would be very low **??**.

The results show a increase in return temperature in both cases. Also a slight increase of heat loss at the supply side and a big increase at the return side. The result show that increasing the bypass flow is actually undesirable causing a high increase in heat loss, especially at the return side.

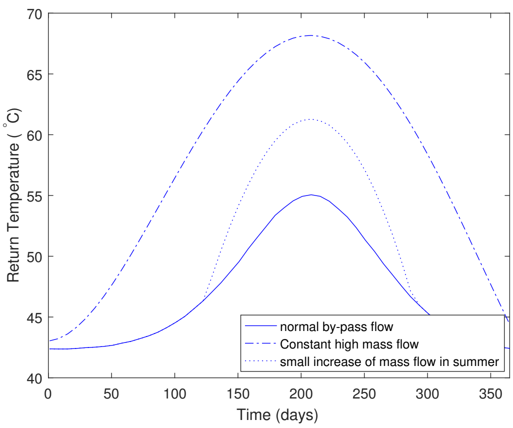

Figure 8.2: Return Temperature for increased mass flow.

8.2. Temperature

The input temperature of the system can be increased or decreased. The temperature input of the system is determined by a heat curve. For a 70 - 40 system with tap water the heat curve is set at a constant temperature of 75 degrees Celsius year round. To check the effect of the input temperature on the amount of heat lost the input temperature of the system is varied between 72.5 degrees Celsius and 80 degrees Celsius in steps of 2.5 degrees. The results for the mass flow are shown in figure 8.3, the mass flow shows as expected a decrease in the bypass flow when the temperature increases.

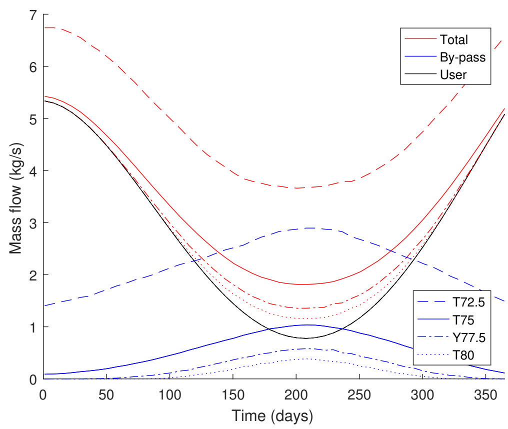

Figure 8.3: Mass flows in the system for a range of inlet temperatures.

8.2.1. Temp decrease (Tin 72.5)

Decreasing the input temperature will mean an increase in bypass flow. From previous results, of the series 1 system with an input temperature of 75 degrees, the conclusion was that the increase in return temperature in summer were causing a relative high heat loss loss in the return flow pipe line, due to the high bypass flow. Decreasing the input temperature only further increases the total heat loss caused by the increase in bypass mass flow.

8.2.2. Temp increase (Tin 77.5 - 85)

Increasing the input temperature means a decrease in bypass flow as expected. Interesting to see is the effect of this on the heat loss and output temperature. The output temperature 8.4 shows in winter time an increase in output temperature as the temperature in the entire system increases. In summer time the output temperature show a decline compared to a lower input temperature, this means that in summer time increasing the in input temperature will decrease the output temperature. This is actually corresponding with the decrease in bypass flow.

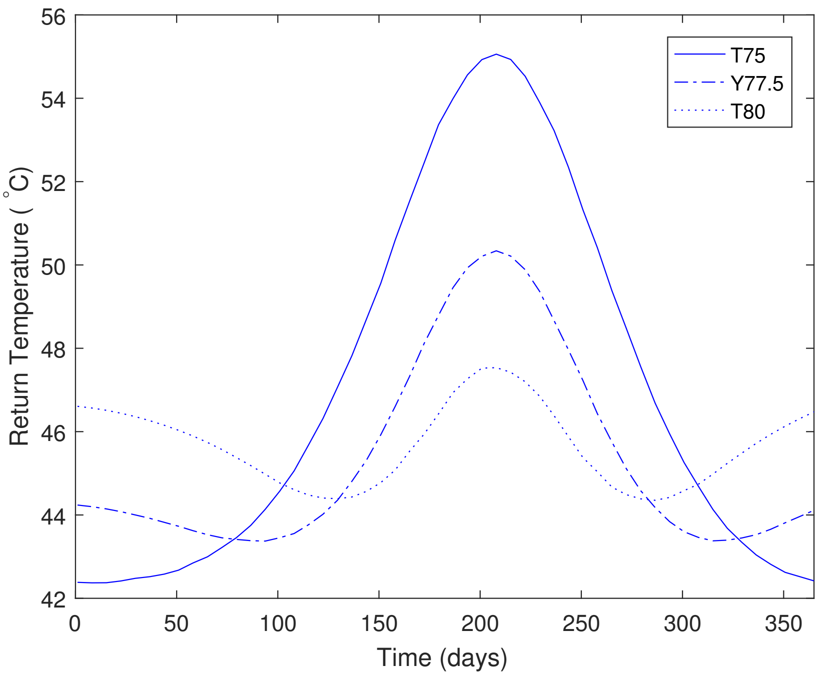

Figure 8.4: Return temperatures for different inlet temperatures.

This will also mean a decrease in heat loss in the return flow in summer time 8.5, while the heat loss in the supply pipeline shows a steady increase year round 8.5. The heat loss on the supply and return side added together show that the total heat loss in summer is actually lower when the input temperature in the system increases to 80 degrees 8.7. At 82.5 degrees the extra heat loss at the supply side is more then the decrease in heat loss at the return side.

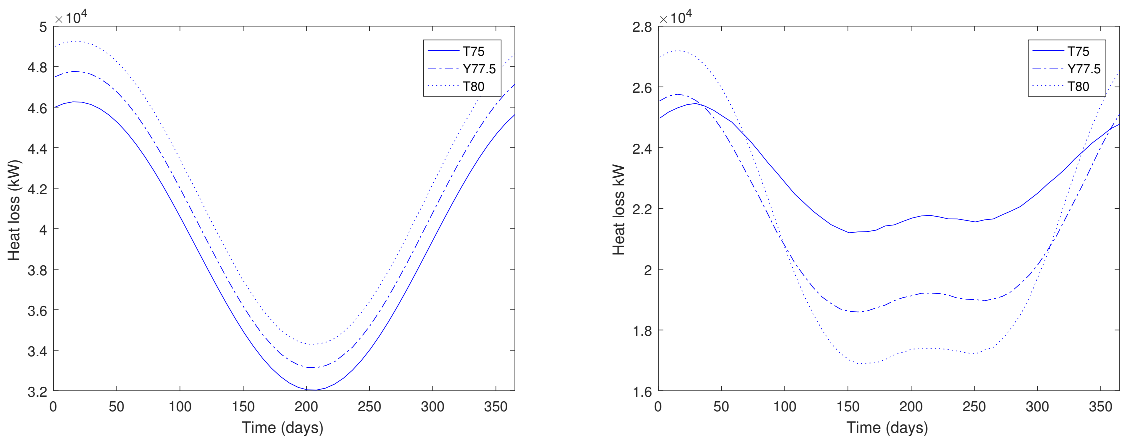

Figure 8.5: Heat loss of the supply pipelines(right) and return pipelines(left).

The heat loss of the supply and return pipelines added together give the total heat loss. From the plots of the total heat loss, figure: 8.7, can be concluded that the least amount of heat is lost when the input temperature is varied. By increasing the temperature in the summer to 80 degrees Celsius and keeping the temperature at 75 degrees in the winter around 20 Gj can be saved increasing the efficiency of the system with approximately one percent 8.1.

Table heat loss GJ/Year

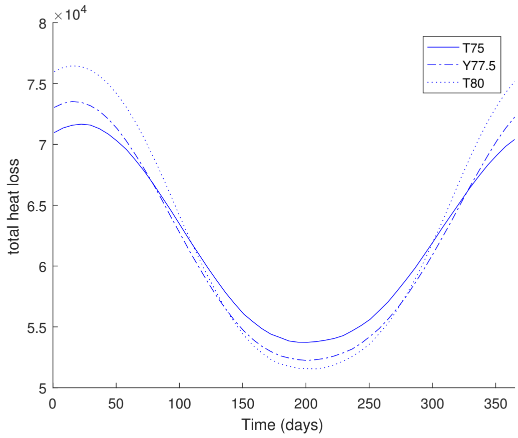

Figure 8.6: Total heat loss of the piping network

Input temperature (C)	Heat loss [GJ/year]
72.5	2062.1
75.0	1957.1
77.5	1951.3
80.0	1989.4
varying	1937.0

Table 8.1: Heat loss in GJ/year.

8.2.3. Changing the location of the bypass flow

The bypass flow valve can be placed at another location in the system then at the consumer sets. For example at the end of a DN32 string. The network is then kept at temperature with bypasses in the DN32 pipes. This way the heat loss in the DN25 lines to the customers is not taken into account. Disadvantages are that the response time of hot water for the customers will increase or the mass flow at the start of demand should be bigger to flush through the cold water. Also could be argued that the temperature at the DN32 bypass should be a little above the 70 degrees to ensure the 70 degrees at the users (this is not done in the calculations). A second advantage of placing the by-pass at the end of the DN32 pipes is that the mass flow can increase in the system without increasing the mass flow though the User's supply set, which would lead to more noise complaints. Also less bypass valves are needed. In the case of our system only 22 instead of the 247 needed originally. Main disadvantages are costly bypass valves needed to be placed in the ground, increase in response time for the customers and increase in return temperature. The results show quite a big decrease bypass flow.

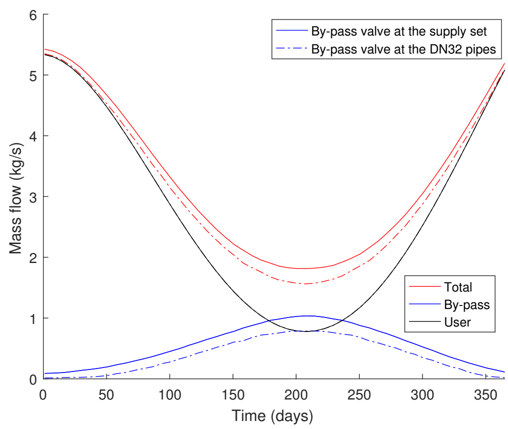

Figure 8.7: Total heat loss of the piping network

9

Total Cost of Ownership Analysis

The total cost of ownership (TCO) is divided in the capital expense (capex) and operating expense (opex). Both will be evaluated in this chapter for series 1, series 2 and twin. The costs for the series 1 system can be calculated pretty accurate, nearly all costs are known at Nuon from the current installed networks. For series 2 and twin a lot of assumptions need to be made. Several sensitivity studies will be done to find out what would happen in different case studies and to cover all the uncertainties.

9.1. Capex

The capex include all the initially investment costs, this divided into two subcategories: the materials and the contractor activities. Using series 2 instead of series 1 will increase the costs because materials get larger and heavier. Also the space needed underground increases meaning the digging of the gutter will cost more. Using twin piping could decrease the costs because there needs to be dug only one gutter and only half of the joints need to be made. However the pipes need to be welded together and this will be a lot harder, also bends and splits will be more complex to make which could increase the costs.

9.1.1. Material Costs

Materials are mostly pipeline sections and joints. Pipelines come in all different sizes and forms, besides the straight lines there is normal 90 degree bends and specials like Z-bends and T-splits. The material costs for the series 1 and series 2 are known at Nuon, all materials are bought from the supplier and manufacturer Logstor. They currently have a contract in which all the prices are negotiated. The prices for the twin system are assumed equal to the deal made in Sweden. The main materials are pipes in different sizes and shapes and joints.

9.1.2. Contractor Costs

Installation of the piping network is done by contractors hired by Nuon. For series 1 the costs are known from current deals with contractors. The prices are given in installation costs per meter depending on the DN-size. For the Series 2 the sizes are the same as series 1, the only difference is the outer dimensions of the insulation layer are one size bigger. That is why it is assumed the costs for series 2 piping are the same as those for the series 1 but DN-size plus one, so the price for intalling a series 2 DN25 pipe will cost the same as a series 1 DN32 pipe and so on. The prizes for the twin system are harder to predict. On the positive side the number of pipes installed is halved and less digging is needed. Also only half the amount of joints need to be placed, however these are bigger in size and so the cost per joint will increase. Other cost increasing factors are the welds that are much harder to make because the two steal pipes are close together.

9.2. Opex

The operating expense of the system and the costs for maintenance on the network. As shown the heat loss compared to series 1 will decrease using series 2 and even more so using the twin pipe system.

For maintenance cost a lot of the failures in the system are at the joints. These are the PE connections between two pipelines. By using series 2 the size of the joints will increase and so will the problems and possibilities of failures. However with the increase of insulation the temperature in the system will be more constant over time and so the stresses in the pipes will decrease. This might decrease the number of failures. In the twin system where the supply and return are interconnected the amount of stress on the pipes will decrease even more. Another big advantage of using twin is that the number of joints is halved because there is only one pipe in the ground.

9.2.1. Maintenance Costs

The maintenance costs are again split into opex and capex maintenance, for example fixing a leakage in the piping system will be opex costs but replacing a part of the network before the end of the initial determined lifetime of the network is capex costs, this will again increase the life time of the network. Over the previous years Nuon has monitored all the maintenance work done on the entire system. This means for the series 1 a good estimation can be done for what the costs will be. For the series 2 these cost will slightly increase because the materials are more expensive and the sizes are bigger, there is also no reason to assume there will be more or less leakages in the system compared to the series 1. The capex part of the maintenance is increased with the same percentage that the material costs increase from series 1 to 2 and ditto the opex part increase with the percentage the activity cost increases. For the twin system however this is different. As explained the amount of joints are halved and the largest number of failures in the piping net occur at the joints. This is why by some it is assumed the costs will decrease. However at every failure both the supply and return are affected so the material and constructor costs will be much higher. Both cases, one with much higher maintenance and one with a much lower costs, are evaluated and the conclusion is that it doesn't really matter because even in the most extreme case with the maintenance cost increasing with 200 % these costs don't have a big influence on the result ore the cost reduction that can be made due to the decrease in heat loss.

9.2.2. Heat loss Costs

The costs for the heat loss are determined as the price per GJ times the amount of GJ lost per year. Again different scenarios and sensitivity analyses are done since the price per GJ and the price development in the future is uncertain. This uncertainty is due to all the different suppliers of the heat. The main heat sources are gas turbine power plants or waste incineration power plants which deliver relatively cheap heat and the prices are agreed on for the coming years. During peak loads back up gas powered heaters are powered up deliver more heat, this will only be for a short period. These systems can also be temporally used when the secondary net is installed before it is connected with the primary net. However the prices of the heat are relatively expensive compared to the main sources.

9.3. Results

The costs of heat loss make up a considerable part of the TCO of the system. The twin system out performs the series 1 and series 2 in in every scenario. Even when the initial material and contractor costs rise significant and the maintenance costs for the twin increase with 200% per meter the total costs over a life time of 30 years decreases.

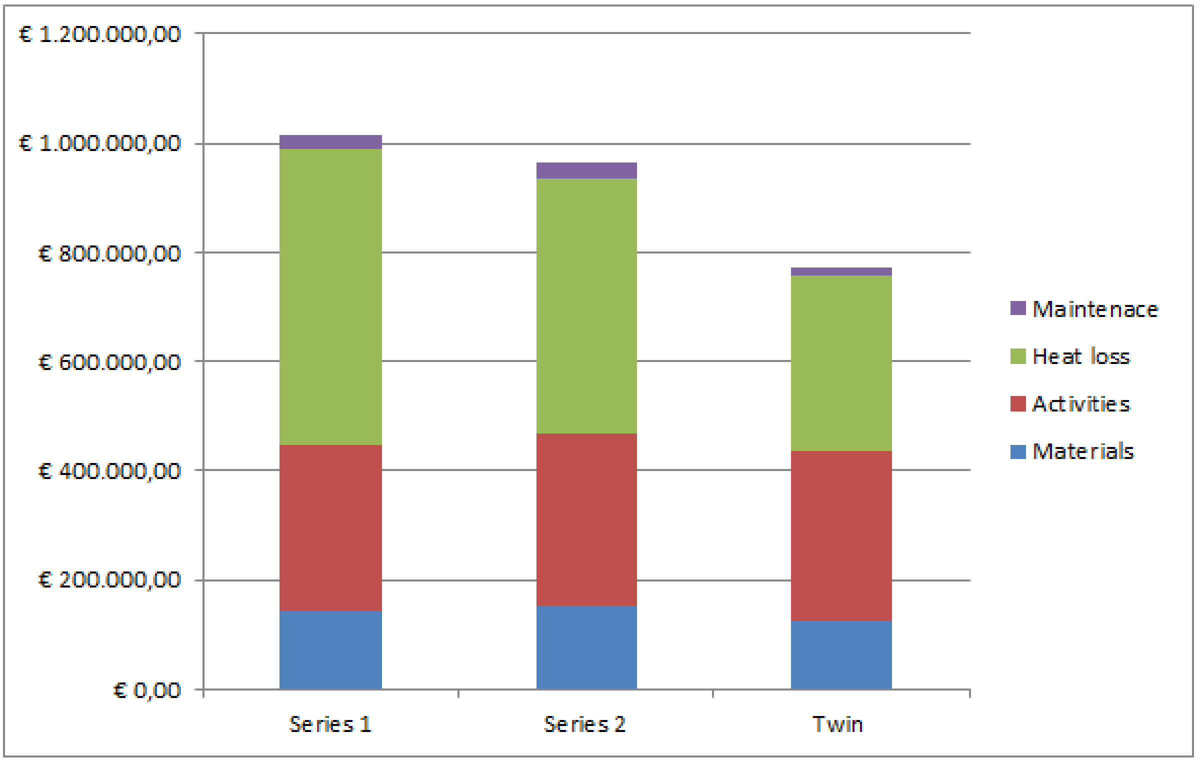

Figure 9.1: Bar graph of the total cost of ownership over 30 years for series 1, series 2 and twin.

10

Conclusion and Recommendation

10.1. Conclusion

The main purpose of this research project was to determine if Nuon should use a different type of pipe in the form of a series 2 or twin pipe instead of the series 1 that is currently being used. From both a heat loss and financial considerations the twin pipe line is the best choice. Using the series 2 or twin instead of the series 1 piping will result in a heat loss reduction of 14.6% respectively 39.70%. By controlling the temperature in a smart way and by placing the by-pass valves in a more strategic place some more savings can be done. Changing the location of the by-pass will be a costly operation in existing systems. In new systems this will mean vulnerable pieces are placed underground where they are harder to reach for maintenance. Besides the response time for customers to receive warm water will increase meaning the overall quality of the product will decrease. These facts combined with the low increase in efficiency and decrease in heat loss result in a negative advise on relocating the location of the bypass flow valve to the end of the DN32 pipes.

Increasing the inlet temperature in summer to reduce the amount bypass flow in the system will also result in a slight decrease in heat loss. This can be achieved by changing the inlet temperature in the HTS. Because of the extra reduction of heat loss that can be accomplished this way it is recommended to experiment with this. Especially when the twin system is used because in that case it is best to change the inlet temperature also during peak-load in winter.

Financially the series 2 is not that more attractive then the series 1. Twin is the best choice with the biggest costs reduction being caused by the save on heat loss. Even taken in account the uncertainty of the price for the installation and maintenance, in the absolute worst case scenario with maintenance costs rising by 200% it is still the better choice.

10.2. Recommendation

Further research is recommended in addition to this report to get even better understanding on how the network is functioning and to further decrease the amount of heat lost during transport and in other parts of the system. In addition on the research done in this report and the conclusion that controlling the inlet conditions in a more passive way can have a positive influence on the reduction of heat loss it is recommended to further investigate and explore this. For example taking in account the daily or hourly fluctuation of heat demand which can fluctuate a lot especially during winter time.

To get an even better understanding on how the system works and how much heat is actually lost field research is needed. The research done during this project is still only theoretical, and so still a lot of assumptions are made. To know what is happening in reality field research can be done. This could be measurements in the ground to measure the actual ground temperature and to measure the increase in ground temperature caused by the heat loss.

Secondly a study can be done on the local conditions of the ground and the affect it has on the heat loss. As explained an average value is taken for the heat conductivity of the ground. These values will actually differ locally, and this will have an effect on the amount of heat lost in the system. Local and regional differences could also effect the choices that are made in the design of the system to get the optimal system for the local conditions.

At last research could be done to further improve the very low efficiency of the system in summer. For example in systems that only deliver heat and no tap-water the temperature of a 70-40 net is lowered in summer to a 50-30 net. In the systems with tap-water where the salmonella regulations need to be met this would mean the tap-water needs to be further heated by for example electricity or a heat pump. A hybrid solution like this could be an interesting solution for the problem of the very low efficiency in summer.

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A

Size Table Logstor Piping

The bonded pipe system
District heating pipes - Insulation series 1

Application

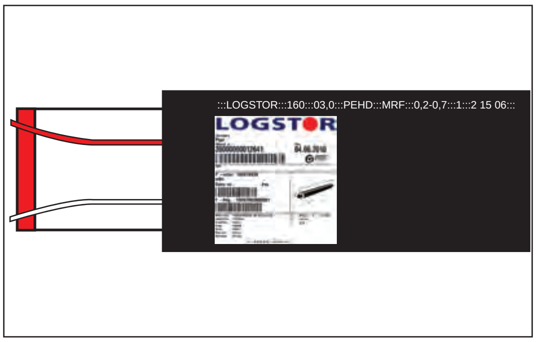
Preinsulated pipes of insulation series 1 are used for all common construction works where it is not necessary to make allowance for e.g. extreme outside temperature influences, especially high energy prices etc.

Description

A preinsulated pipe of insulation series 1 can be immediately identified by its pipe label, from which other data also appear, see page 1.3.0.2.

All preinsulated pipes are delivered with embedded copper wires for surveillance.

The dimensions ø 26.9/90 - ø 219.1/315 mm are available with diffusion barrier in 12 and 16 m lengths. See page 2.0.1.1.



The bonded pipe system
District heating pipes - Insulation series 1

Component overview/data

Component No. 2000

Steel pipe			Outer casing					Pipe	Water content
ø nom. mm	ø out. mm	Wall thick. mm	ø out. mm	Wall thick. mm	6 m pipe	12 m pipe	16 m pipe	Weight kg/m	l/m
20	26.9	2.6	90	3.0	x	x		2.9	0.4
25	33.7	2.6	90	3.0	x	x		3.3	0.6
32	42.4	2.6	110	3.0	x	x		4.2	1.1
40	48.3	2.6	110	3.0	x	x		4.6	1.5
50	60.3	2.9	125	3.0	x	x		6.1	2.3
65	76.1	2.9	140	3.0	x	x		7.5	3.9
80	88.9	3.2	160	3.0	x	x		9.4	5.3
100	114.3	3.6	200	3.2	x	x	x	14	9.0
125	139.7	3.6	225	3.4	x	x	x	16	14
150	168.3	4.0	250	3.6	x	x	x	21	20
200	219.1	4.5	315	4.1	x	x	x	31	35
250	273	5.0	400	4.8	x	x	x	45	54
300	323.9	5.6	450	5.2		x	x	58	77
350	355.6	5.6	500	5.6		x	x	66	93
400	406.4	6.3	560	5.7		x	x	81	120
450	457	6.3	630	6.0		x	x	93	160
500	508	6.3	710	6.6		x	x	108	190
600	610	7.1	800	7.8		x	x	142	280
700	711	8.0	900	8.7		x	x	180	380
800	813	8.8	1000	9.4		x	x	230	500
900	914	10.0	1100	10.2		x	x	280	630
1000	1016	11.0	1200	11.0		x	x	340	780
1100	1118	11.0	1300	11.8		x	x	378	943
1200	1219	12.5	1400	12.5		x	x	460	1120

The bonded pipe system
District heating pipes - Insulation series 2

Application

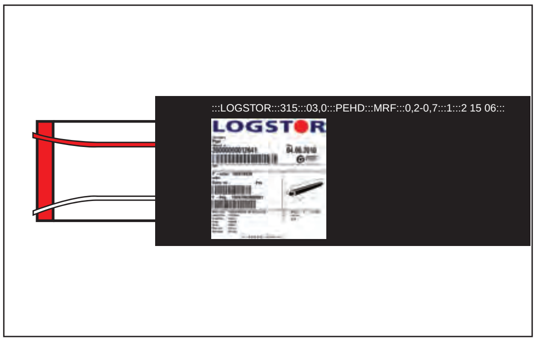
Preinsulated pipes with extra insulation thickness, series 2, are used where there are special temperature conditions such as constant low ambient temperatures, constant high media temperatures, demand for slow cooling at shutdown, high production costs on the energy side etc.

Description

A preinsulated pipe of insulation series 2 can be immediately identified by its pipe label, from which other data also appear, see page 1.3.0.2.

All preinsulated pipes are delivered with embedded copper wires for surveillance.

The dimensions ø 26.9/110 - ø 168.3/280 mm are available with diffusion barrier in 12 and 16 m lengths. See page 2.0.1.1.



Component overview/data

Component No. 2000

Steel pipe			Outer casing					Pipe	Water content
ø nom. mm	ø out. mm	Wall thick. mm	ø out. mm	Wall thick. mm	6 m pipe	12 m pipe	16 m pipe	Weight kg/m	l/m
20	26.9	2.6	110	3.0	x	x		3.3	0.4
25	33.7	2.6	110	3.0	x	x		3.7	0.6
32	42.4	2.6	125	3.0	x	x		4.6	1.1
40	48.3	2.6	125	3.0	x	x		5.0	1.5
50	60.3	2.9	140	3.0	x	x		6.5	2.3
65	76.1	2.9	160	3.0	x	x		8.0	3.9
80	88.9	3.2	180	3.0	x	x		10	5.3
100	114.3	3.6	225	3.4	x	x	x	15	9.0
125	139.7	3.6	250	3.6	x	x	x	18	14
150	168.3	4.0	280	3.9	x	x	x	23	20
200	219.1	4.5	355	4.5	x	x	x	34	35
250	273	5.0	450	5.2	x	x	x	49	54
300	323.9	5.6	500	5.6		x	x	63	77
350	355.6	5.6	560	5.7		x	x	70	93
400	406.4	6.3	630	6.0		x	x	89	120
450	457	6.3	710	6.6		x	x	104	160
500	508	6.3	800	7.2		x	x	120	190
600	610	7.1	900	7.9		x	x	156	280

TwinPipes

Preinsulated TwinPipes, series 1, 2, and 3

Application

Preinsulated TwinPipes are used for all common construction works and for systems with reduced trench width.

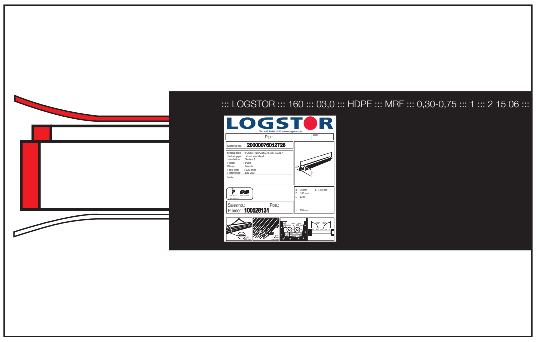
Applicable for installation methods: Preheating or high axial stress installation.

Description

A preinsulated TwinPipe series 1, 2 or 3 can be identified by its label, from which other data also appear. See page 1.3.0.2.

In all preinsulated TwinPipes copper wires for surveillance are embedded.

The dimensions ø 125-315 mm in series 1, ø 140-280 mm in series 2, and ø 160-315 in series 3 are available with diffusion barrier in 12 m or 16 m length, see page 2.0.1.1.



Materials

TwinPipes are manufactured according to the same specifications as for other straight pipes.

Component overview/Data Series 1

Component No. 2090.

Steel pipe			Outer casing		Distance between steel pipes	Delivered length			Weight	Water content
ø nom mm	ø out. mm	Wall th. mm	ø out. mm	Wall th. mm		6 m*	12 m	16 m	kg/m	l/m
20	26.9	2.6	125	3.0	19	x	x		5.2	0.7
25	33.7	2.6	140	3.0	19	x	x		6.5	1.3
32	42.4	2.6	160	3.0	19	x	x		8.1	2.1
40	48.3	2.6	160	3.0	19	x	x		8.8	2.9
50	60.3	2.9	200	3.2	20	x	x		12.4	4.7
65	76.1	2.9	225	3.4	20	x	x		15.4	7.8
80	88.9	3.2	250	3.6	25	x	x		19.5	10.7
100	114.3	3.6	315	4.1	25	x	x	x	28.4	18.0
125	139.7	3.6	400	4.8	30	x	x	x	38.2	27.6
150	168.3	4.0	450	5.2	40	x	x	x	49.4	40.4
200	219.1	4.5	560	6.0	45		x	x	72.5	69.3

* 6 m TwinPipes are produced traditionally.

B

Heat Transfer Station(HTS)

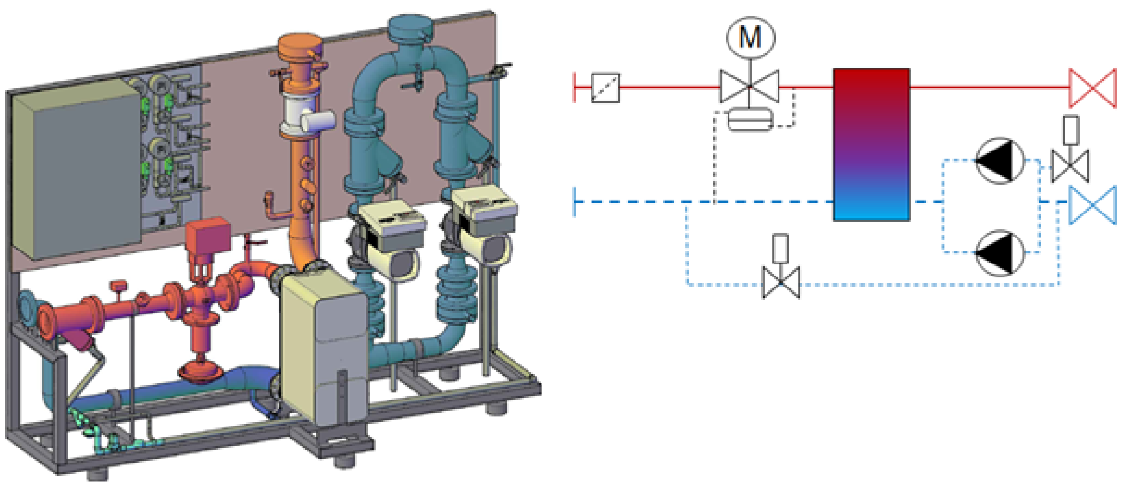
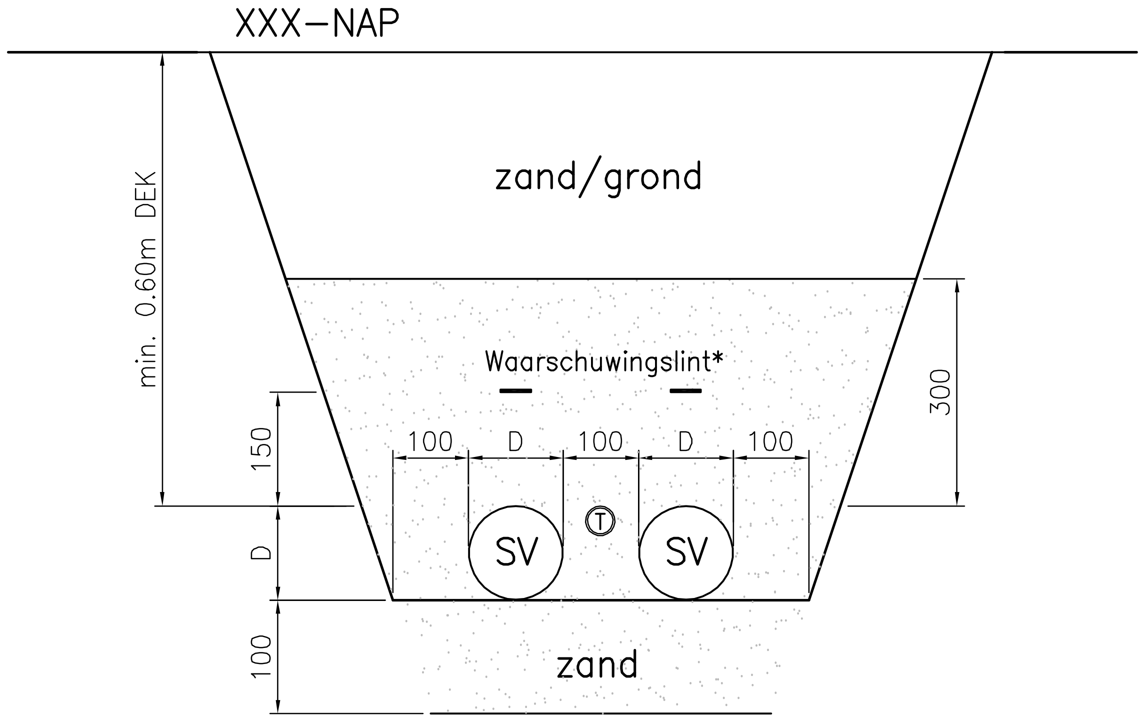


Figure B.1: Heat transfer station.

C

Side view of the pipes in the ground.

SLEUFINDELING T.B.V PB OF PE LEIDINGEN



* afhankelijk van de locatie van de leidingen moet het lint vervangen worden door afdekband. zie voorschrift 'afdekken leidingen'

ENGINEERINGSBUREAU
PLAATS
TEKENAAR (TEL.NR.)

Amerikaanse projectie

Schaal: – Formaat: A4 Afdeling: TECHNIEK

Rev.	Wijziging	Datum	Get.
H			
G			
F			
E			
D	Hoogte zandpakket	20-04-15	PvdB
C	Verwijz. voorschrift	07-08-14	SB
B	Maten veranderd	11-01-11	BV
A	zandpakket	19-12-08	GB

	Datum	Naam
Get.	20-04-2015	
Gec.		
Gez.		

STANDAARD SLEUF T.B.V
KUNSTSTOF LEIDINGEN

NUON Warmte Techniek & Innovatie

TG010

Blad 001

Doorkiesnr.:

Doc.nr.:

Verv.:

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D

Numerical results

D.1. Series 2

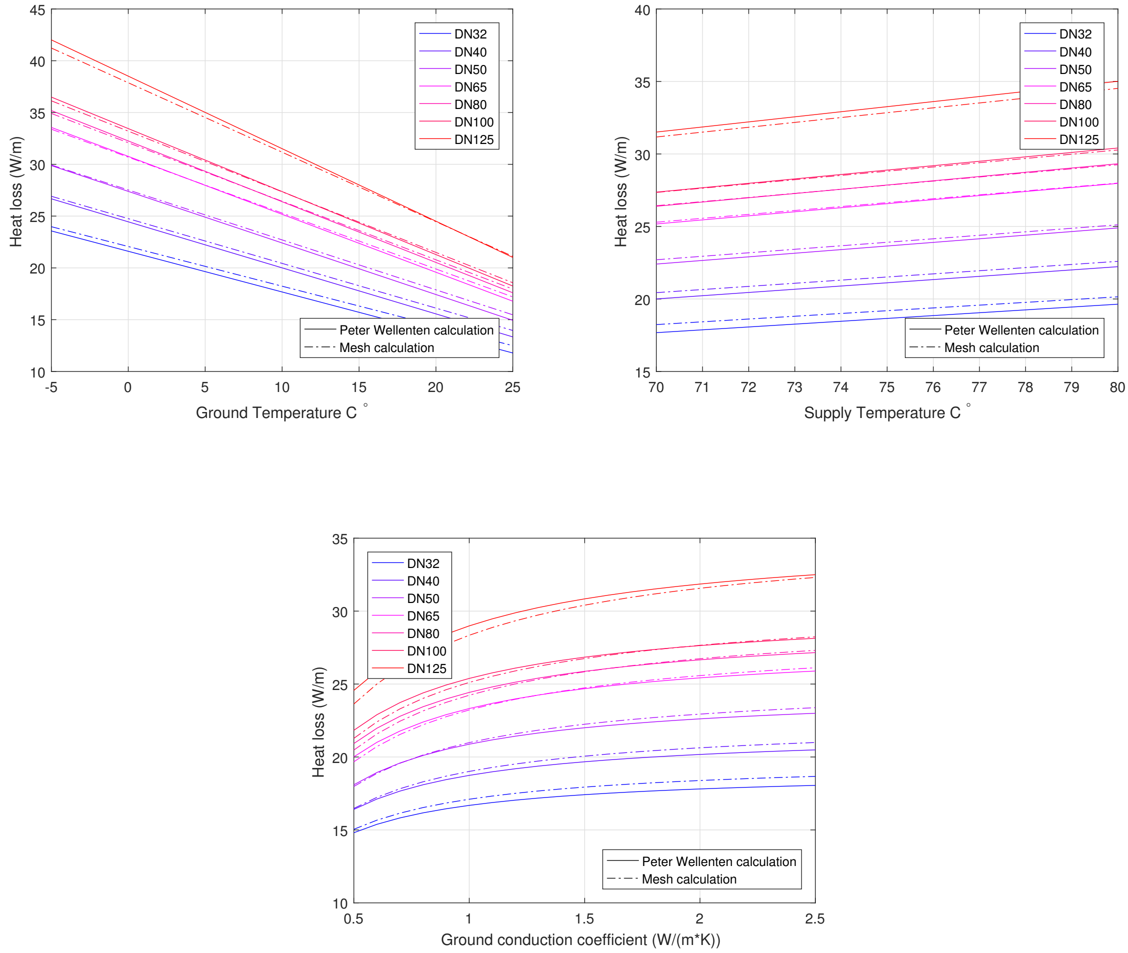


Figure D.1: Difference between the numerical and empirical heat loss calculations varying: ground temperature (top left), supply temperature(top right) and ground conductivity (bottom).

D.2. Twin

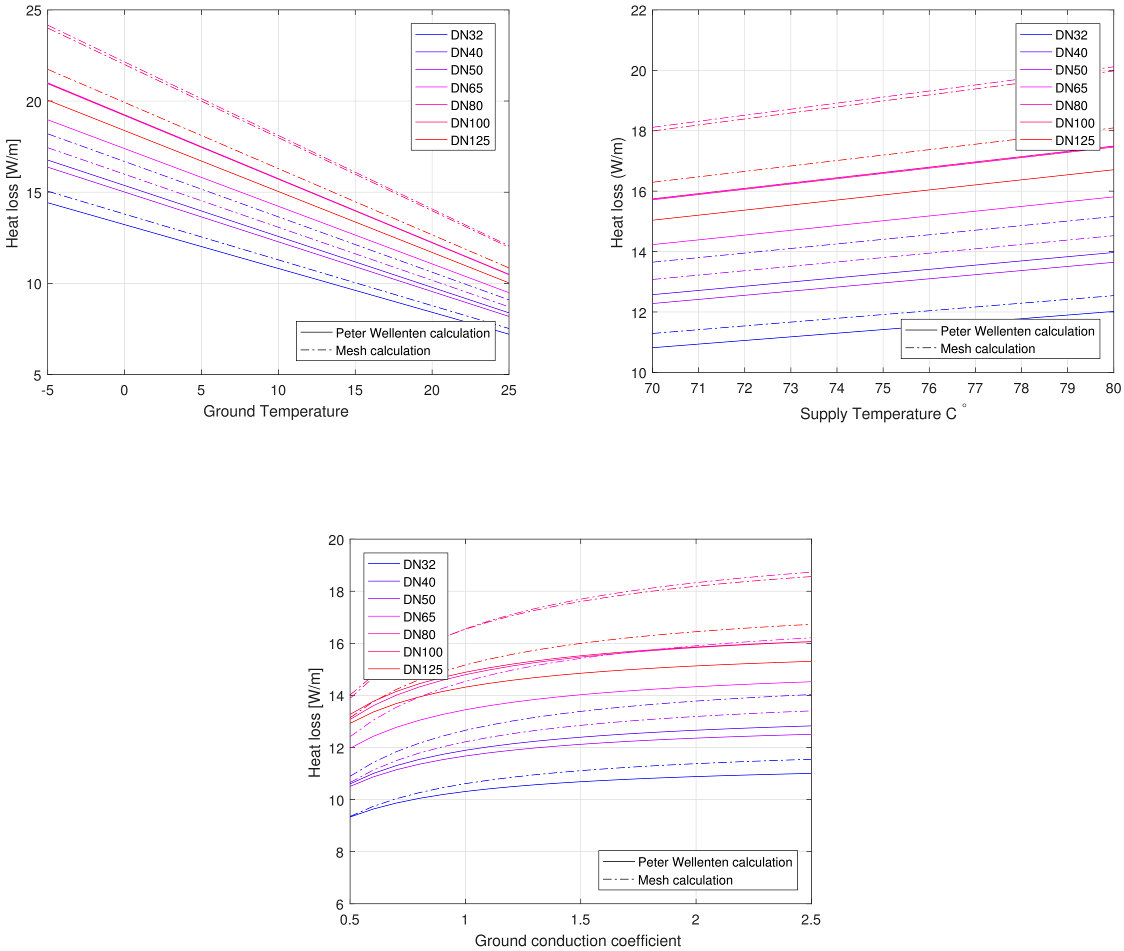


Figure D.2: Difference between the numerical and empirical heat loss calculations varying: ground temperature (top left), supply temperature(top right) and ground conductivity (bottom).

E

Pressure triangle

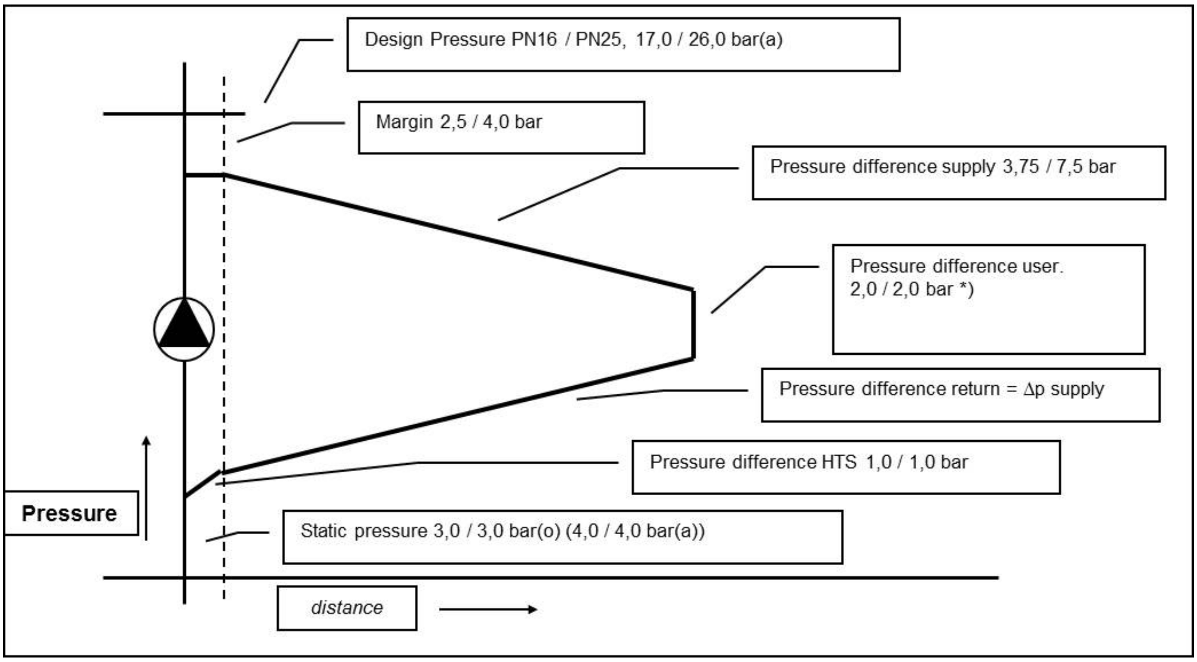


Figure E.1: Pressure triangle.

F

Pump Curve

Wilo Stratos GIGA 65/1-21/2,3,

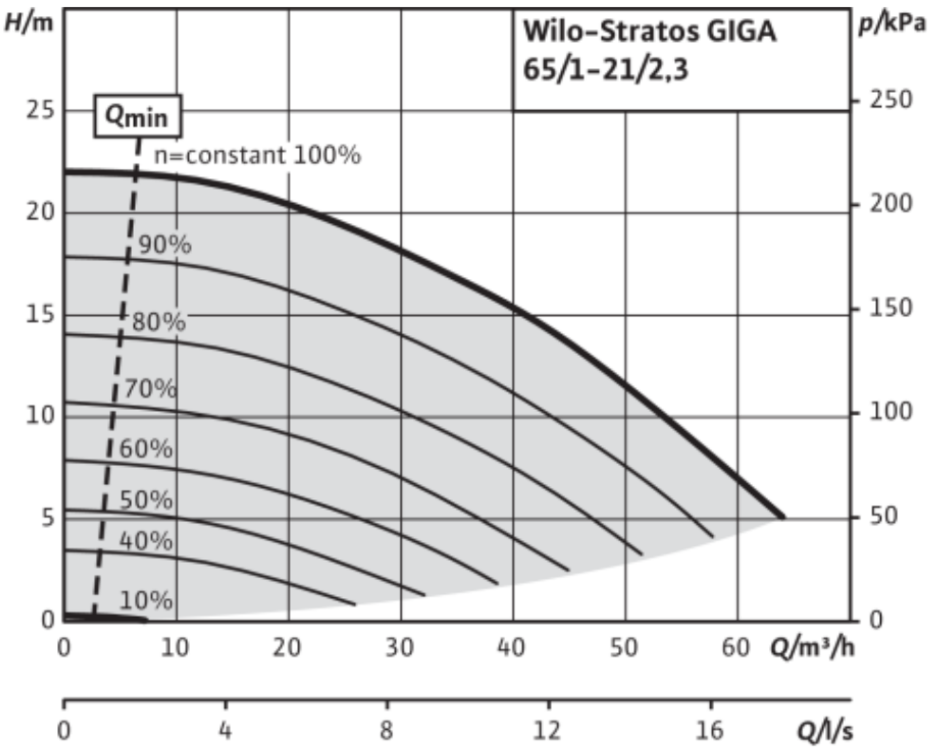


Figure F.1: Pump curve of the Wilo Stratos GIGA 65/1-21/2,3.

G

Ground

Two formulas can be found that describe the ground temperature at a certain depth. These are given by Baggs[10] and Kasuda[11]. The formula given by Baggs is developed for the ground temperature in Australia and is checked using measurements from a research done in the region of Poznan, Poland by Popiel [10]. The formula of Kasuda which is developed using measurements from different stations over the united states showed good agreement with experimental data from measurements done by Georgios on Nicosia, Cyprus[9]. Because the formula given by Kasuda is easier to interpret and probably less developed for a specific area or ground type that is the one used to predict the ground temperature.

The formula of Baggs is given as:

$$T(x,t) = (T_m + \Delta T_m) - 1.07 k_v A_s exp(-0.00031552 x a^{-0.5}) * cos(2\pi/365(t - t_0 + 0.018335 x a^{-0.5}). \quad (G.1)$$

$$T_{soil(D,tyear)} = T_{mean} - T_{amp} * exp(-D\sqrt{\frac{\pi}{365\alpha}} * cos\frac{2\pi}{365}(t_{year} - t_{shift} - D/2\sqrt{\frac{365}{\pi * \alpha}})) \quad (G.2)$$

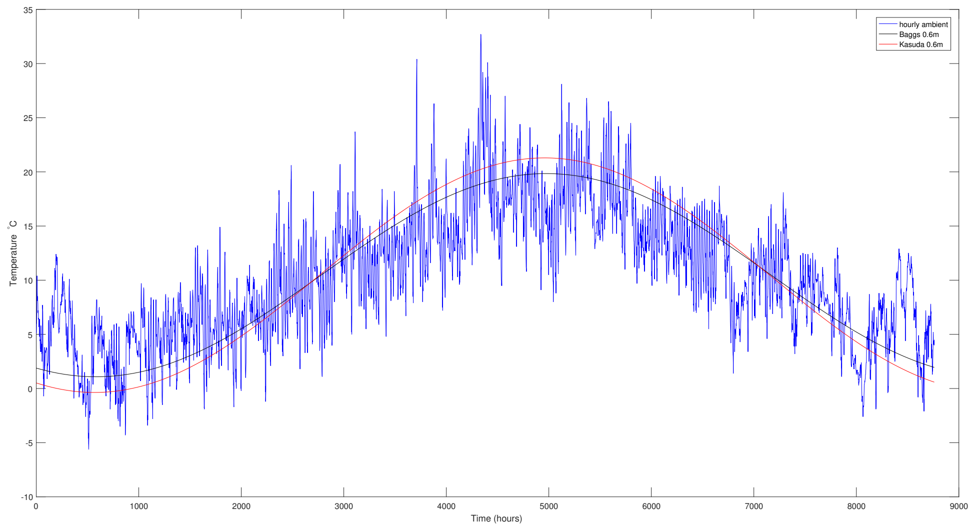


Figure G.1: Daily ambient temperature plot with both Kasuda and Baggs prediction fot the temperature.

H

Kasuda

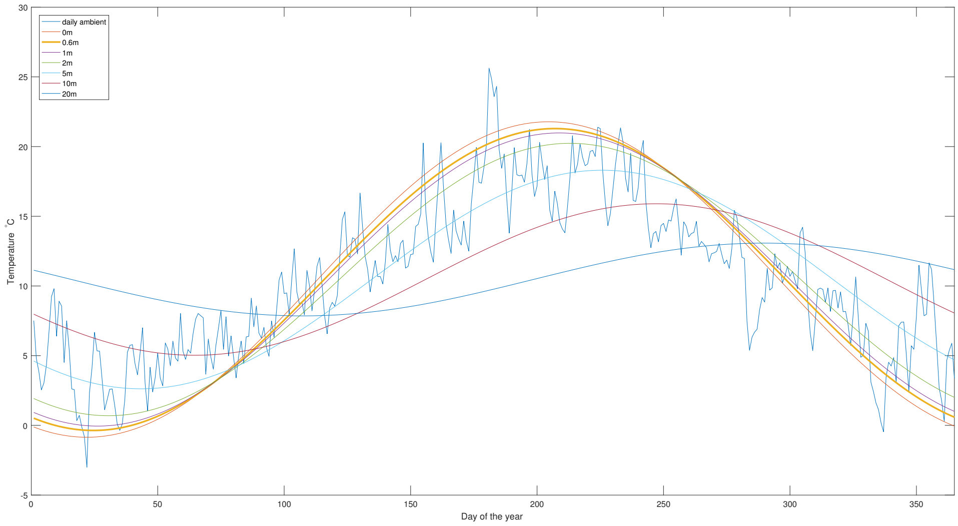


Figure H.1: Temperature predictions done different depths with the formula of Kasuda at.

I

TCO

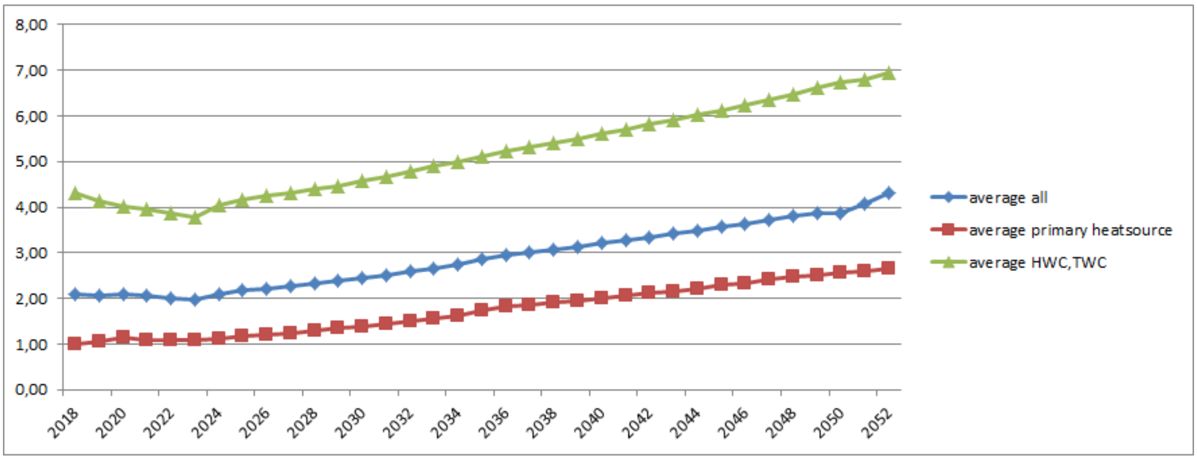
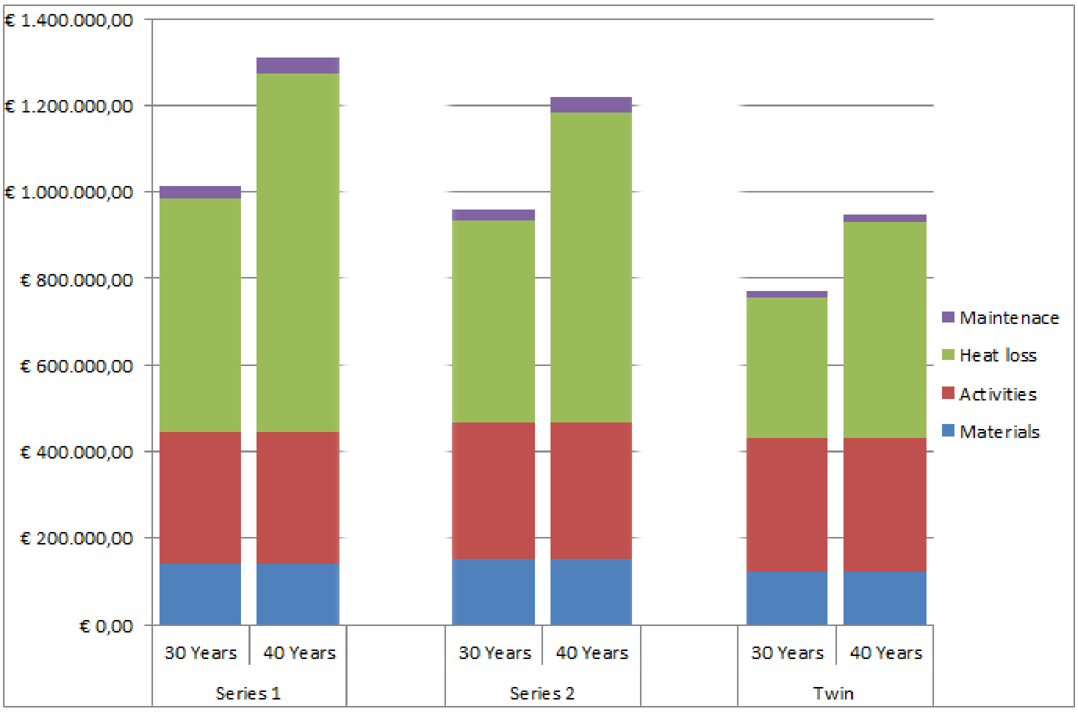


Figure I.1: Normalized prize development of the cost of heat for a gigajoule of heat.



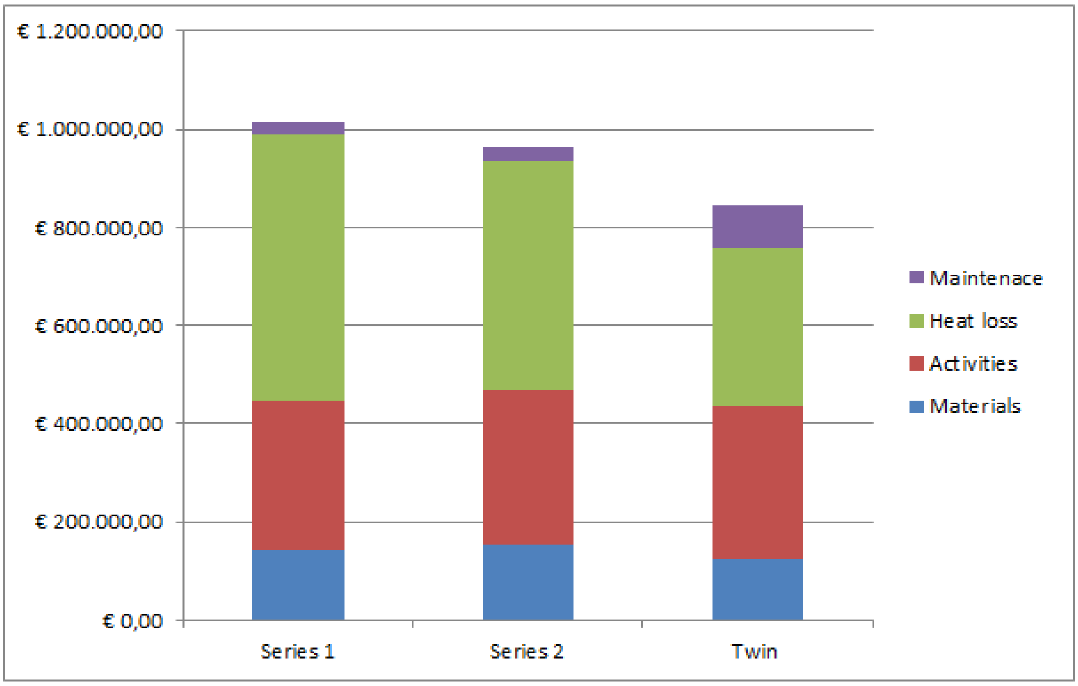


Figure I.2: Bar graph of the TCO, difference between 30 and 40 year lifetime(top) and 200% increase in maintenance costs(bottom).