A technical design framework for cold heating and cooling networks.

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1. Introduction

One of the aims of the Dutch Climate Agreement is that by 2050 7 million residential buildings and 1 million other buildings will be closed off from natural gas as an energy source for heating. This means that natural gas, a high quality energy source (high exergy), makes room for an alternative energy source to meet the low quality energy (low exergy) demand in the built environment.

A cool heating network (5GDHC or energy exchange network) is inspired by the "low exergy" vision of decarbonizing the thermal energy use of the built environment, based on maximal use of low grade thermal sources to serve the low grade thermal needs of heating and cooling.

Many new energy concepts developed in this context assume an all-electric solution with solar panels, infrared heating and/or heat pumps. Experience has learned that these concepts are developed for low-rise buildings and are difficult to adept to high-rise buildings, such as apartment blocks, are frequently occurring in densely built urban areas. Furthermore, due to their design or monumental status, many buildings build before the Second World War are difficult to renovate to enable an all-electric heating solution. An alternative solution for a natural gas free heat supply is needed.

Existing heat networks in The Netherlands are mostly based on high temperatures and high quality energy sources , fed by sources (e.g. waste incinerators, fossil power plants) that are highly likely to disappear in the future. In a society without fossil fuels, high temperature sources are rare in the Netherlands, making it difficult to maintain the business as usual for heat networks.

However, there still is a significant amount of untapped low temperature industrial waste heat that could potentially be used for heating purposes. In the Netherlands, data centres alone could provide about 1.5 TWh per year in waste heat [1], enough to heat to about 150.000 households.

Additionally, higher insulation standards in residential and utility buildings decrease the heat demand in winter and increase on the other hand the cooling demand in summer. Thus, there is a need for heating and cooling solutions.

This report presents a technical design framework for a thermal network that provides both heat and cold energy to customers and enables tapping of low temperature (industrial waste) energy sources.

1.1 Reading guide

The conceptual design of a (low temperature) heating and cooling network is discussed in chapter 2, describing the definitions and setting the scope of the technical design framework.

Engineering principles of heating and cooling networks, such as pressure drop and thermal losses, are introduced in chapter 3. These principles provide the base for making design choices, such as pipe and pump sizing. The derivation of equations in this chapter are found in chapter 9.

Network topologies are discussed in chapter 4. Network components for mass balancing, energy balancing, network connections and heat interface units at a customer's premises, are described in chapter 5. Operational concepts, such as energy balancing, storage sizing and production sources, are reviewed in chapter 6.

All the above-mentioned concepts, designs and principles come together in chapter 7, where a step-by-step design framework is presented.

References to used literature sources can be found in chapter 8.

A select of heat network pipe data can be found in chapter 10.

2. Conceptual design

This chapter outlines the concepts for the framework design of the heating-cooling network. It starts with a technical definition which is examined in the upfollowing paragraphs. This chapter provides the line of thinking used to create this framework and states its base principles. Refinement of the framework components is found in following chapters.

2.1 Technical definition

A cool heating network is based on the bidirectional exchange of thermal energy between buildings with different load profiles maximizing the share of low grade renewable and waste energy sources. Active and distributed energy substations upgrade the required temperatures in the buildings minimizing the input of external high grade energy. Temporal fluctuations in the supply and demand of heat and cold are buffered by storage at various time and space scales. The demand driven network aims to have zero carbon emissions.

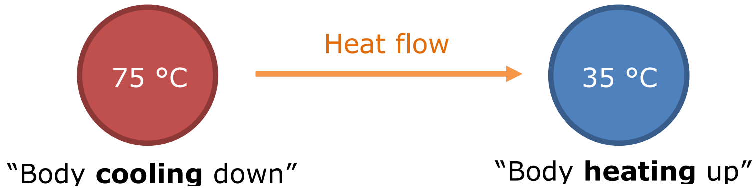
2.2 Heating and cooling

Thermal energy is the energy contained within a system or body that is responsible for its temperature. The higher the body's temperature, the more thermal energy it contains. **Heat** is the spontaneous flow of thermal energy from a body with a higher temperature to one with a lower temperature. As the amount of thermal energy reduces in the higher temperature body, its temperature decreases. Similarly, the thermal energy added to the lower temperature body, increases its temperature. The spontaneous flow of heat stops when the temperatures of the two bodies have equalized.

In a district heating network, hot water is provided to customers through a district or city-wide network of pipes. Heat flows from the hot water into the buildings. The transfer of thermal energy from the network to the buildings increases the temperature in the buildings and decreases the temperature of the water in the network.

In a district cooling network, the opposite occurs. Cold water is provided to customers and the heat flows from the buildings into the network, cooling the building down and heating the water in the network up. Although there is still a heat flow, this process is referred to as cooling.

The terms heating and cooling are used depending on the perspective. If the reference frame is a body of which the temperature is increased by a heat flow, the term heating applies, but if the temperature of that body is lowered by a heat flow, the term cooling applies. Thus, the only difference between heating and cooling is the direction heat flow with respect to a reference frame or body.



Such strict physics definitions are not commonly used in practice. A district heating network simply provides heating, a district cooling network provides cooling and the reference frame is always the end customer. But this changes when one network provides both heating and cooling.

In a heating network, the temperature of the return flow is lower than the temperature of the supply flow, as thermal energy has been transferred from the supply flow to the customer. A traditional district heating network typically has a supply temperature of around 100 °C and a return temperature of 70 °C.

When the supply temperature of the heat grid would be lowered to 25 °C, the return flow reaches a temperature of about 15 °C or lower. At that point, the return flow could be used for (indirect) cooling. This cooling process could bring the temperature back up to 25 °C and used again for supplying heat.

This creates a system in which a customer demanding 'heat' from the network, simultaneously produces 'cold' for that network. The reverse is also true, a customer demanding 'cold', produces 'heat' at the same time. It's all just a transfer of thermal energy through a heat flow. Some customers require thermal energy, others want to dispose of it. The result: one network that provides both district heating and cooling simultaneously.

2.3 Two-pipe system

The traditional two-pipe system can provide heating and cooling as depicted in Figure 2.1. One pipe provides heating ('heat-pipe'), the other one provides cooling ('cold-pipe'). But there is no return pipe. Each pipe functions as the return of the other. When a consumer needs heating, it receives warm water from the heat-pipe. The water cools down because of extraction of thermal energy, after which the water is returned into the cold-pipe. This process is reversable. A consumer that requires cooling receives cold water, adds thermal energy to the water and returns the warmed-up water into the heat-pipe.

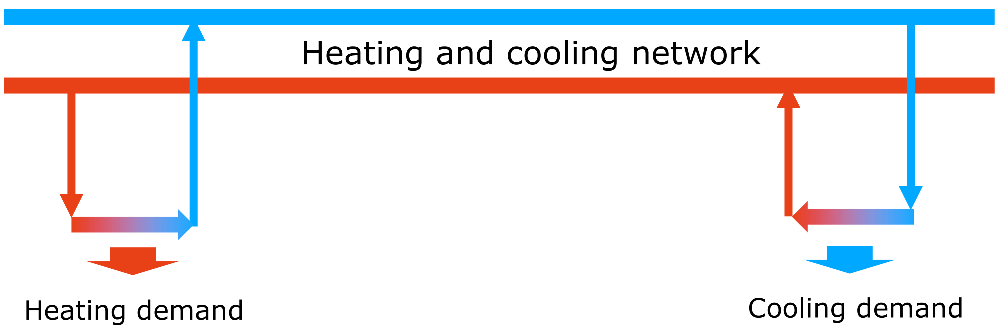


Figure 2.1: Outline of a two-pipe district heating and cooling network with a simultaneous heating and cooling demand.

There are several advantages to this design. Traditional combined district heating and cooling network commonly use a 4-pipe system, where heating and cooling each have their own supply and return pipes. This doubles the amount of piping required and thus is (at least) twice as expensive than a 2-pipe district heating and cooling network.

Another advantage is the availability of cooling. Many district heat networks don't offer cooling during the summer months, despite the increasing demand for cooling in residential buildings, due to better insulation standards. With the district heating and cooling network, customers always have the availability of heating and cooling. The flow direction determines whether the customer is heating or cooling. When the water flows from heat-pipe to cold-pipe, the customer is provided with heating, if the water flow is reversed, the customer is provided with cooling.

For example, take a well-insulated house in the summer. In the early morning, heating is required to make domestic hot water, so the water flows from heat-pipe to cold-pipe. In the afternoon, the house warms up and requires spatial cooling. The water then flows from cold-pipe to heat-pipe. And in the evening, more domestic hot water may be needed, so the flow is reversed again.

The third advantage of the two-pipe system is that less thermal energy is transferred to higher level networks (e.g. a transmission network or backbone), as there is a (partial) match between heating and cooling. This means the capacity of the higher-level network and substations could be designed smaller, saving significantly on investment costs. The need for a top-down investment structure is broken and decentralized investments are enabled.

2.4 Temperature levels

2.4.1 Active heating and cooling

Passive heating is generally feasible with a supply temperature of at least 35 °C, while passive cooling is feasible with a temperature of at most 15 °C.

The potential of untapped industrial waste heat is mainly found at temperatures between 20 °C and 35 °C. If the temperature of the waste heat is lower than that of the heat pipe, it requires an upgrade. The larger the temperature gap between the waste heat and the heat-pipe, the costlier it is to unlock the potential.

Optimally, the heat-pipe has a temperature equal to the temperature of the waste heat. This allows the waste heat to be fed in the network directly and saves on the investment of upgrading the temperature. It does however make the network dependant on this heat source. This becomes an issue when the source disappears, or when there are multiple sources with different waste heat temperatures available for the network.

Furthermore, the provider of the waste heat does not have control over the return temperature. The waste heat provider may need the return flow for cooling, requiring an upper limit on the return temperature.

The dependency on the flow's temperature can be negated by using a heat pump at every customer connection. The heat pump provides the temperature - either for heating or cooling - needed by the customer, regardless of the temperature of the heating/cooling network. Any change in network temperature will not be noticed by the customer.

As such, every customer, whether it is residential, commercial or industrial, has a heat pump providing active heating and cooling. With the heat pumps exchanging thermal energy through the heating/cooling network, their effectiveness (coefficient of performance) is very high.

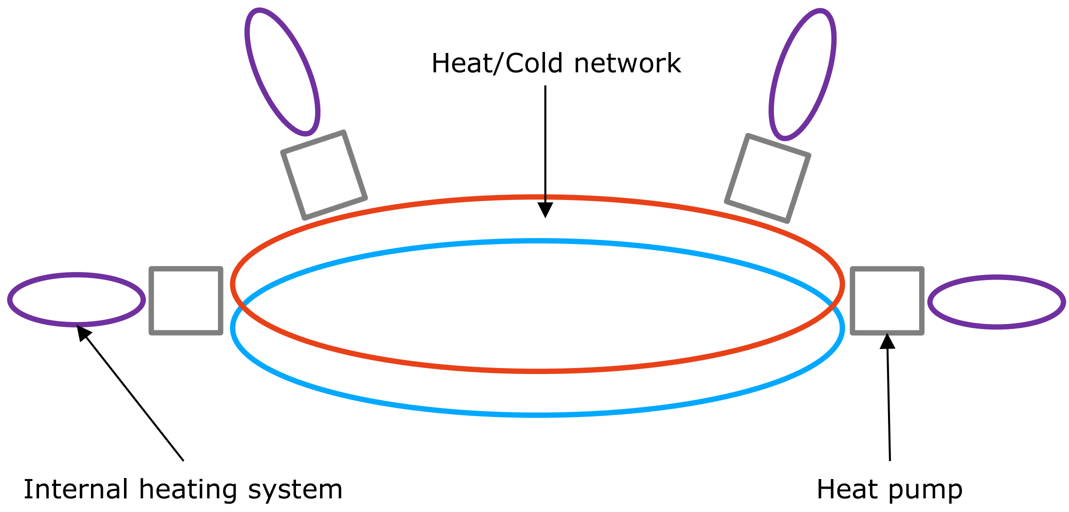


Figure 2.2: Every customer in the network has a heat pump connecting the heat/cold network with the internal heating and/or cooling system of the customer.

2.4.2 Temperature difference

The temperature difference between the heat-pipe and cold-pipe has a direct impact on the thermal capacity of the network, i.e. the amount of thermal energy that is transferred per unit of time. The smaller the temperature difference, the lower the thermal capacity.

Heat pumps ensure that the difference is overcome. However, the effectiveness of heat pumps is directly related to the size of the gap. The larger the gap, the lower the effectiveness.

This would favour a small temperature difference during normal operations and an increased temperature difference in times of peak demand, sacrificing a small bit of heat pump effectiveness in favour of additional thermal capacity.

Widening the temperature gap means that the temperatures in the heat-pipe and/or cold-pipe change. With a fixed pipe temperature, thermal capacity cannot be (significantly) increased.

2.4.3 Conclusion

When it comes to temperature levels, low temperature district heating and (high temperature) district cooling are characterised by:

- The temperatures in the heat-pipe and cold-pipe are not (pre)defined. They may fluctuate.
- The temperature in the heat-pipe is likely 35 °C with a maximum of 50°C. This is not a hard limit nor guaranteed.
- The temperature in the cold-pipe is likely to be above 15 °C. This is not a hard limit nor guaranteed.
- The temperatures in the pipes (can) vary in time, daily and seasonal.
- Every customer has a heat pump, separating temperature levels at the network side from the customer's wishes.
- Networks are designed based on the temperature difference between the heat-pipe and cold-pipe.
- The temperature difference between the heat-pipe and cold-pipe is not constant. Extra thermal capacity can be provided by temporarily increasing the temperature difference while accepting a small penalty on heat pump effectiveness.
- A temperature difference of 10 °C during normal operations seems to be a good trade-off between network capacity and heat pump effectiveness

2.5 Bottom-up approach

Traditional heating (and cooling) networks are built around a few large production plants (e.g. boilers, waste incinerators, power plants) feeding directly into a primary network (sometimes called 'transmission network' or 'backbone'). Vast amounts of hot water are transported to multiple districts in a city and nearby villages. Through substations and distribution networks, the hot water is distributed to the end customers. The design principles of traditional heat networks are thus based on a top-down and one-directional approach. Heat sources such as boiler plants and incinerators are at the top, from where heat is transported and distributed to the heat demanding customers (Figure 2.3).

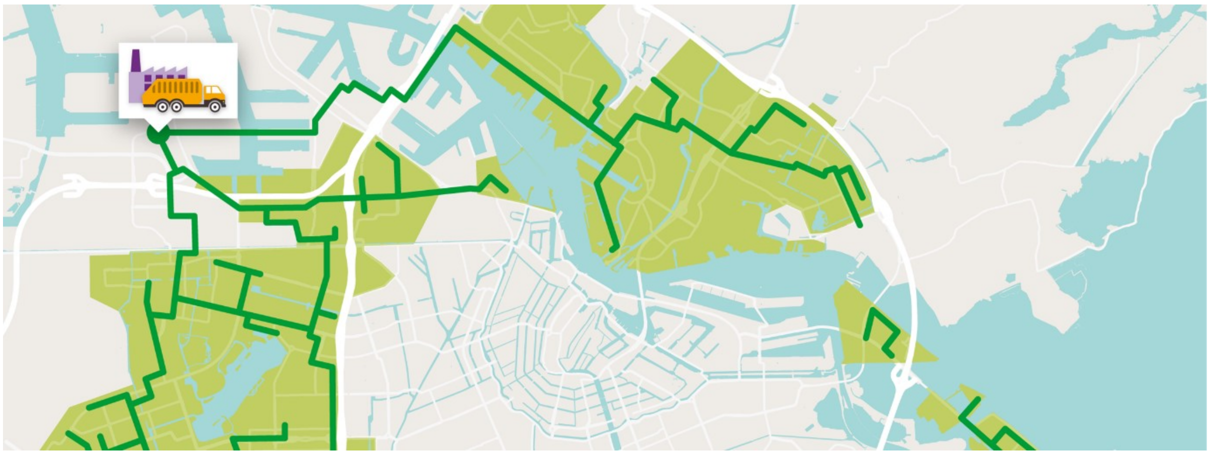


Figure 2.3: District heating in North-West Amsterdam, provided by an incinerator. (source: Nuon)

In a low temperature heating and cooling network, a cooling demand can be met by a heating demand. As any customer that demands heating, is a supplier of cooling and vice versa, matching customers locally is the key to success. It makes central heating and cooling plants less important, or even redundant. Thus, designing a heating and cooling network starts with matching customers locally instead of looking from a central holistic view. Starting from the bottom instead from the top.

Because matching and joining customers in a district heating and cooling network is a local process, it can be initiated in multiple districts at the same time. As opposed to traditional heat networks, no comprehensive design and architecture of the entire future city-wide system is required.

Once the local networks have been realized, further optimisation may take place by connecting the individual networks, increasing performance and energy efficiency, reducing operational costs and opening opportunities for other districts and nearby cities to join the system.

2.6 Decentralized operations

Traditional heat networks have centralized operations: a single pumping station that ensures everyone gets enough flow to fulfil their heating demand using passive heat exchangers at the end users. This is difficult to realise in a modular and decentralized network for several reasons.

First, the size of the pump(s) is decided at design time. With future expansions of the network expected, two approaches can be taken.

In the first approach, the pumps are over dimensioned right at the start of the project. When future expansion occurs, the right pump size is already in place. This however requires higher investments at the start. And that comes with additional financial risks. What if the future expansion doesn't take place?

In the second approach, components like pumps are replaced by bigger versions once the expansion takes place. This means less financial risk earlier in the project, but every expansion will lead to disinvestments. This makes expansions costlier than it should be.

Another problem with centralized operations is that the pump(s) are always running to create a constant pressure, as it can't detect whether customers require heating.

But the most important issue with centralized operation is the complexity of a 2-pipe district heating and cooling system: the flows are bidirectional. A pump cannot put pressure on the heat-pipe, as it prevents the flow going back from the cold-pipe to the heat-pipe.

Instead, a decentralized pumping system for the heating-cooling networks is proposed. Each customer connection has its own bi-directional pump. If the customer has a heat demand, the pump creates a flow from the heat-pipe into the cold-pipe. If there is a cold demand, the pump creates a flow the other way. If there is no demand, the pump shuts down.

Each pump is designed for the customer. If a customer requires more heat or cold, the thermal capacity can be increased, within the limits of the network, by installing a larger pump.

The design of the customer connection point (the heat interface unit) is further discussed in section 5.3. The sizing of the pumps is discussed in section 3.5.

2.7 A new generation

With the scope and boundaries laid out for a novel low-temperature district heating and cooling network, a new generation of thermal networks is born: the fifth generation of district heating (and cooling). Fifth generation networks are characterized by a heating supply temperature below 30 °C and a decentralized approach. All five generations of heat networks are shown in Figure 2.4.

	1st gen	2nd gen	3rd gen	4th gen	5th gen
Heat carrier	Steam	Pressurized water	Pres surized water	Water	Water
Indicative temperature	150 - 200 °C	100 - 140 °C	70 – 100 °C	35 - 70 °C	< 35 °C
Control parameter	Pressure	Pressure	Supply temperature	Supply temperature	Temperature difference
Circulation system	Steam pressure	Central pumps	Central pumps	Central and decentralized pumps	Decentralized pumps
Energy efficiency	Low	Mediocre	Mediocre	High	Very high
Cooling	No	No	No	No	Yes
Best available	1880-1930	1930-1980	1980-2020	2020-2050	In development

Figure 2.4: Infographic of generations of district heat networks. The information is partially derived from [2].

Because of the decentralized approach, fifth generation networks are more flexible. They can be extended or connected with other networks more easily. This is further achieved by using a modular design for the network and network components. Flexibility and modularity ensure that new customers can be cost-effectively connected as no disinvestments or re-engineering is required. It also enables fast upscaling as standardized components reduce the amount of engineering.

As customers play the key role in fifth generation networks, they provide and consume heat and cold, the network is open. There is no monopoly that produces, sells and distributes the thermal energy. Instead a fair and competitive market is created. Because of the competitiveness, sustainable sources, such as waste and renewable heat, gain preference over non-sustainable sources (e.g. fossil fuels, waste incineration).

In an open network, a customer can be a supplier of thermal energy, a consumer of thermal energy, both supplier and consumer of thermal energy, or providing thermal energy services such as thermal energy storage.

3. Network design principles

This chapter introduces design principles used in the engineering of heat networks and are applied to a low temperature district heating and cooling network. The information in Frederiksen and Werner [3] has been used as a base. Derivations of equations can be found in chapter 9.

3.1 Thermal Power demand

Traditional heat networks are sized based on the aggregated heat demand curve of its customers. Aggregated demand curves can be calculated without having to know the demand curves of individual customers. As individual behaviour is averaged out by aggregation, statistics are used to predict the aggregated demand curve relatively accurate.

The highest demand in such an aggregated curve determines the thermal power of the distribution network and substations. Further aggregation towards transmission level determines the thermal power of the transmission network and production units.

Sizing traditional district heating systems is therefore relatively easy. However, this is not the case for low temperature district heating and cooling networks.

As thermal energy is exchanged locally between customers that demand heating and cooling, an aggregated demand curve is not only more difficult to calculate, it may also underestimate the thermal capacity of the network.

Let’s say there is a demand for 10 MW in heating and 8 MW in cooling continuously in an arbitrary network. To achieve a heat balance, 2 MW of cooling must be provided at substation level. In traditional heating grids, the network would be dimensioned at 2 MW of thermal power as a top-down structure is assumed. However, with local exchange taking place, there is a thermal energy flow up to 10 MW in the network. The network needs to be able to accommodate this flow too. While the substation may be sized for 2 MW, the network itself needs a thermal power of 10 MW.

The easiest way to determine the required thermal power of the network, is to assume the worst-case scenario: all the cold consumers are on one side of the network and all the heat consumers are on the other side of the network. The peak demand in the aggregated cooling demand and aggregated heating demand curves are determined. The highest of the two peaks equals the required thermal power of the network.

The above method is near optimal if there is no customer diversity in the network. However, in networks with a high diversity of customers, this method could lead to significant overengineering of the network.

The optimal way of determining the required thermal power of the network is to model the thermal demand curve for every individual customer connection. A thermal demand curve is the aggregation of the heating demand curve subtracted by the cooling demand curve as shown in Figure 3.1.

When the thermal demand curves for all customers are known, the thermal flows in the network can be determined for each of the time intervals. The section with the highest thermal flow across all time intervals then equals the required thermal power of the network.

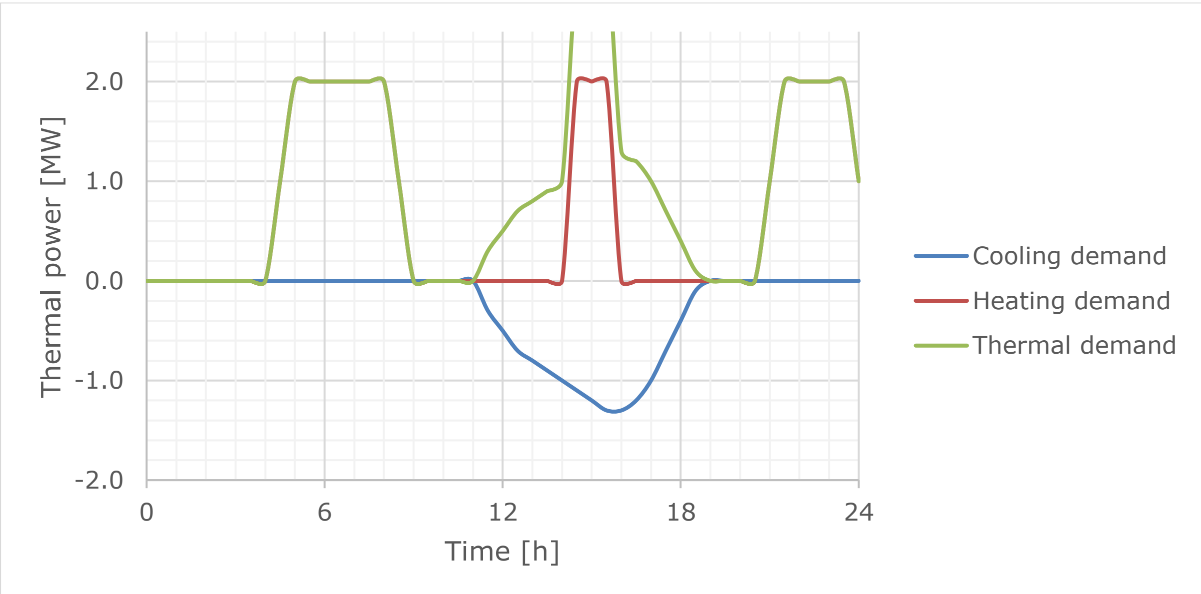


Figure 3.1: Demand curves for heating (red), cooling (blue) and the aggregated thermal demand (green) for an individual customer. Cooling has a negative value as it is thermal power directed in the other way.

3.1.1 Thermal power in residential area

The thermal power for a residential area is determined using the specific heat loss rate (expressed in Watt per Kelvin) of each building. This quantity relates the aggregated thermal losses (transmission, infiltration, ventilation) to the temperature difference between inside and outside the building. The specific heat loss rate is calculated with building physics programs, such as Energeyes [10]. Typical values for the specific heat loss rate are 40 W/K for a well-insulated house and 400 W/K for a poorly insulated terraced house.

Say that in the worst-case scenario, the temperature of the building should be maintained at 20 °C, while it is -10 °C outside, a difference of 30 K. A poorly insulated house then requires 12 kW of continuous heating. When there are 100 similar houses in the district, the local heating grid needs to be able to provide 1.2 MW in thermal energy.

An easier, but less accurate, way is to determine the net heat demand per degree-day. Say a household has a net heat demand of 100 GJ/year and there are 2.800 degree-days in a year, the specific heat demand is 35.7 MJ/degree-day. In the worst-case scenario (30 K temperature difference on a single day equals 30 degree-days), the average required capacity per house is 12 kW[1].

In case of the thermal power for tap water, it is assumed there is a buffer vessel that can hold enough hot tap water to last for a day. The required thermal power is the buffer charge speed times the simultaneous factor for the area. The simultaneous factor is roughly equal to the daily tap water demand, divided by the charge speed, divided by one day. For example, if the charge speed is 2 kW and the daily tap water demand is 12 kWh, then the simultaneous factor is (12 kWh / 2 kW / 24h = 25%. The required thermal capacity per household for tap water is thus 500 W.

[1] 35.7 MJ/degree-day x 30 degree-days/day / 86400 s/day * 1000 kW/MW

3.2 Transport capacity

The thermal power provided by a district heating and cooling network to customers depends on three parameters: the diameter of the pipes, the velocity of the water flowing through the pipes and the temperature difference between the two pipes. In equation form it is written as:

$$P_{th} = \frac{\pi}{4} c_p \rho \Theta_{hc} d_i^2 v \tag{3.1}$$

Where

P_{th}	[W]	The thermal power
Θ_{hc}	[K]	the temperature difference between the two pipes.
d_i	[m]	the inner diameter of the pipe
v	[m/s]	the velocity of the fluid through the pipe
c_p	[J kg^{-1} K^{-1}]	the specific thermal energy of the fluid
ρ	[kg/m^3]	the volumetric density of the fluid

The velocity of the water has impact on the sound, on the wear of the piping and on the risk and impact of pressure waves through the system. The velocity commonly lies between 1 m/s and 3 m/s. In and near houses, the velocity is usually limited to 1 m/s to prevent noise complaints. In some district heat networks, such as in London [7], higher velocities (up to 6 m/s) are used in transmission networks with long straights, although special measures have been taken to prevent damage to piping from pressure waves through the system.

Figure 3.2 shows the relationship between inner pipe diameter, water velocity and the thermal transport capacity for a district heating and cooling network with a temperature difference of 10 °C.

In (traditional) district heating networks, the inner pipe diameter is rarely larger than 1000 millimetre (e.g. Stockholm, Sweden or Flensburg, Denmark). Figure 3.2 shows that the maximum thermal capacity of a heat trajectory (for low temperature heating grids) is thus limited to about 400 MW. If more thermal power is needed, multiple trajectories are required.

Transportation of thermal energy is therefore more limited than other forms of energy transportation such as electricity.

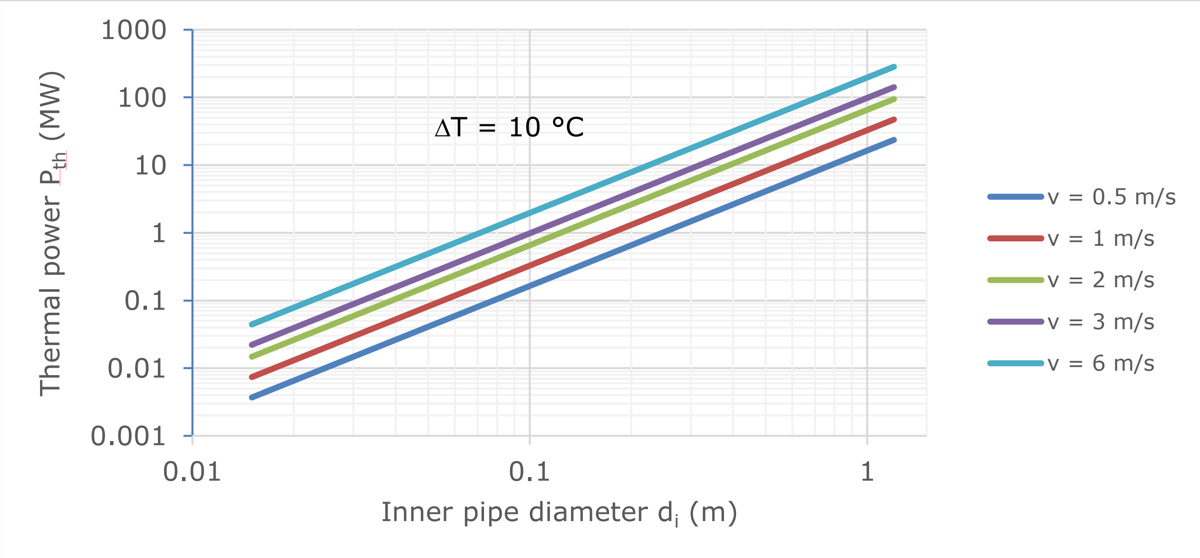


Figure 3.2: Relation between inner pipe diameter and thermal transport capacity for different fluid velocities.

3.3 Pressure drops

Due to pipe friction, the pressure of a flow drops over distant. To keep the system working, this pressure drop is overcome by a pump. The pressure drop in a circular pipe is calculated using the Darcy-Weisbach equation:

$$\Delta p = -\frac{8fL}{d_i^5 \pi^2 \rho}\left(\frac{P_{th}}{c_p \Theta_{hc}}\right)^2 \qquad (3.2)$$

With

Δp	[Pa]	pressure drop in the system
L	[m]	the length of the pipe
F	[-]	the friction factor of the pipe

The friction factor is determined from the Colebrook-White equation. However, this equation requires an iterative solution and is not practical in its use. Therefore, solutions are commonly looked up in a Moody diagram, which is a graphical representation of all solutions from the Colebrook-White equation. An example of a Moody diagram can be found in Figure 3.3. To determine the friction factor in a Moody diagram, the Reynolds number for the fluid and relative roughness of the pipes need to be known. The Reynolds number is calculated following:

$$\mathrm{Re} = \frac{\rho v d_i}{\mu} \tag{3.3}$$

With

Re	[-]	the Reynolds number
μ	[Pa s]	the dynamic viscosity of the fluid.

and the relative roughness following:

$$\mathrm{Roughness} = \frac{\epsilon}{d_i} \tag{3.4}$$

With

ϵ	[m]	the pipe surface roughness

The typical range for the friction factor in district heating and cooling networks is between 0.015 and 0.04.

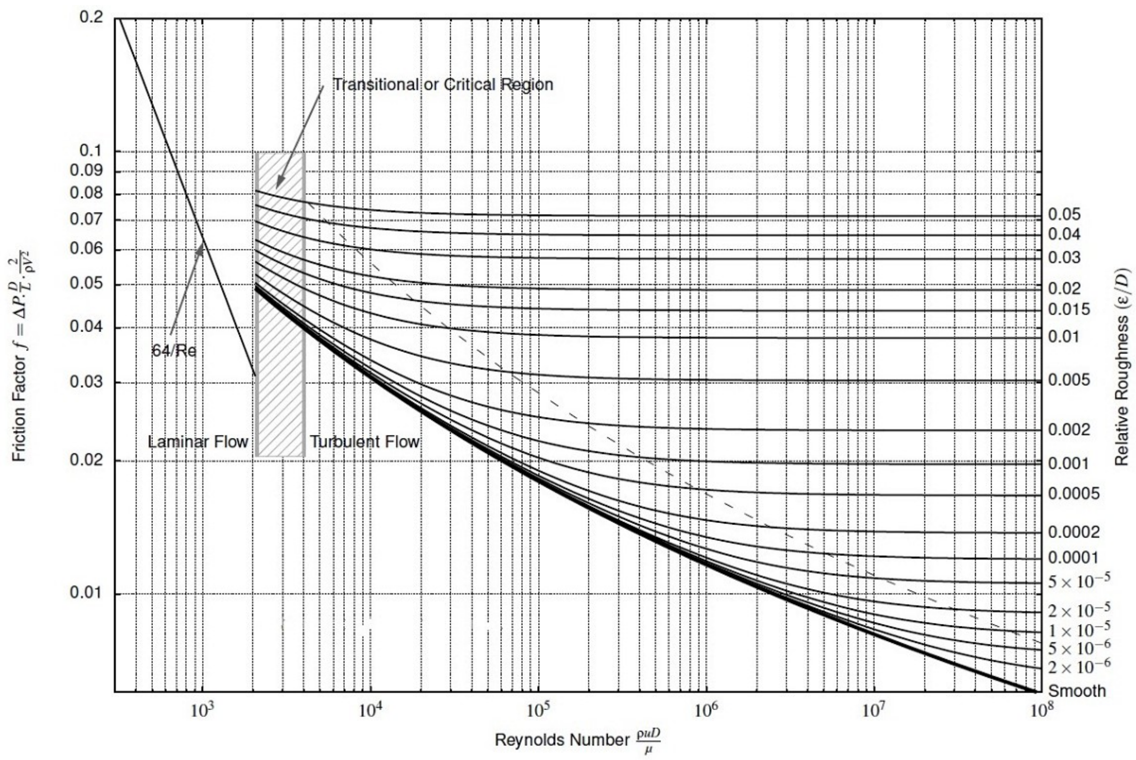


Figure 3.3: A Moody diagram allows one to empirical deduce the friction factor from the Reynolds number and relative roughness

Equation (3.2) shows that the pressure drop is inversely proportional to the inner diameter to the fifth power. This means the pressure drop increases exponentially with smaller pipe diameter. If the diameter is decreased by a factor of 2, the pressure drop increases by a factor of 32. The impact of such exponential behaviour is shown in Figure 3.4, where the pressure drop has been calculated as function of the thermal capacity for different pipe sizes.

The exponential behaviour of pressure drop makes proper pipe sizing important. Although the pressure drop increase of a slightly smaller pipe can be overcome by a larger pump, there are clear limits. At a certain point, the pressure drop is simply too large to be compensated by larger pumps.

An economic trade off must be made. Smaller pipe sizes are cheaper. The pipes itself are cheaper, but there is also less amount of excavating work to be done. Smaller pipe sizes result in higher costs for a more powerful pump. Larger pipe sizes are more expenses but result in lower pump investments. An optimal trade-off between pipe size and pump capacity may be found. However, Future expansions of the network must be considered.

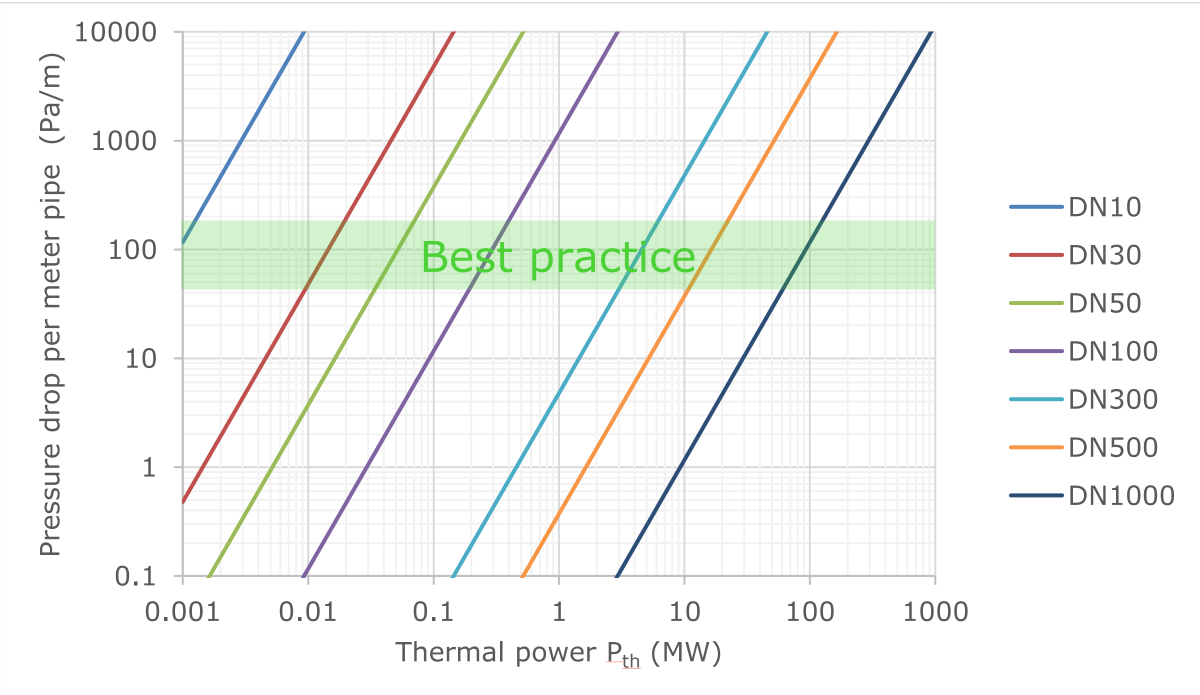


Figure 3.4: Pressure drop per meter of pipe versus the thermal power transport capacity for different pipe sizes.

Current design practice revolves around pressure drops in pipes between 50 and 200 Pa/m, which is marked as the green area in Figure 3.4.

Other components, such as bends, T-sections, valves and heat exchangers, also contribute to pressure drops and should be included when calculating the total pressure drop in the system.

For practical purposes, equation (3.2) can be further simplified. If the following values are substituted:

- Recommended pressure drop is 50-200 Pa/m.
- Friction factor lies between 0.015-0.04.
- The density of water equals 998.19 kg/m^3 (at 20 °C, atmospheric pressure).
- The specific thermal capacity of water is 4,180.44 J kg K^{-1}.
- The temperature difference between the two pipes is 10 °C.

It is deduced that the optimal inner pipe diameter lies in the interval:

$$5.1102\ \times 10^{-4}\ (P_{th})^{0.4}\ \leq d_i \leq 8.205\ \times 10^{-4}\ (P_{th})^{0.4} \qquad (3.5)$$

The interval relationship in equation (3.5) has been visualised in Figure 3.5.

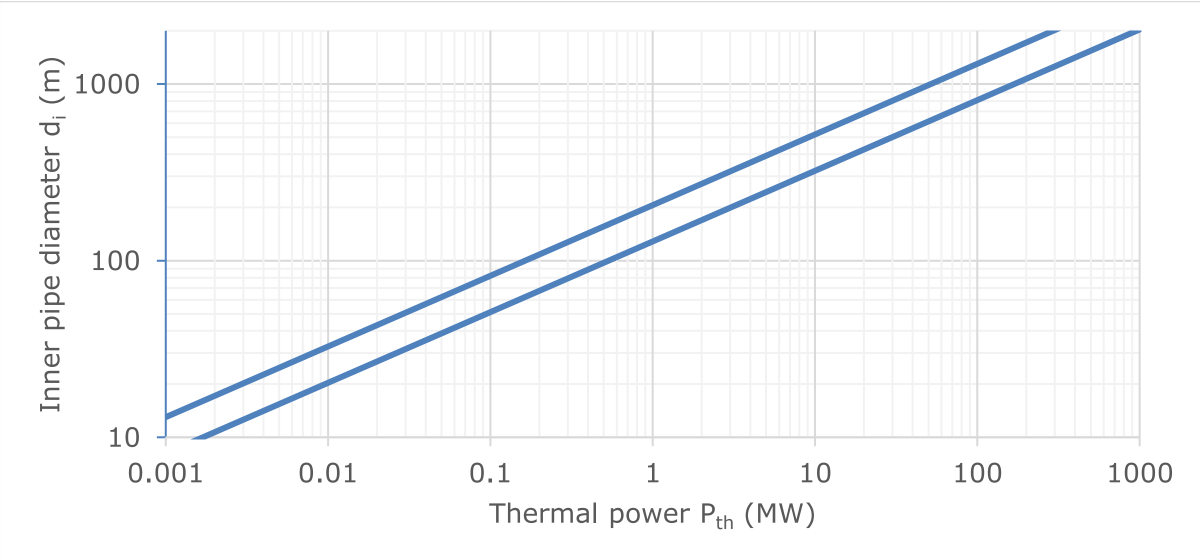


Figure 3.5: Simplified relationship between thermal capacity and inner pipe diameter for a low temperature district heating and cooling network

3.4 Distribution losses

High temperature district heat networks have significant thermal losses, which can be up to 25% of the total heat demand of the network. More fuel needs to be burned to compensate these losses, impacting the environment and operational costs.

Figure 3.6 shows a generic pipe used for district heating and cooling networks. The inside is made from a metal cylinder, which is covered by an insulating layer of material with a low thermal conductivity. The insulation is covered by a jacket consisting of a thin layer of waterproof material (not shown in the figure) to prevent the insulation material become wet and lose its effectiveness.

Piping that are buried underground, gain additional insulation through the ground itself.

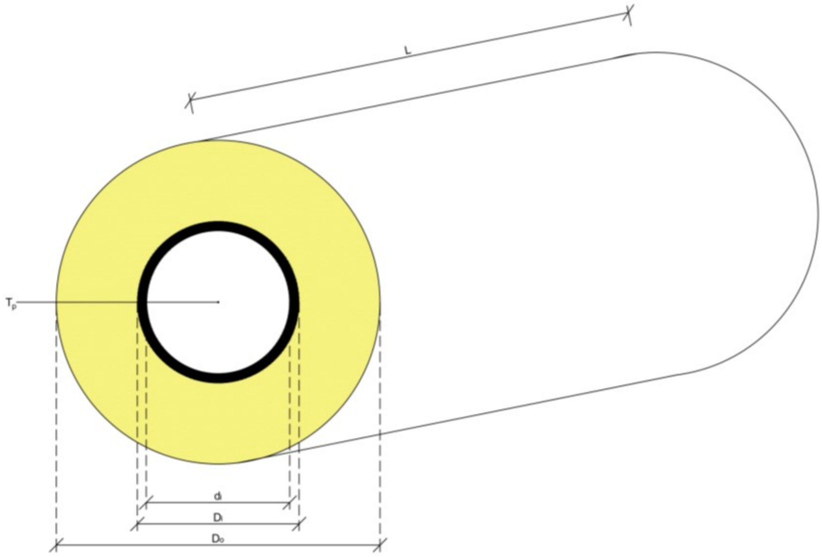


Figure 3.6: 3D schematic of an insulated pipe used for district heating or cooling.

3.4.1 Thermal resistance

Thermal resistance is a measure of how difficult it is for heat to flow from a body with a higher temperature to a body with a lower temperature. The higher the resistance, the less heat flow between the two bodies. In a general equation form this can be written as:

$$P = \frac{\Theta}{R} \tag{3.6}$$

with

P	[W]	the heat flow from the body with the higher temperature to the body with the lower temperature.
θ	[K]	the temperature difference between the two bodies.
R	[K/W]	the total thermal resistance between the two bodies.

For an insulated pipe, the thermal resistance of the insulation can be calculated by solving Fourier's equation in cylindrical coordinates, of which the derivation can be found in chapter 9. The thermal resistance R_i of the insulation of the pipe is calculated by:

$$R_i = \frac{1}{2\pi\lambda_i L} \ln\left(\frac{D_o}{D_i}\right) \tag{3.7}$$

Similarly, for a buried pipe, the thermal resistance R_g of the soil can be approximated by:

$$R_g = \frac{1}{2\pi\lambda_i L} \ln\left(\frac{4h}{D_o}\right) \tag{3.8}$$

with

D_i	[m]	the inner diameter of the insulation layer.
D_o	[m]	the outer diameter of the insulation layer.
L	[m]	the length of the pipe
h	[m]	the underground depth measured from surface edge to the centre of the pipe
λ_i	[$Wm^{-1}K^{-1}$]	the specific thermal conductivity of the insulation layer.
λ_g	[$Wm^{-1}K^{-1}$]	the specific thermal conductivity of the ground

The approximation in equation (3.8) is valid when $h \geq 2D_o$ [4][5].

To get a feeling what realistic numbers are for thermal resistance of district heating piping, specifications from the manufacturer Weijers-Waalwijk for the Prinspipe series have been used. Prinspipe are classic steel pipes with a PUR insulation, available in many diameters. The specifications for the Prinspipe series are listed in chapter 10.

Figure 3.7 shows the thermal resistance of the pipe insulation for different pipe sizes and for three different product lines of Prinspipe. Each line has a different thickness of insulation. The thermal resistance for larger pipe sizes is lower, as the insulation thickness / inner pipe diameter ratio lowers. For example, the insulation thickness for the DN25 and DN500 (type1) pipe are respectively 56mm and 200mm. Although the inner diameter of the DN500 pipe is 17 times greater than of the DN25 pipe, the insulation thickness is only 4 times greater.

Although larger pipes carry relatively less insulation, they do transport more water volume. As the volume increases with the square of the diameter and the heat loss surface linear, the volume / heat loss surface is greater for larger pipe sizes. This means that although the heat losses are higher, the corresponding temperature drop is lower.

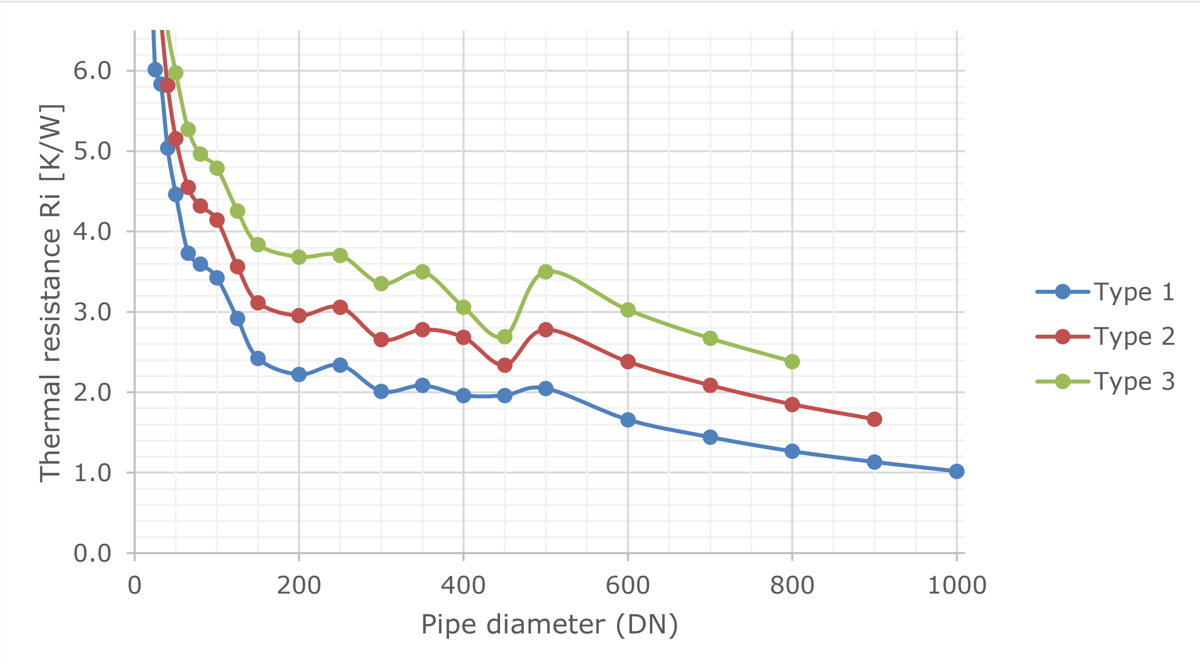


Figure 3.7: Thermal resistance of the insulation for three different series of pipes from the Prinspipe range with a pipe length of 1 meter.

For the thermal resistance of the ground, the depth plays an important role. The specific thermal conductivity of soil is typically 1.5 $Wm^{-1}K^{-1}$ [3], but greatly depends on the type of soil and its water content. Values can range between 0.25 $Wm^{-1}K^{-1}$ and 5.0 $Wm^{-1}K^{-1}$. The specific thermal conductivity for different types of soil are listed in section 10.1. In Figure 3.8, the thermal resistance is plotted for numerous pipe sizes. Distinction is being made by the depth the pipes are buried underground. The pipe data for the Prinspipe type 1 pipe series has been used.

The thermal resistance of the soil is lower for larger pipe sizes, as larger diameter pipes expose more surface area to the surface of the ground. The thermal resistance of the soil increases with increasing depth as the soil acts as a thicker insulation layer.

Comparing Figure 3.7 with Figure 3.8, it is clear that the soil provides an additional 7% to 9% of extra insulation for small pipe sizes and up to 30% for large pipe sizes.

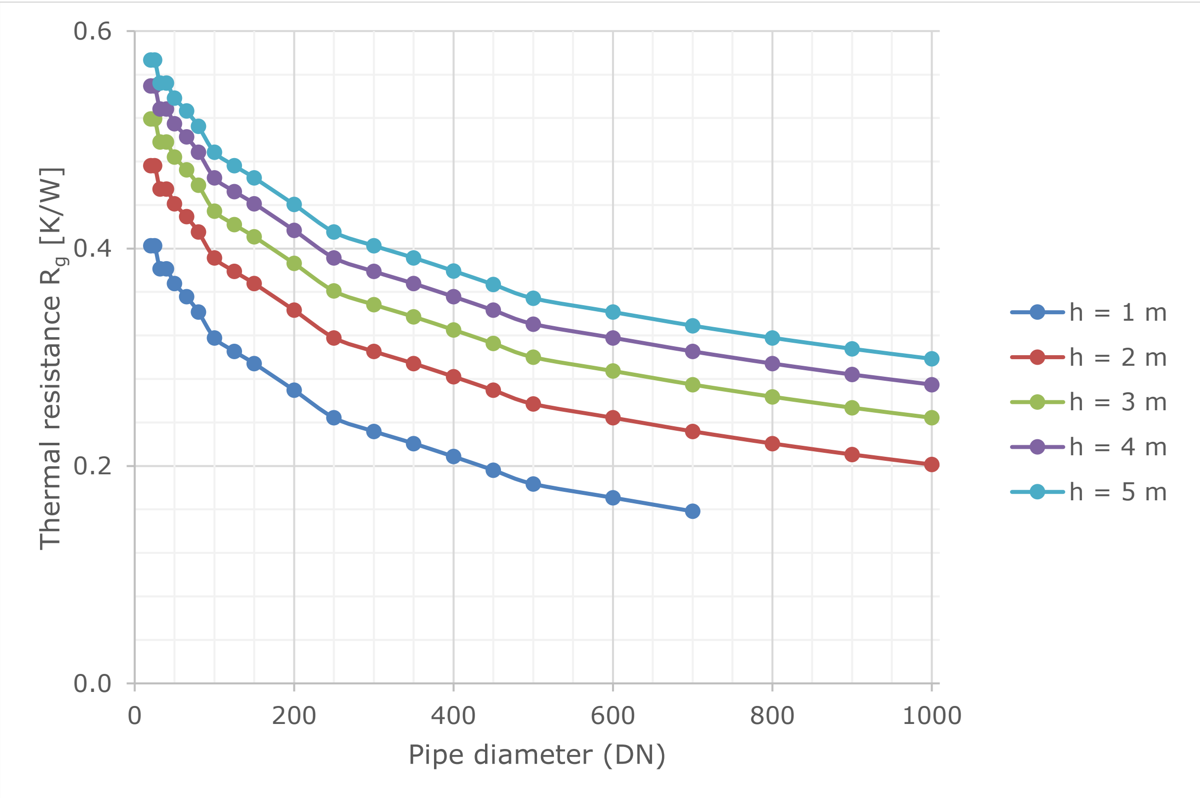


Figure 3.8: Thermal resistance of the soil for one meter long pipes with different diameters and at different depths for a single Prinspipe type.

Friction (pressure loss) adds heat to the system. In large pipes, the friction may be more than the heat loss.

3.4.2 Single insulated pipe

The heat loss of a single insulated and buried pipe is derived from equation (3.5) and expressed as:

$$P_{loss} = \frac{\Theta_{pa}}{R_i + R_g} \tag{3.9}$$

with

P_{loss}	[W]	the thermal loss
Θ_{pa}	[K]	the temperature difference between pipe and ambient air.

Thermal resistance of the interface layers (e.g. water and steel pipe, and air and soil), as well as thermal resistance of the steel pipe and waterproof pipe jacket have been ignored but could contribute to another 0.1 K/W in thermal resistance.

For a type 1 Prinspipe pipe buried 2 meters underground, the thermal losses per unit length are shown in Figure 3.9 for a variety of temperature differences between pipe temperature and ambient temperatures. Note that nor the absolute temperature of the water in the pipe, nor the ambient temperature are relevant. It is the difference between the two that is important.

If the ambient temperature is higher than the pipe temperature, then $\Theta_{pa} < 0$ and P_{loss} will become negative. In such a case, there is no heat loss, but a heat gain. Due to the low temperatures used in the heat-pipe, this situation could occur during the summer season.

Thermal losses for the cold-pipe are calculated the exact same way. However, heat losses are beneficial for the cold-pipe as it cools the pipe of, while a heat gain requires additional cooling to keep the cold-pipe at its maximum temperature.

The transfer of thermal energy from heat-pipe to a cold-pipe that are buried underground next to each other, is discussed in section 3.4.4.

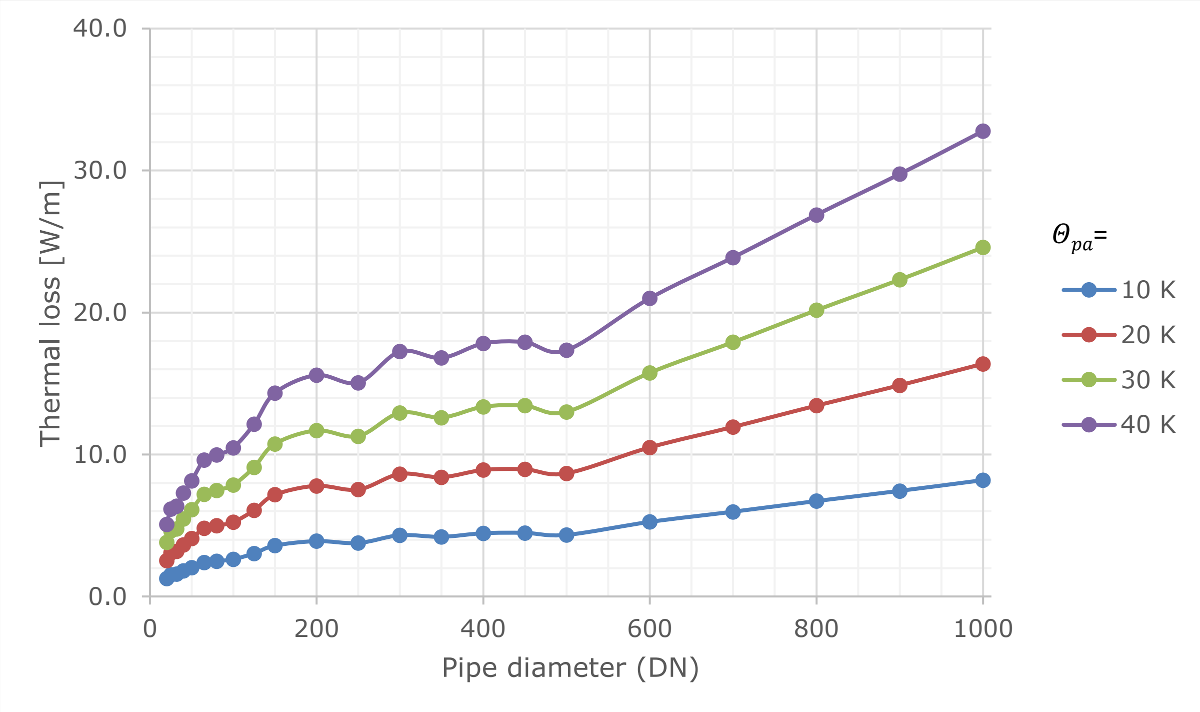


Figure 3.9: Thermal losses per meter for a pipe 2 meter underground for various pipe sizes and varies differences between the pipe and ambient temperature.

The temperature change as result of the thermal losses in a heat-pipe and cold-pipe are estimated following:

$$\Delta T = \frac{-4}{\pi v d_i^2 \rho c_p} \frac{\Theta_{pa}}{R_i + R_g} \tag{3.10}$$

Where the estimation is valid if $\Delta T \ll \Theta_{pa}$.

If $\Delta T < 0$, the temperature of the pipe is decreasing. Say that the temperature change is calculated at $\Delta T = -0.1\ \mathrm{K}$. The temperature drop is then 0.1 K, meaning that a pipe that has e.g. a temperature of 25 °C at the beginning, has a temperature of 24.9 °C at the end. Keep in mind that the length of the pipe is already included in the thermal resistance R_i and R_g.

For the pipes in Figure 3.9, the temperature drop per unit of length has been calculated, assuming a flow velocity of 1 m/s. Higher flows proportionally reduces the temperature drop. The results are shown in Figure 3.10.

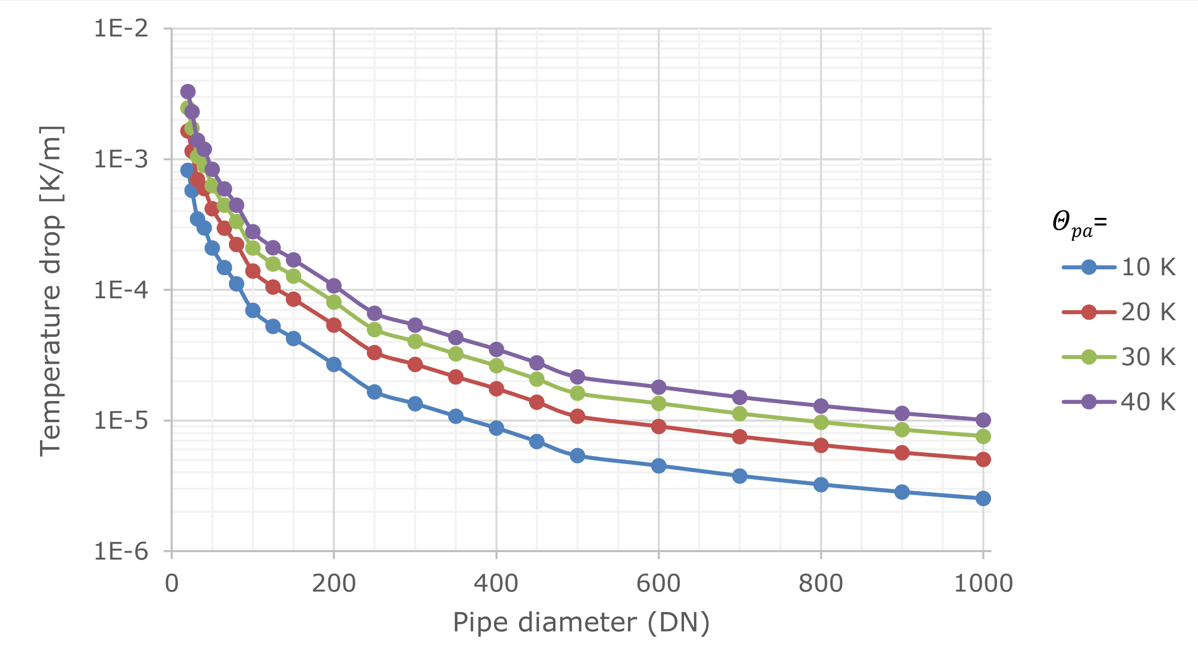


Figure 3.10: Temperature drop per meter for a pipe buried 2 meter underground with a flow velocity of 1 m/s for various pipe sizes and varies differences between the pipe and ambient temperature.

Important to realise is that traditional district heat networks have high losses, because of the large temperature difference between the pipe and ambient air, which can be up to 130 K on cold days.

Figure 3.10 clearly shows that insulated pipes in low temperature district heating and cooling networks are not significantly impacted by the temperature drop. For example, a DN50 heat-pipe of 1 km long and a temperature of 25 °C has a temperature drop of almost 1 K if the ambient temperature is -5 °C. Another example, a DN600 heat-pipe of 10 km long has a temperature drop of 0.02 K when the ambient temperature is -15 °C.

It is important to realise that a 1 km long DN50 is unlikely, as its thermal capacity will be extremely limited due to the pressure losses, as discussed in section 3.3.

Overall, one can conclude that proper dimensioned insulated piping for low temperature district heating and cooling networks with some water flow, have acceptable temperature changes caused by thermal losses and thermal gains.

3.4.3 Single non-insulated pipe

Because low temperature district heat networks have relative low temperatures, thermal losses are also significantly lower than for traditional high temperature networks. As such, the question arises whether pipe insulation is needed.

Abandoning pipe insulation has a few advantages. First, the production costs of the pipes are lower as less material is used and less production steps are required to build the pipe. Second, the trench to be dug can be narrower and less deep, as the outer pipe diameter is smaller and as such, reducing costs. Finally, creating a pipe joint is cheaper as the insulation layer over the joint is no longer needed, again reducing costs.

There are also several disadvantages. First, the temperature losses are higher. This means more thermal energy is required to keep the network operational, but also increases the risk of significant temperature deviations as a result of the temperature drop. This could lead to insufficient thermal capacity at the customers connection point. Secondly, leak detection for steel pipes is commonly present in the insulation layer (where the steel pipe itself serves as grounding). The lack of an insulation layer requires a different detection system.

The temperature drops for the pipes in Figure 3.10 have been recalculated, where the insulation layer has been removed. The thermal resistance for the insulation layer has been set to zero and the outer diameter of the pipe equals the outer diameter of the steel cylinder. The pipes are still buried 2 meters underground. The results are shown in Figure 3.11.

Temperature drops are about an order of magnitude higher for pipes without insulation compared to pipes with insulation. This may pose a problem for smaller sized piping. However, for larger sized piping, the temperature drop is still small enough that it would be feasible option.

Its feasibility is not only cost driven, but also flow driven. The above calculations have been performed with a flow of 1 m/s. But what if the flow is (almost) non-existent?

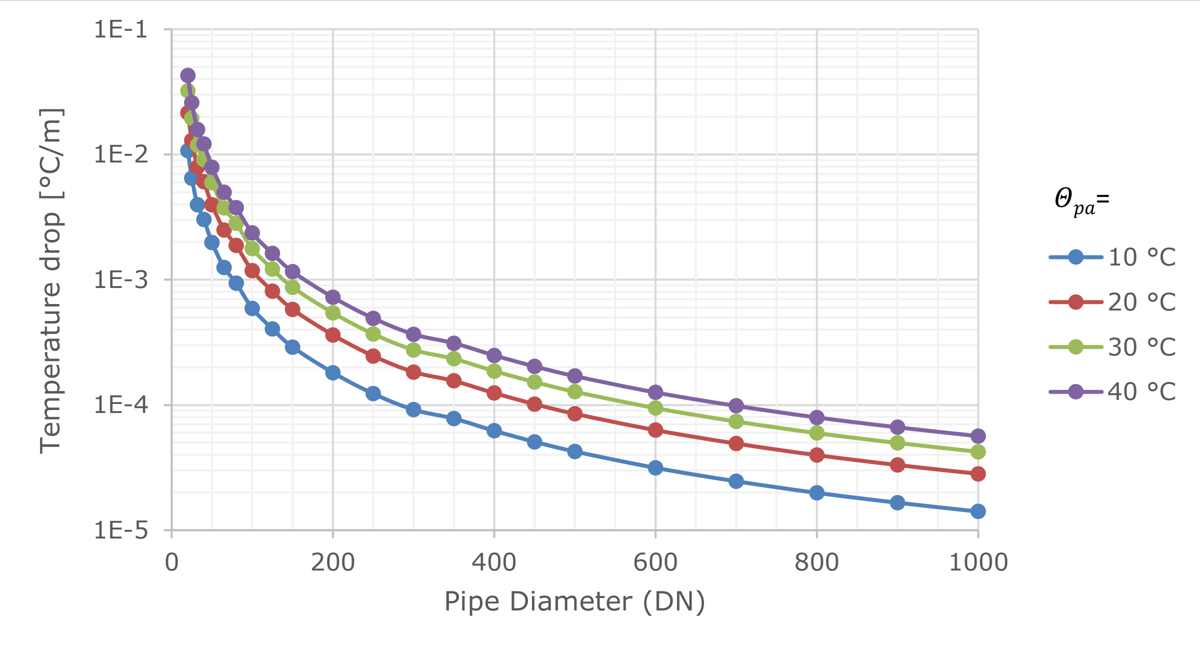


Figure 3.11: Temperature drop per meter for a pipe 2 meter underground with a flow velocity of 1 m/s for various non-insulated pipe sizes and varies differences between the pipe and ambient temperature.

3.4.3.1 No flow

When there is very little flow, the estimation required $\Delta T \ll \Theta_{pa}$ will not hold. As such, the temperature drop can no longer be determined by a steady-state approach but requires a time component.

Such situations may occur during night-time, when the demand for thermal energy is low. The lower the demand, the smaller the flow will be.

The temperature curve in time for a non-insulated pipe with no flow can be calculated following:

$$T(t) = \Theta_{pa} \mathrm{e}^{-\frac{4t}{\pi d_i^2 \rho L c_p R_g}} + \mathrm{T_a} \qquad (3.11)$$

With t the time in seconds. Consider an uninsulated heat-pipe of 1 meter in length, with a water temperature of 25 °C, buried 2 meters underground and an ambient temperature of -5 °C. For four different pipe sizes, the temperature curve over a period of 10 hours (600 minutes) has been calculated. The results are shown in Figure 3.12.

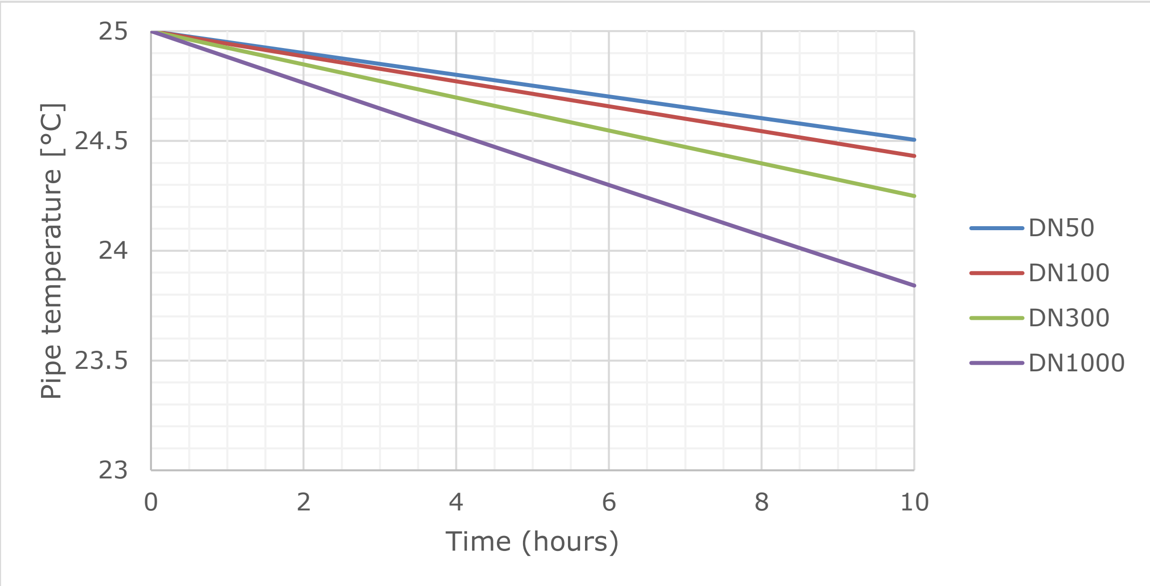


Figure 3.12: Temperature in a pipe with no flow for 10 hours.

Using this method, it can be determined whether the temperature drop is acceptable.

Imagine that there is no flow in a district during the night. The temperature of the heat-pipe drops from 25 °C to 23 °C and the temperature in the cold-pipe remains approximately the same. When customers start to demand heating, the effective thermal power is initially reduced by 20%, as the temperature difference between the two pipes is no longer 10 °C, but 8 °C. After a while, the heat-pipe will be "flushed" and its temperature will be back to its nominal value.

In reality, the reduced thermal power is dampened as the cold-pipe also drop in temperature. When the heat-pipe drops from 25 °C to 23 °C, the cold-pipe may drop from 15 °C to 14 °C. The effective thermal power is then only 10% lower from the nominal thermal power.

The reduction of thermal power can be further reduced, by “flushing” the pipes intermittently, for example by having one storage unit demanding heating and another storage unit demanding an equal amount of cooling.

3.4.3.2 Above ground

When non-insulated pipes are above ground, the thermal resistance of the ground is non-existing too. The thermal resistance is then only determined by the components that have been ignored earlier on: water-pipe interface, air-pipe interface, (steel or plastic) pipe.

As the combined thermal resistance is very low, it is estimated to be about 0.1 K/W, thermal losses will be high. However, in certain situations, these uninsulated, non-buried pipes may be preferred.

An example is the piping in an apartment building. As the temperature in the non-heated areas are likely to be less extreme than the ambient temperature and the length of the piping is relatively short, the cost-benefit may preference over the increased thermal losses.

These pipes may be made of a plastic instead of steel, making them cheaper, but also increasing its thermal resistance.

3.4.4 Two-pipe system

A district heating and cooling system consists of two pipes, a heat-pipe and a cold-pipe. It is very likely that these two pipes are buried underground next to each other. As they have different temperature levels, there is a heat transfer from the heat-pipe to the cold-pipe. This heat transfer always occurs, regardless of the ambient temperature.

In traditional heat networks, the supply and return pipes benefit from each other’s temperature fields, reducing losses. For a district heating and cooling network, the heat flow and has a negative impact on both pipes: the heat-pipe cools off and the cold-pipe warms up.

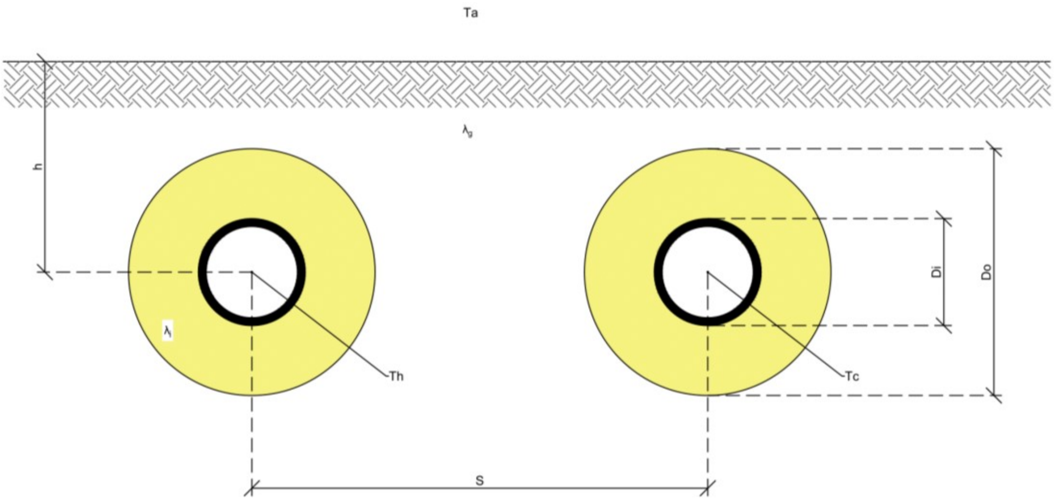


Figure 3.13: Schematic of two thermal pipes underground.

The heat flow from heat-pipe to cold-pipe is estimated following [1][4]:

$$P_{hc} = \frac{R_c}{\left(R_g + R_i\right)^2 - R_c^2} \Theta_{hc} \tag{3.12}$$

with

$$R_c = \frac{1}{2\pi\lambda_g L} \ln\left[\sqrt{\frac{4h^2}{s^2} + 1}\right] \tag{3.13}$$

and

P_{hc}	[W]	the thermal power exchange between heat-pipe and cold-pipe.
Θ_{hc}	[K]	the temperature difference between heat-pipe and cold-pipe.
s	[m]	the horizontal distance between the two pipes, measured from the centre of the pipes.

Because of the equal-sized pipes, the heat resistance between the two pipes only depends on the ration between depth (h) and distance (s).

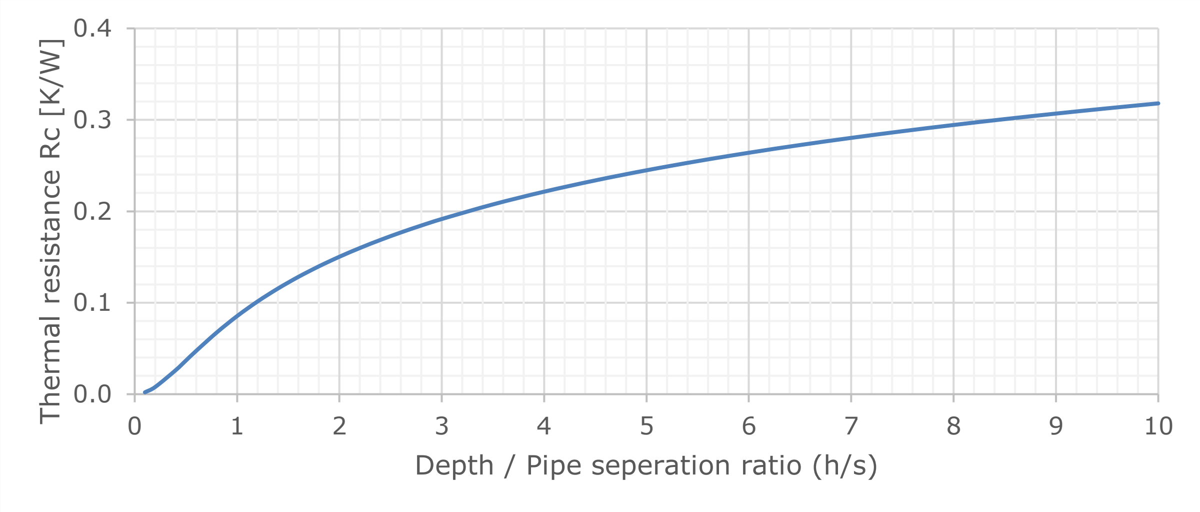


Figure 3.14: Thermal resistance between two underground pipes for different ratios of the underground depth (h) and distance between pipes (s).

In Figure 3.14, the heat resistance R_c is plotted against the ratio h/s for the type 1 Prinspipe. Note that the larger the ratio of h/s is, the closer the pipes are to each other with respect to the depth. A ratio of 1.0 means that the pipes buried as deep as they are separated horizontally.

The heat flow between two pipes are plotted in Figure 3.15. Compared to Figure 3.9 the values are relatively small but gain significance for larger sized pipes that are close to each other.

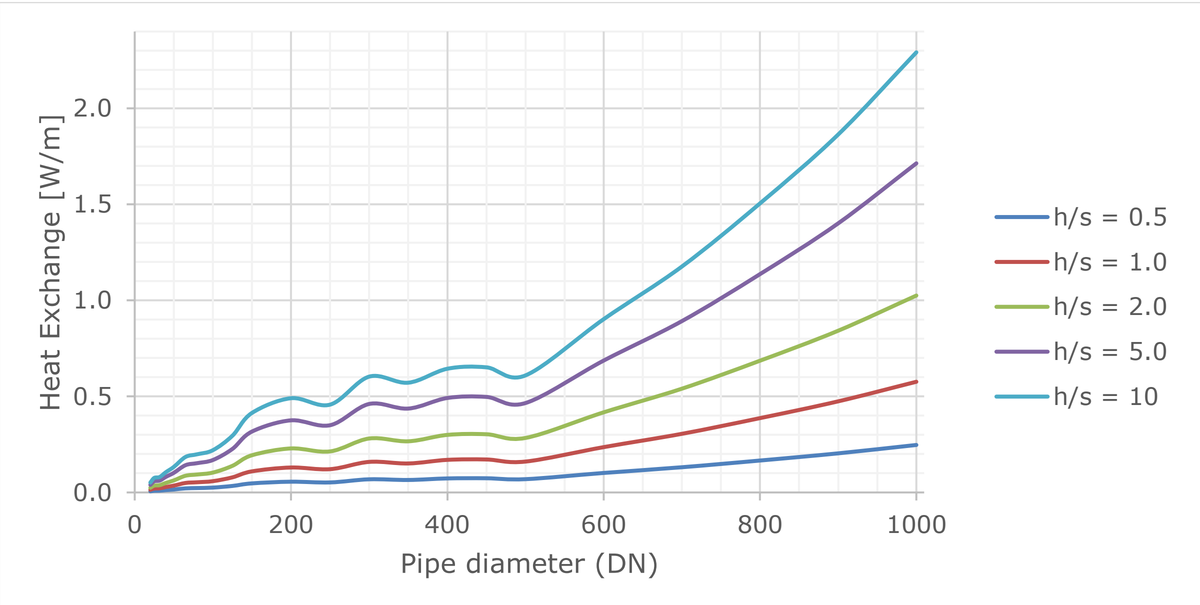


Figure 3.15: Heat exchange between heat and cold pipe for two underground pipes for different ratios of the underground depth (h) and distance between pipes (s).

In wintertime, when the ambient temperature is lower than the temperature of the cold pipe, the cold pipe gets cooled down by the thermal losses between pipe and ambient. At the same time, it gets warmed up by the thermal losses from the hot pipe. These two counteract each other. On the other hand, the heat-pipe loses its heat to both the ambient and the cold-pipe. In summertime, this situation is reversed, where the cold-pipe loses cold to both ambient and the heat-pipe, while the heat-pipe loses heat to the cold-pipe and gains heat from the ambient.

3.4.5 Twin pipe system

In a twin pipe system, two pipes are included in the same circular insulation. Therefore, only one pipe (with the two smaller pipes inside) is put underground. The main advantage in traditional heat networks is that the coinciding temperature fields of the supply and return line, up to 40% for smaller distribution pipe sizes.

The main purpose of a twin pipe system, creating coinciding temperature fields, does not fit into the concept of a two-pipe heating and cooling network as there is no return. On the contrary, the heat-pipe and cold-pipe should be separated as much as possible.

3.5 Pump sizing

In a hydraulic system, a pump provides the work to provide a flow by overcoming the pressure difference. The work done by a pump is calculated by:

$$W_{pump} = \Delta p \, \Phi_{\mathrm{v}} \tag{3.14}$$

with

W_{pump}	[W]	the work done by the pump per unit of time.
Φ_{v}	[m^3/s]	the volumetric flow through the pipe.

The electric power absorbed by the pump unit to provide this work is given by:

$$P_{pump} = \eta_{\mathrm{el}} \, \eta_{pump} \, W_{pump} \tag{3.15}$$

with

η_{el}	[-]	the efficiency of the electric motor.
η_{pump}	[-]	the pumping efficiency of the pump.

The characteristics of a pump are described in a pump curve (or head-flow curve). This curve gives a relationship between pressure difference and flow. A fixed speed pump (a pump that is either on or off) can only operate on the curve. The pressure loss in the system is known as the system curve and increases quadratically with the flow. The intersection of the pump and system curve is the operating point of the pump. This is visually shown in Figure 3.16

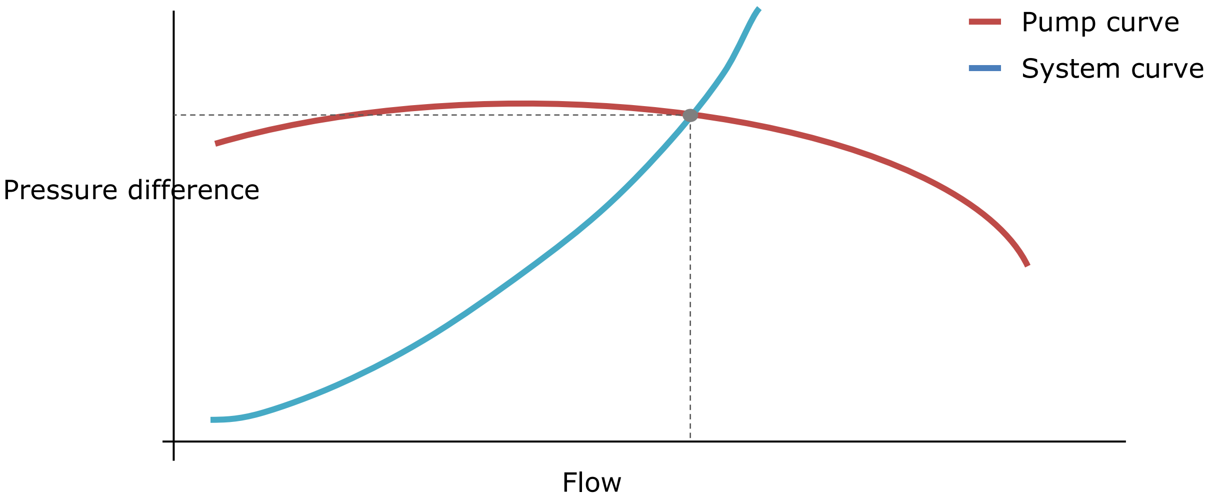


Figure 3.16: The intersection of the system and pump curve determines the flow.

The efficiency of the pump depends on where it operates on the pump curve. The maximum efficiency for small centrifugal pumps is about 50-70% and for large pumps up to 90%. The efficiency of the electric motor powering the pump typically lies between 90-97%.

Two pumps in parallel add the individual pump curves along the flow-axis (for the same pressure difference, it gains twice the flow). Two pumps in series add the individual pump curves along the pressure difference-axis (for the same flow, it can overcome twice the pressure). The operating point of the pumps moves along the system curve. Two pumps in series or parallel moves the operating point along the system curve as shown in Figure 3.17 and Figure 3.18

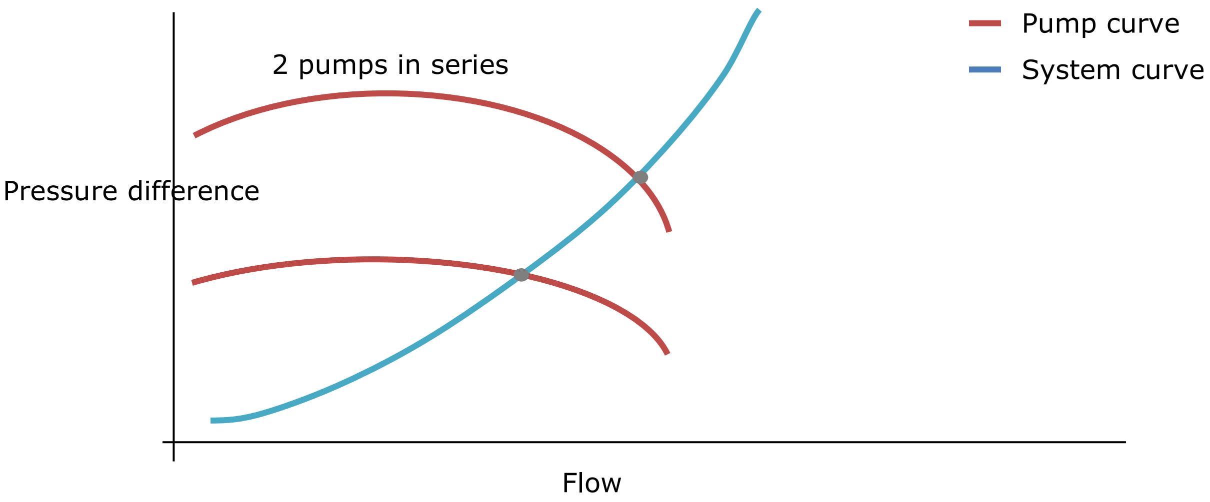


Figure 3.17: Pump and system curves of two pumps in series. The pumps combined can overcome a higher pressure difference.

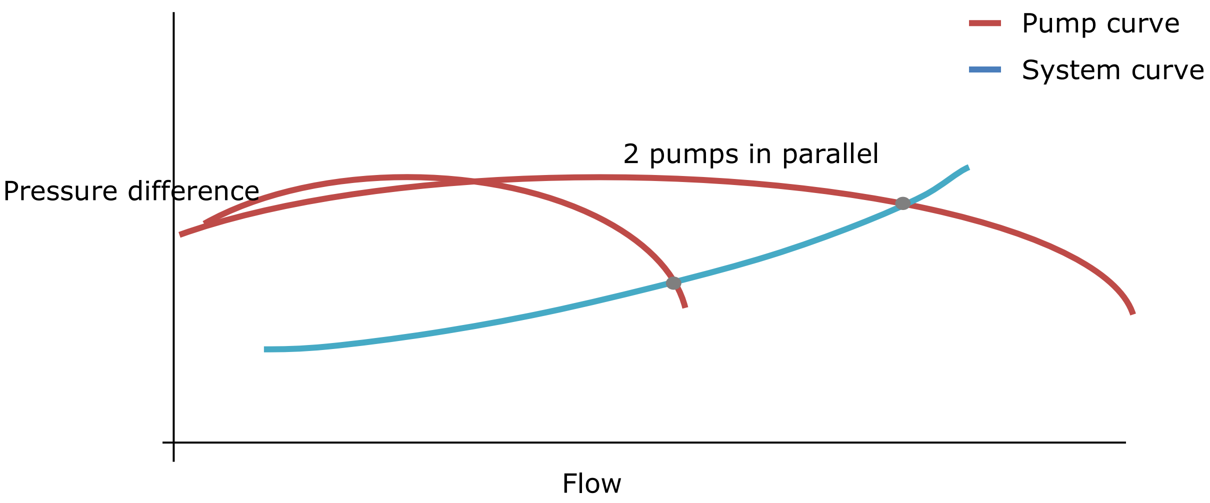


Figure 3.18: Pump and system curves of two pumps in parallel. The pumps combined provide more flow.

Many pumps are not fixed speed but variable speed. They can control the throttle. Each speed has its own pump curve, providing more flexibility and more control of the flow.

3.5.1 Pump configuration

Every customer in a network has a bi-directional pump, which controls the amount of thermal energy exchanged between customer and network, by adjusting the flow. The pump must therefore be variable speed.

The pumps are parallel connected in the network. Say one customer has its pump turned on at 50% speed to get a flow of 2 l/s from the heat-pipe to the cold-pipe. A second customer also decides to turn their (identical) pump on at 50% speed. Because of the quadratic pressure drop, each customer now has a flow that is less than 2 l/s. Both pumps must increase the throttle to reach the desired 2 l/s.

Now say a third, customer does the same. All three pumps now must run at 100% to obtain the desired flow per customer. If a fourth customer wants to join in, the desired flow of 2 l/s is no longer achievable.

In other words, the more customers are pumping, the higher the speed of the pumps to achieve the desired flow. Pumps could reach their limit if there is high demand for thermal energy. Whether this may become an issue, depends on several factors.

The first factor is the simultaneousness operation of the pumps. The higher the percentage of pumps operating at the same time, the higher the risk the limits are reached. Distributing pump operations to reduce the simultaneousness factor can be achieved by using thermal buffers or intelligent control

Another factor is the maximum pump capacity of the pumps. The higher the maximum pump capacity in relation to the desired pump capacity, the more the pump can adjust to increasing system pressure difference. A significant downside of a higher pump capacity is the increased investment costs in pumps.

Thirdly, the amount of control over the temperature difference affects the risk. If there is a high demand for thermal energy (i.e. many pumps are operating), the total flow (and pressure drop) can be reduced by increasing the temperature difference between the two pipes.

And finally, the differentiation of customers has a major impact. In a relative uniform network (where all the customers are similar, e.g. houses with heat demand), pumps operate in the same direction. In highly differentiated networks (the network has a wide range of customers locally demanding cold and heat), up to half the pumps are pumping the other way. i.e. 50% of the operating pumps are pumping flow from the heat-pipe to the cold pipe and the other half are pumping from the cold-pipe to the heat-pipe. This has two effects:

- The two groups of pumps are connected in series, providing a boost to overcome the pressure difference.
- The length through which the water flows is significantly shorter, reducing the pressure drop (which is a function of pipe length).

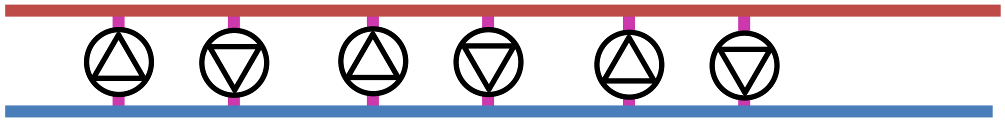


Figure 3.19: A diverse network has pumps supporting each other and shorter distances through which thermal energy is exchanged.

In case the network is relatively uniform, and the risk of reaching the decentralized pump limits cannot be mitigated, a centralized pump at e.g. substation can be placed. As this pump is placed in series with all the other pumps, it helps overcoming the pressure drop in the system.

Figure 3.20: An extra pump helping to overcome the pressure difference in a uniform network.

3.5.2 Conclusion

- Every customer in the network has a bi directional pump.
- The size of the pumps is determined by the normal operation conditions, e.g. 100 pumps of which 30% running at 50% capacity, each providing a flow of 0.3 l/min.
    - The total flow in the network is known (100 * 30% * 0.3 l/min).
    - The pressure drop is calculated
    - A variable speed pump has a pump curve that would match the system curve to obtain the required operating point.
- The risk of reaching the limits of the individual pumps in a worst-case scenario is determined. Additional measures are taken which can include
    - Thermal buffers to reduce the simultaneousness of the pumps.
    - Intelligent control to reduce the simultaneousness of the pumps.
    - Management of the temperature difference to reduce the flow speed.
    - More local diversity in heat and cold demand.
    - A centralized pump to provide a pressure boost.

4. Topology

4.1 Single network topology

Traditional heating (and cooling) networks have tree-like topologies. Commonly, a transmission (or primary) network is used to transport the heat from a central heat source to numerous districts. Distribution stations in the district take the heat from the transmission network and feed it into a distribution network. The distribution network is characterised by many branched pipes to get the heat to all customers. Sometimes a (partially) meshed structure is used to improve the flow and capacity in the network.

They key actors in a low temperature district heating and cooling network are the customer connections. After all, each customer is both a consumer and producer. If a customer requires heating from the network, it provides cooling to the network and vice versa. If the demand for heating and cooling among customers is balanced out, a central source of heating and/or cooling may not be present in the network at all. Therefore, a different topology than the classic tree-structure is needed.

A topology of ring networks provides the solution. Ring networks have the characteristic that they have no beginning and no end. In a ring topology each customer connection has two neighbouring customer connections, assuming a minimum of three customers are present in the network. For a district heating and cooling network, the heat-pipe and cold-pipe each are a closed loop, as sketched in Figure 4.1.

With the lack of a return pipe, there is no predefined flow direction. The direction is determined by the demand for cold and heat. This means that in some part of the networks the flow can be clockwise, and in other parts counter clockwise.

In practice, the network topology is not a perfect circle, but more likely to be polygon shaped (following the streets in a district) and has multiple smaller loops. An example is shown in Figure 4.2.

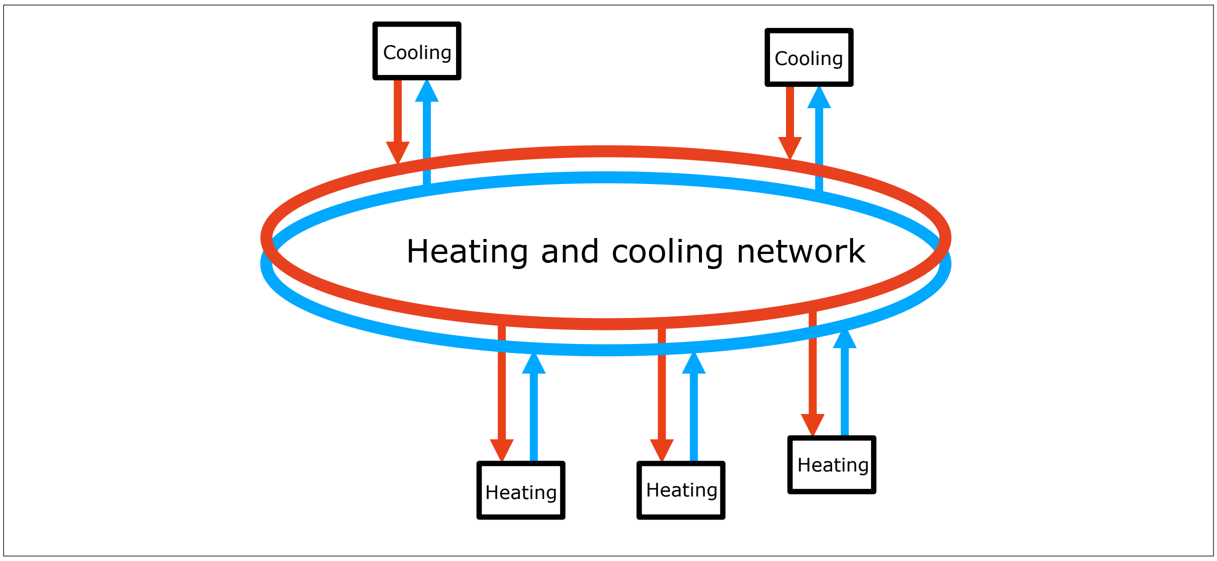


Figure 4.1: A ring topology for a low temperature district heating and cooling network.

To transfer thermal energy to and from a customer, there is a mass flow between the heat-pipe and cold-pipe in either direction. Customers that require heating, take water from the heat-ring and return it in the cold-ring, while customers requiring cooling do the reverse. When there is a mismatch between the demand for heating and the demand for cooling, there is a mismatch in thermal energy transfer, but also a mismatch in mass flow.

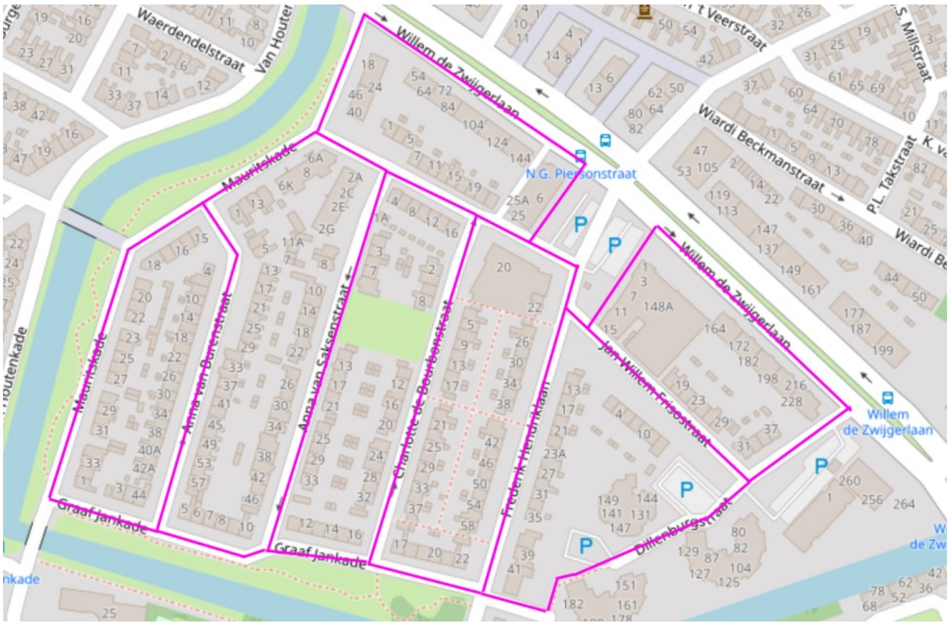


Figure 4.2: Example of a closed loop network topology applied on an actual district.

Thermal balance must be achieved for the network to maintain temperature levels in the heat-pipe and cold-pipe. Mass balance must be achieved for the network to be able to operate at all. To ensure mass and thermal balance, a balancing station is introduced. The balancing station is further discussed in 5.1.

4.2 Multi-network topology

The design process of low temperature district heating and cooling is bottom up. It is likely that multiple district heating and cooling networks appear in a city. If one of these networks has a shortage in cooling and another network has a shortage in heating, it could be economically desirable to have thermal energy exchanged between the two networks by connecting them.

Connection two or more single networks to exchange thermal energy can be interesting for:

- **Peak shaving**: The networks have distinctively different load profiles, in particular when the peak loads in each of the network occur at different moments in time. Connection the networks allows one network to cover the peak demand in the other network.
- **Balancing**: The networks have an opposite structural mismatch in thermal energy. When one network has a deficit in heating and the other network has a deficit in cooling, the connection exchanges these deficits.
- **Scaling and sizing**: Two or more networks have a different size of thermal capacity. For example, several housing districts have a net heating demand of 10 MW each. Nearby is an industrial area that has a net heating supply of 50 MW. A single network in which all three housing districts and industrial area are represented, results in oversizing the housing districts. By creating three smaller networks in the housing districts and one larger network in the industrial area, all networks are sized appropriately (and cost-effective). The smaller networks are then connected to the industrial network.

4.2.1 Two networks

Two networks are connected by a network exchange station (NES) as shown in Figure 4.3. The NES ensures hydraulic separation: water from one network cannot enter the other network. A heat exchanger is required to transfer thermal energy between the networks. The functionality and design of a NES is further discussed in NES section 5.1.

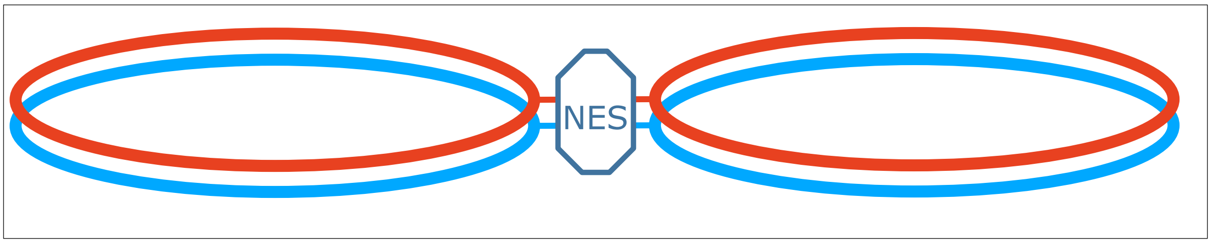


Figure 4.3: Two similar rings connected by a network exchange station (NES).

4.2.2 Hierarchical networks

The topology of a single network is no longer tree-structured, but the use of some form of tree structure in a large district heating and cooling system does make sense. It all has to do with matching thermal capacity. From an engineering and economic perspective, it is suboptimal to have a sizeable customer (e.g. a 15 MW datacentre) physically in the same network as a group of smaller customers (e.g. a 500 kW housing district).

In a hierarchical topology of networks, the sizing can be overcome by having one or more networks connect to a parent network. This parent network could on its turn, together with other child-networks, have its own parent. A hierarchical topology of three layers of ring networks is shown in Figure 4.4.

The number of layers in the hierarchy is unlimited. At any time a new layer can be added, either by creating a new parent ring network, or by creating a new child ring network.

Say for example that two nearby cities already have a two-layered network and decide that it is financially interesting to connect the two system, A new parent network is build connecting the two individual two-layered networks, creating a single three-layered network.

Say that a new block with apartments and a supermarket is being build in a city with a three-layered network. A block-level district heating and cooling network is preferred as the supermarket can provide heating to the individual apartments. This block-level network is then connected as a child to one of the existing networks at the third layer, thus creating a four-layered network.

Each connection between a parent and a child is realized using a network exchange station, where the NES must act as a balancing station for the child network. The design of a NES with balancing functionality is further discussed in section 5.1.

Additionally, the highest layered network requires a separate balancing station to ensure system-wide thermal balancing and mass balancing at network level. All other layers are balanced through their respective NES.

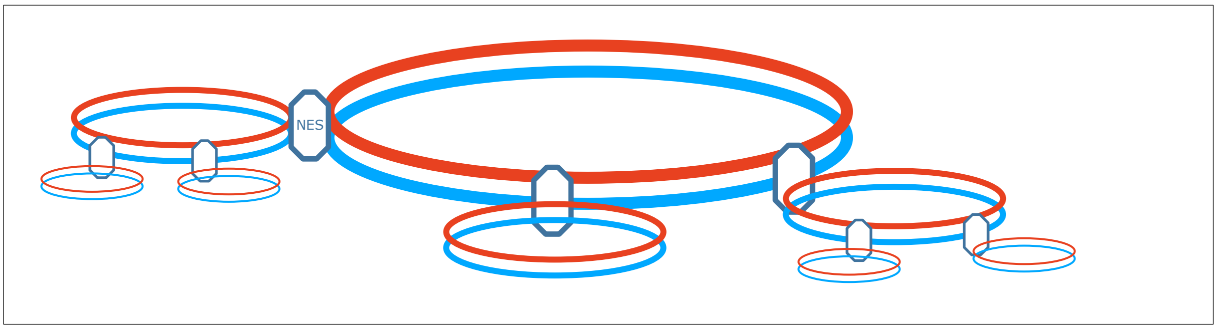


Figure 4.4: Hierarchical topology of three levels of district heating-cooling ring networks.

The layer in which a ring-network resides, does not necessarily say something about the thermal capacity of the network. A child network could have a higher thermal capacity than its parent network. This situation occurs when there is lots of thermal energy transfer between heating and cooling demanding customers, while the parent network is only used for balancing purposes.

To summarize, the characteristics of a hierarchical network are:

- Every network is connected to one, and only one, higher-level network, except for the highest-level network.
- A network is never connected to a network of the same level

- A network can have zero, one or multiple connections to networks that are one level lower.
- Each connection between two networks has a NES that functions as a balancing station for the network with the lowest level.
- The highest-level network requires a separate balancing station.

4.2.3 Meshed networks

A different approach from a hierarchically topology is a meshed topology. Instead of connecting to a single parent network, a district heating and cooling network connects to one or more sibling networks. This concept is sketched in Figure 4.5.

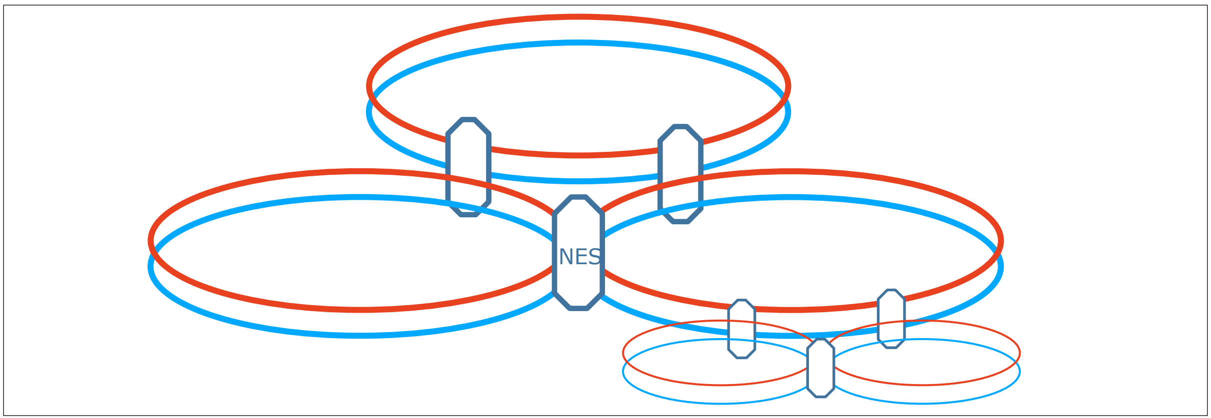


Figure 4.5: Example of a meshed network topology.

Networks are connected by a NES. As opposed in hierarchical topologies, the NES does not necessarily have to perform a balancing function. However, each network must have some form of balancing to ensure an equilibrium in mass flow. The balancing function could be performed by one or more NESs, one or more balancing stations or a combination of both.

Meshed network topology enables more decentralisation and freedom than a hierarchical network topology but are more difficult to scale up.

The advantages of meshed networks are:

- District sized networks are likely to touch other district sized networks due to the way city districts are designed. Meshed networks allow for direct thermal energy exchange of neighbouring networks.

- It provides more openness in the network. Citizens in a district could actively participate in a district sized network without depending on a single network company providing the backbone.

The disadvantages of meshed networks are:

- Long distance exchange of thermal energy may strain the network. If a large source of heating is at one side of the cluster and the demand is at the other side of the cluster, the thermal energy must go through all of the rings that are in between. The thermal capacity of this exchange is limited by the thermal capacity of the in-between network.
- If the in-between networks have different temperature levels, there is a risk of increased electricity consumption, reducing the cluster's overall efficiency.
- Meshed network clusters have a higher number of NESs than hierarchical networks, which may increase costs.
- Meshed networks are more complex. This could cause loss of overview and control. Decentralized technologies, such as multi-agent-based control, could provide solutions here.

Meshed networks work well if the amount of thermal energy exchanged between networks is relatively small compared to the thermal energy exchanged within a network.

It is expected that a combination between meshed networks at district level and hierarchical networks at building and city level is feasible, but more research is needed.

5. Network components

5.1 Balancing station (BAS)

The purpose of a balancing station or BAS is to balance the mass flow and thermal energy flow in a network.

The mass flow is balances by creating a short-circuit between the heat-pipe and cold-pipe as shown in Figure 5.1. If a mismatch occurs in mass flow in the network, this short-circuit allows away for the mismatch to compensate.

For example, customers are pumping a total of 10 litres per second from the heat-pipe to the cold-pipe. At the same time, other customers are pumping 6 litres per second from the cold-pipe to the heat-pipe. In the balancing station, 4 litres per second will flow through the short-circuit from the cold-pipe to the heat-pipe compensating the mass flow imbalance.

While this resolves the mass balance, there is still a mismatch in the thermal energy flow. Without additional measures, it would mean that warm and cold water are mixing, resulting in loss of useable thermal energy.

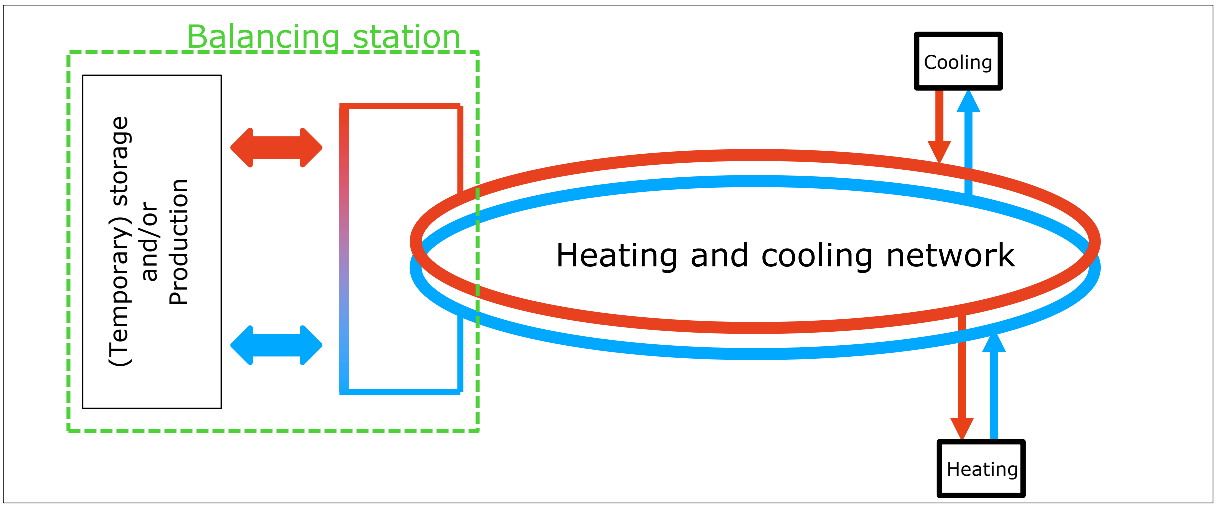


Figure 5.1: Concept drawing of a ring heating-cooling network with a balancing station.

Therefore, the balancing station must also be able to provide both heating and cooling, either by producing it or by using storage solutions. The balancing station may need a heat pump to provide sufficient flexibility and control matching the temperature of the heat and cold source to that of the network.

It is possible for a network to have more than one balancing station. The mismatch in flow and energy is then distributed over the balancing stations. The distribution depends purely on the friction resistance of the network but could be partially controlled by using valves in the short-circuit.

The advantage of multiple balancing stations is the dispersion of storage locations. For example, if a single storage location doesn’t provide enough space. This could be the case with underground storage solutions.

Balancing stations can be combined with NESs, by either by integrating them or by being physically present in the same building, but with its own network connection.

5.2 Network exchange station (NES)

The network exchange station allows thermal energy transfer between two networks, while keeping them hydraulically separated. The design, operation and objectives for a NES varies depending on the topology of the system and the intend of the designer.

There are three base types of network exchange stations: trading, balancing of one network by the other network and balancing both networks. A great number of variants can exist on these base types.

5.2.1 Trade NES (T-NES)

A network exchange station (NES) can be used for the simple purpose of trading thermal energy between two networks. Thermal energy is transferred from one network to the other, whereby the owner of the NES has full freedom and control over the amount of thermal energy and the direction the thermal energy is transferred to.

Such a network exchange station is called a Trade NES or T-NES. Its business decision is purely based on economics. Money is earned because there is a price difference for thermal energy between the two networks.

In both networks, the T-NES acts as purely as a customer. If the T-NES transfers thermal energy, it demands cooling (supplies heating) in one network and demands the same amount of heating (supplies cooling) in the other network.

The T-NES has no balancing functionality. This means that each connected network must have a different way of balancing its mass flow and thermal flow.

5.2.2 Transmission-Distribution NES (TD-NES)

In a 'transmission-distribution' setting, the transmission network is used to balance the distribution network. A Transmission-Distribution NES or TD-NES is thus used to balance one network with the other network.

TD-NES are the type of NES used in hierarchical network topologies but are optional in other topologies. The TD-NES has a pure technical function in a district heating and cooling system, there is no economic driver that decides over the operation of the TD-NES.

A TD-NES balances the mass flow, by having an open connection between the heat-pipe and cold-pipe at the distribution network side. An imbalance in mass flow in the distribution network results in a mass flow through the heat exchanger in the TD-NES. The imbalance of thermal energy in the distribution network then needs to be compensated by transferring thermal energy from the transmission network. (bi-directional) pump at the transmission network side that is controlled by a central controller will ensure this.

The TD-NES thus acts as a full balancing station in the distribution network and as a customer in the transmission network. Note that the transmission network requires a means of balancing for itself.

5.2.3 Dual Network Balancing NES (DNB-NES)

Two connected networks can both be balanced by a single NES. Such a NES is called a Dual Network Balancing NES or DNB-NES. This type of NES performs a triple function: the balancing function in each of the networks and thermal energy transfer between the networks.

The DNB-NES has a pure technical function in a district heating and cooling system, there is no economic driver that decides over the operation of the DNB-NES.

5.3 Heat interface unit (HIU)

The heat interface unit or HIU is the unit located at the customers premises and provides the customer with a connection to the district heating and cooling network. The design of a HUI can have one or more of the following components:

- A connection to the heat-pipe and cold-pipe
- A heat pump to segregate temperature levels of the network from those at the customer's end.
- A bidirectional variable speed pump to regulate the flow between heat-pipe and cold-pipe.
- A two-way thermal energy meter to measure the exchange of thermal energy from and to the network.

6. Network operations

This chapter discusses how a low temperature district heating and cooling network is operated.

6.1 Balancing

Balancing of a district heating and cooling network can be performed in several ways:

- **Thermal energy exchange between networks**. If one network has an excess in heat and another network simultaneously has a shortage in heat, they can exchange thermal energy to minimize this mismatch.
- **STSS: Short-term storage solutions**. These types of solutions typically work when there is mismatch in demand and supply profiles that reverses throughout the day or week. For example, if there is excess heat during the day from industry, but there is a shortage of heat during the evening for houses, a short-term storage solution can store the excess heat during the day and provide it during the evening. through e.g. buffer vessels. This type of solution is thus best used for peak shaving and can be realized with e.g. hot and cold-water buffer vessels.
- **LTSS: Long term storage solutions**. These types of solutions typically work when there are seasonal variations. For example, a large business area with lots of offices requires cooling in the summer and heat in the winter. But average over the years, its net thermal energy demand is low. Seasonal storage replaces the need for heat and cold production units during respectively the winter and summer season. Typical technologies for long term storage are e.g. underground heat-cold-storage, phase change materials and thermochemical storage.
- **Production**: If the imbalance between heat and cold demand is not temporarily, but structurally, i.e. there is a net demand for either cooling or heating in the long term, production of either heat and or

cold may be required. Production may also be required if storage solutions are not available. Different production sources are further discussed in 0.

Which balancing solutions are required in a system is determined by the demand and supply profiles for heating and cooling over time and how they (mis)match. An example is given in Figure 6.1. An industrial facility has additional waste available during the day, when production is peaking (blue). A group of houses connected on the same heating-cooling network have a heat demand throughout the day but peaking in the evening hours (red). The mismatch becomes clear when the two profiles are added (green). By using a short-term storage solution, the excess heat is stored and released again in the evening. This results in a net profile with significant lower peaks (purple).

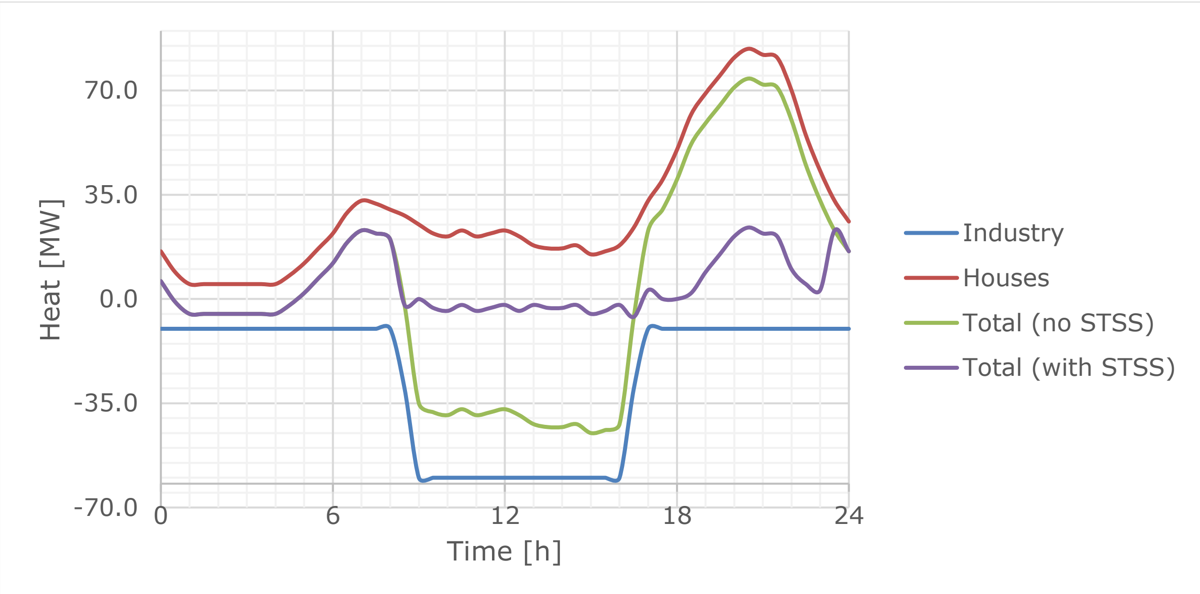


Figure 6.1: An example of a short-term storage solution (STSS), where the mismatch between industrial heat production and heat demand from households is mitigated using storage.

6.1.1 Control signal

When there is a thermal imbalance in the network (but not a mass imbalance), the temperature in the network increases of decreases. Say that there is more heat demand than cold demand. In the balancing station, there is a mass balancing flow from the cold-pipe to the heat-pipe. If no thermal exchange takes place, the mass balancing flow (with the temperature of the cold-pipe) mixes with the water in the heat-pipe, causing the temperature in the heat-pipe to drop. As the heat-pumps in the system maintain the temperature difference between the two pipes, the temperature of the cold-pipe also starts to drop.

Reversely, if there is more cold demand than heat demand and no thermal exchange at the balancing station, the temperature in the pipes will start to increase.

While only the balancing station can measure imbalance through mass flow, any actor in the network can measure imbalance through temperature fluctuations in the network. The temperature fluctuations act as a control signal in the same way voltage frequency does for electricity grids.

6.2 Storage sizing

A balancing solution can be sized using the aggregated demand profile of the network.

Assume that $b(t)$ is the aggregated demand function in time where $b(t) > 0$ is a net heating demand and $b(t) < 0$ is a net cooling demand.

The cumulative imbalance at time t within the interval $t \in [t_a, t_b]$ is given by:

$$B(t) = \int_{t_a}^{t} b(t)\,dt \quad t \in [t_a, t_b] \tag{6.1}$$

The cumulative imbalance at the start of the interval is zero, i.e. $B(t_a) = 0$.

The total structural imbalance is the imbalance that remains at the end of the interval and is given by $B(t_b)$. The true imbalance at any given time within the interval is therefore given by

$$\bar{B}(t) = B(t) - B(t_b)\,\frac{t}{t_b} \tag{6.2}$$

From here, the required storage size equals the maximum absolute imbalance:

$$\max|\bar{B}(\hat{t})| \tag{6.3}$$

Where $B(\hat{t})$ are the values of the local maxima and minima, which can be found by solving for $\hat{t}$:

$$b(\hat{t}) = 0 \quad \forall \hat{t} \in [t_a, t_b] \tag{6.4}$$

Many types of heating and cooling demand has periodic behaviour, such as the summer/winter day/night or weekdays/weekends cycles. Structural imbalance therefore greatly depends on the chosen interval. Take for example an office building with approximately a summer cooling demand that equals a winter heating demand. If the time interval equals a year, the structural imbalance is very low, as the heating and cooling demand cancel each other out. If the time interval only contains the winter months, there is a high structural imbalance.

As such, choosing a proper interval is important. The choice should be based on the periodic behaviour of the aggregated demand curve, but also the time frame a storage solution works best on. This is a case-by-case engineering task.

Periodic behaviour can be analysed by looking for local maxima in the frequency space, by applying a Fourier Transform on the aggregated demand curve.

To size both short term and long-term storage, first short term storage is sized over interval $t \in [t_a, t_b]$. Then the long-term storage is sized for time interval $t \in [t_c, t_d]$ following:

$$\max|\bar{B}_{cd}(\hat{t}_{cd})| - \max|\bar{B}_{ab}(\hat{t}_{ab})| \tag{6.5}$$

6.3 Production

Production of low temperature heat energy and high temperature cold energy can be performed with sustainable energy sources, of which a number have been listed below. The use of waste heat is not, as it usually comes forth from a demand for cooling and thus, a 'normal' customer of the network. Combining sources with a heat pump leads to a higher utilization of the source, at the expense of electricity consumption.

6.3.1 Heat sources

- **Flat plate solar collectors**: A type of solar collector that uses a flat (copper) plate with a solar irradiation absorbent coating. Water through piping underneath the plate collects the heat. The plate and piping are imbedded in a layer of insulation and a glass sheet on top for protection. Works best in environments ambient temperatures above 0 °C.
- **Vacuum solar collectors**: A type of solar collector that uses an inner tube covered solar irradiation absorbent coating, in a vacuum glass tube. Water through the inner tube collects the heat. The vacuum provides extra insulation, making these types also work well in ambient temperatures below 0 °C.
- **PVT**: A flat plate solar collector with photovoltaic cells (PV) on top. Although the heat yield is significantly lower that that of a flat plate collector, the sum of the electric and thermal efficiency is higher than for individual systems. The cooling effect on the PV cells, give the electric efficiency another boost.
- **Geothermal**: As the earth's core is superhot, the closer one gets to the core, the warmer it gets. The deeper one digs, the warmer it gets. The geothermal gradient is generally about 20-30 °C/km and 0.04-0.08 W/m^2. Near tectonic plate boundaries, these numbers may be significant higher as magma resides a lot closer to the surface. In Iceland, the geothermal gradient has been measured over 200 °C/km.
- **Surface water**: Depending on the local climate, surface water may reach temperatures over 25 °C and could therefore be used as a source, although in Europe this is mainly applicable for the Mediterranean Sea during the summer months.
- **Air**: During summer months, the ambient air may reach temperatures well over 25 °C. Using an air fan with a heat exchanger, heat can be captured into the heating network.

6.3.2 Cold sources

- **Surface water**: Depending on the local climate, surface water may reach temperatures below 15 °C and could therefore be used as a source. In the Netherlands, sea temperatures rarely exceed the 15 °C.

- **Air**: In the Netherlands, 6 to 10 months a year, the ambient air may reach temperatures below 15 °C. Using an air fan with a heat exchanger, cold can be captured into the cooling network.

7. Design guide

This design guide takes a step by step approach to design a low temperature district heating and cooling system. These steps may need to be repeated as different configurations lead to different business cases.

Step 1. Determine heating and cooling demand profiles

The **aggregated** method determines the profiles based on a statistical approach. For example, the total yearly heating demand of a block of houses is distributed over time based on weather information.

The **individual** method determines the profile of each individual customer through advanced modelling or by using measurement data.

The **hybrid** method combines the aggregated and individual method. For smaller similar customers (e.g. a housing district), the aggregated method can be used, while the individual method is applied to larger customers (e.g. supermarket, data centre.)

Step 2. Determine balancing and production option

Balancing and production options are determined in four steps.

a. A short-term mismatch between heating and cooling demand (e.g. day/night or weekday/weekend cycles) is resolved by using short-term balancing solutions, such as buffer vessels. The size of the vessels and the amount of energy matched is determined through equations in section 6.2
b. A long-term mismatch between heating and cooling demand (e.g. winter/summer season cycle) is resolved by using long-term balancing solutions. The size of the storage solution and the amount of energy matched is determined through equations in section 6.2.
c. Any-term mismatches may be resolved by connecting the network to other networks.

d. A structural mismatch between heating and cooling demand, or a mismatch that cannot be matched with a balancing solution, requires heating and/or cooling production units. The size of structural mismatch is determined through equations in section 6.2. The type of production units depends on the technological and economic feasibility.

Step 3. Determine topology

Determine the physical topology of the network, i.e. where are the pipes running such that it connects to all the customers. This provides the total length of the network.

Step 4. Determine thermal capacity

The thermal capacity of the network is determined by one of the following methods:

The **simple** method estimates the peak load demand for heating and cooling separately. The thermal capacity of the network equals the highest peak load of the two.

The **optimal** method models the thermal demand curve (heating demand minus cooling demand for each moment in time) for every individual customer connection. The thermal flows in the network are then calculated for each time interval. The network section with the highest thermal flow at any given time interval is the thermal capacity of the network.

Step 5. Determine pipe size

Assuming a worst-case scenario for the friction factor, best-case scenario for the pressure drop and recommended temperature difference, the pipe size equation (3.5) can be simplified to:

$$[\text{Inner pipe diameter (m)}] = 0.206 \times [\text{Thermal capacity (MW)}]^{0.4} \tag{7.1}$$

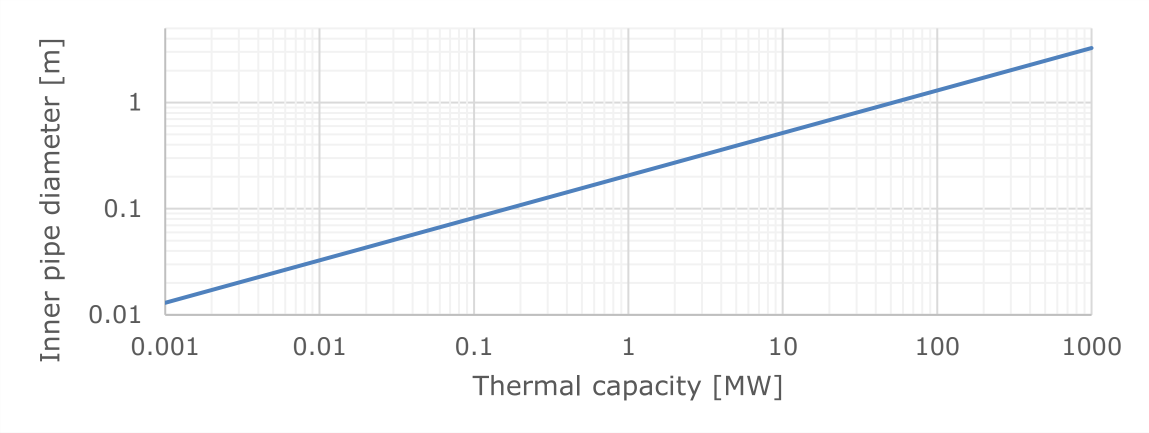


Figure 7.1: Inner pipe diameter as function of the thermal capacity of the network.

Step 6. Determine pump size and pump configuration

- A pump is selected following:
  - Determine normal operation conditions of the network (individual pump speed, required flow and simultaneousness)
  - The expected pressure drop under these conditions.
  - A variable speed pump with a pump curve that matches the flow and pressure drop in for the given pump speed.
- The risk of reaching the limits of the individual pumps in a worst-case scenario is determined. Additional measures are taken which can include
  a. Thermal buffers to reduce the simultaneousness of the pumps.
  b. Intelligent control to reduce the simultaneousness of the pumps.
  c. Management of the temperature difference to reduce the flow speed.
  d. More local diversity in heat and cold demand.
  e. A centralized pump to provide a pressure boost.

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9. Equation derivations

This chapter provides derivations for equations used in this document.

9.1 Symbols

Symbol	Unit	Description
d_i	[m]	the inner diameter of a pipe
D_i	[m]	the inner diameter of the insulation layer of a pipe
D_o	[m]	the outer diameter of the insulation layer of a pipe
L	[m]	the length of the pipe
h	[m]	the underground depth measured from surface edge to the centre of the pipe
s	[m]	the horizontal distance between the two pipes, measured from the centre of the pipes.
ϵ	[m]	the pipe surface roughness
f	[-]	the friction factor of the pipe
v	[m/s]	the velocity of the fluid through a pipe
Φ_v	[m^3/s]	The volumetric flow of the fluid through a pipe
Re	[-]	the Reynolds number
μ	[Pa s]	the dynamic viscosity of the fluid.
c_p	[J kg^{-1} K^{-1}]	the specific thermal energy of the fluid
ρ	[kg/m^3]	the volumetric density of the fluid
Δp	[Pa]	The pressure drop in the system
θ	[K]	the temperature difference between two mediums.
θ_{xy}	[K]	the temperature difference between medium x and medium y.
Θ_{hc}	[K]	the temperature difference between the heat pipe and cold-pipe in a low temperature district heating and cooling network.

Θ_{pa}	[K]	the temperature difference between pipe and ambient air.
λ_i	[Wm^{-1}K^{-1}]	the specific thermal conductivity of the insulation layer.
λ_g	[Wm^{-1}K^{-1}]	the specific thermal conductivity of the ground or soil
P	[W]	thermal power or heat flow
P_{th}	[W]	Heat flow specifically to indicate the thermal capacity of a system.
P_{loss}	[W]	the thermal loss of a pipe towards the ambient air.
P_{hc}	[W]	the thermal heat transfer between heat-pipe and cold-pipe.
t	[s]	time
T	[°C]	temperature

9.2 Flow

The flow of a fluid through a pipe is given by

$$\Phi_v = \frac{\pi}{4} d_i^2 v \tag{9.1}$$

9.3 Heat capacity equation

The thermal capacity of water in a pipe can be derived from the well-known heat capacity equation.

$$\begin{aligned} P_{th} &= \frac{dQ}{dt} \\ &= \frac{d}{dt}\left(m c_p \Delta T\right) \\ &= c_p \Delta T \frac{d}{dt}(\rho A L) \\ &= \frac{\pi d_i^2}{4} \rho c_p \Delta T \frac{d}{dt}(L) \\ &= \frac{\pi}{4} \rho c_p \Delta T d_i^2 v \end{aligned} \tag{9.2}$$

From which equation (3.1) is derived.

9.4 Heat transfer equations

The heat flux for any object can be calculated through Fourier's Law [6].

$$\vec{q} = -\lambda \nabla T = \frac{P_{loss}}{A} \tag{9.3}$$

For a pipe with a relatively small diameter compared to its length ($d \ll L$), this equation can be rewritten into a one-dimensional radial equation:

$$\begin{aligned} P_{loss} &= -A\lambda \frac{dT}{dr} \\ &= -2\pi r L \lambda \frac{dT}{dr} \end{aligned} \tag{9.4}$$

For a metal pipe wrapped in an insulation layer, the heat transfer through the metal is neglectable. Only the insulation layer is of importance. By integration both sides of equation (9.4) one obtains:

$$\begin{aligned} P_{loss} \int_{\frac{D_i}{2}}^{\frac{D_o}{2}} \frac{dr}{r} &= -2\pi\lambda L \int_{T_p}^{T_a} dT \\ P_{loss} [\ln r]_{\frac{D_i}{2}}^{\frac{D_0}{2}} &= -2\pi\lambda L [T]_{T_p}^{T_a} \\ P_{loss} \ln\left(\frac{D_o}{D_i}\right) &= -2\pi\lambda L (T_a - T_p) \\ P_{loss} &= \frac{2\pi\lambda L \Theta_{pa}}{\ln\left(\frac{D_o}{D_i}\right)} \\ P_{loss} = \frac{\Theta_{pa}}{R_i} \quad R_i &= \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi\lambda L} \end{aligned} \tag{9.5}$$

Which gives the results in equation (3.7).

For a pipe buried in the ground, the thermal resistance is given by [4][5]:

$$R_g = \frac{\ln\left(\frac{2h}{D_o} + \sqrt{\left[\frac{2h}{D_o}\right]^2 - 1}\right)}{2\pi\lambda L} \tag{9.6}$$

The actual derivation requires a solution to a complex differential equation and is left out in this report. if the outer pipe diameter is relatively small compared to the depth, i.e. $h \gg D_o$, equation (9.6) can be simplified to:

$$R_g = \frac{\ln\left(\frac{4h}{D_o}\right)}{2\pi\lambda L} \tag{9.7}$$

In practice 'relatively small' is $h \geq 2D_o$.

For a pipe with multiple layers of insulation and/or a pipe that is buried underground, the thermal loss is calculated following:

$$P_{loss} = \frac{\Theta_{pa}}{\sum R_k} \tag{9.8}$$

and such obtaining equation (3.9).

The temperature drop of the water in the pipe, as posited in equation (3.10), is calculated by knowing that the reduction in thermal capacity of the water in equation (9.2) equals the thermal losses in equation (9.8):

$$\begin{aligned} P_{th} &= -P_{loss} \\ \frac{\pi}{4}\rho c_p \Delta T d_i^2 v &= -\frac{\Theta_{pa}}{\sum R_k} \\ \Delta T &= -\frac{4}{\pi\rho c_p d_i^2 v}\frac{\Theta_{pa}}{\sum R_k} \end{aligned} \tag{9.9}$$

This steady-state equation is valid only if $\Delta T \ll \Theta_{pa}$, which holds up even for very a low flow in a small pipe. However, if there is no flow at all, the water in the pipes starts to cool down in time. The heat capacity equation is derived differently, as the temperature is now a function of time.

$$\begin{aligned} -P_{loss} = P_{th} &= \frac{dQ}{dt} \\ \frac{T(t) - T_a}{\sum R_k} &= \frac{d}{dt}\left(mc_p(T(t) - T_a\right) \\ \frac{T(t) - T_a}{\sum R_k} &= \frac{\pi}{4}d_i^2 \rho L c_p \left[\frac{d\big(T(t)\big)}{dt} - T_a\right] \end{aligned} \tag{9.10}$$

The resulting first-degree differential equation can be solved by substituting

$$T(t) = \mathrm{X}e^{-Yt} + \mathrm{Z} \tag{9.11}$$

Knowing the boundary condition $T(0) = T_p$, the differential equation can be solved:

$$T(t) = \Theta_{pa} e^{-\frac{4t}{\pi d_i^2 \rho L c_p \sum R_k}} + \mathrm{T_a} \tag{9.12}$$

If there is a flow, but it very small, equation (9.9) must be rewritten as

$$\frac{d\left(T_p(l)\right)}{dl} = -\frac{4}{\pi \rho c_p d_i^2 \frac{dl}{dt}} \frac{T_p(l) - T_a}{\sum R_k(l)} \tag{9.13}$$

For which only a numerical solution can be found.

9.5 Pressure drop

The pressure drop is calculated using the Darcy-Weisbach equation

$$\Delta p = -\frac{1}{2}\frac{L}{d_i} f \rho v^2 \tag{9.14}$$

If equation (9.2) is rewritten as:

$$\begin{aligned} P_{th} &= \frac{\pi}{4}\rho c_p \theta_{hc} d_i^2 v \\ v &= \frac{4P_{th}}{\pi \rho c_p \theta_{hc} d_i^2} \end{aligned} \tag{9.15}$$

And equation (9.15) is then substituted in equation (9.14), the pressure drop equation is obtained:

$$\begin{aligned} \Delta p &= -\frac{1}{2}\frac{L}{d_i} f \rho \left(\frac{4P_{th}}{\pi \rho c_p \theta_{hc} d_i^2}\right)^2 \\ &= -\frac{8fL}{d_i^5 \pi^2 \rho}\left(\frac{P_{th}}{c_p \Theta_{hc}}\right)^2 \end{aligned} \tag{9.16}$$

10. Data

10.1 Thermal conductivity of soil

The thermal conductivity of different types of soil have been listed below. The data has been taken from [9].

Soil type	Water content (%)	Bulk density (Mg/m^3)	Dry density (Mg/m^3)	Thermal conductivity (W m^{-1} K^{-1})	Specific heat capacity (J kg^{-1} K^{-1})
BH C13 88	21.3	1920	1583	2.89	1520
China CLAY (D)(sat.)	46.2	1730	1183	1.52	2362
China CLAY (D)(dry)	0	1390	1390	0.25	800
Sandy CLAY	26.5	1890	1494	1.61	1696
Sandy CLAY	19.5	2100	1757	2.45	1459
Soft dark grey sandy gravely CLAY	28.5	1912	1488	3.57	1764
Soft grey fine sandy CLAY	54.6	1650	1067	4.20	2646
Soft grey fine sandy CLAY	41.4	1741	1231	3.03	2200
Stiff dark grey sandy gravely CLAY	10.1	2299	2088	3.69	1141
Stiff dark grey sandy gravelly CLAY	9.6	2369	2161	3.28	1125
Stiff grey brown sandy gravelly CLAY	9	2352	2158	3.20	1104
Very soft grey fine sandy CLAY	46.2	1711	1170	3.51	2362
Grey slightly silty sandy GRAVEL	11.1	1983	1785	4.44	1175
Grout	166	1250	470	0.64	6412
Grey limestone (very hard)	0.1	2690	2687	2.54	803
Course SAND (dry)	0	1800	1800	0.25	800
Course SAND (sat.)	20.2	2080	1730	3.72	1483
Dark grey clayey fine sand/silt	28	1848	1444	4.26	1747
Fine SAND (dry)	0	1600	1600	0.15	800
Fine SAND (sat.)	24.6	2010	1613	2.75	1632
Made ground (Silty gravely sand)	13.9	2182	1916	5.03	1270
Medium SAND (dry)	0	1700	1700	0.27	800
Medium SAND (sat.)	20.2	2080	1730	3.34	1483

10.2 Pipe data

10.2.1 Prinspipe type 1

Pipe with an inner steel cylinder, an insulation layer of PUR and a jacket. Made by Weijers-Waalwijk.

DN	d_i (mm)	D_i (mm)	D_o (mm)	Mass (kg/m)	Fluid volume (l/m)	Standard length (m)
20	21.7	26.9	90	2.76	0.37	6
25	28.5	33.7	90	3.17	0.67	6
32	37.2	42.4	110	4.56	1.09	6/12
40	43.1	48.3	110	5.08	1.46	6/12
50	54.5	60.3	125	6.30	2.33	6/12
65	70.3	76.1	140	7.79	3.88	6/12
80	82.5	88.9	160	9.22	5.35	6/12
100	107.1	114.3	200	13.34	9.01	6/12/16
125	132.5	139.7	225	16.21	13.79	6/12/16
150	160.3	168.3	250	21.10	20.18	6/12/16
200	210.1	219.1	315	31.36	34.67	6/12/16
250	263.0	273.0	400	45.49	54.33	6/12/16
300	312.7	323.9	450	58.90	76.80	6/12/16
350	344.4	355.6	500	67.02	93.16	6/12/16
400	393.8	406.4	560	85.25	121.80	6/12/16
450	444.6	457.2	630	99.11	155.25	6/12/16
500	495.4	508.0	710	115.50	192.75	6/12/16
600	595.8	610.0	800	150.20	278.80	6/12/16
700	695.0	711.0	900	190.10	379.37	6/12/16
800	795.4	813.0	1000	232.80	496.98	6/12/16
900	894.0	914.0	1100	288.70	627.72	6/12
1000	994.0	1016.0	1200	346.90	776.00	6

Thermal conductivity of the PUR insulation is $\lambda_{pur} = 0.026\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the jacket is $\lambda_{jacket} = 0.4\, Wm^{-1}\, K^{-1}$

10.2.2 Prinspipe type 2

Pipe with an inner steel cylinder, an insulation layer of PUR and a jacket. Made by Weijers-Waalwijk.

DN	d_i (mm)	D_i (mm)	D_o (mm)	Mass (kg/m)	Fluid volume (l/m)	Standard length (m)
20	21.7	26.9	110	3.19	0.37	6
25	28.5	33.7	110	3.60	0.67	6
32	37.2	42.4	125	5.01	1.09	6/12
40	43.1	48.3	125	5.44	1.46	6/12
50	54.5	60.3	140	6.69	2.33	6/12
65	70.3	76.1	160	8.36	3.88	6/12
80	82.5	88.9	180	9.84	5.35	6/12
100	107.1	114.3	225	14.44	9.01	6/12/16
125	132.5	139.7	250	17.56	13.79	6/12/16
150	160.3	168.3	280	22.85	20.18	6/12/16
200	210.1	219.1	355	34.34	34.67	6/12/16
250	263.0	273.0	450	50.02	54.33	6/12/16
300	312.7	323.9	500	64.08	76.80	6/12/16
350	344.4	355.6	560	74.01	93.16	6/12/16
400	393.8	406.4	630	94.15	121.80	6/12/16
450	444.6	457.2	670	104.90	155.25	6/12/16
500	495.4	508.0	800	130.20	192.75	6/12/16
600	595.8	610.0	900	165.90	278.80	6/12/16
700	695.0	711.0	1000	207.40	379.37	6/12/16
800	795.4	813.0	1100	251.90	496.98	6/12/16
900	894.0	914.0	1200	310.30	627.72	6/12

Thermal conductivity of the PUR insulation is $\lambda_{pur} = 0.026\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the jacket is $\lambda_{jacket} = 0.4\, Wm^{-1}\, K^{-1}$

10.2.3 Prinspipe type 3

Pipe with an inner steel cylinder, an insulation layer of PUR and a jacket. Made by Weijers-Waalwijk.

DN	d_i (mm)	D_i (mm)	D_o (mm)	Mass (kg/m)	Fluid volume (l/m)	Standard length (m)
20	21.7	26.9	125	3.55	0.37	6
25	28.5	33.7	125	3.96	0.67	6
32	37.2	42.4	140	5.40	1.09	6/12
40	43.1	48.3	140	5.83	1.46	6/12
50	54.5	60.3	160	7.25	2.33	6/12
65	70.3	76.1	180	8.97	3.88	6/12
80	82.5	88.9	200	10.62	5.35	6/12
100	107.1	114.3	250	15.74	9.01	6/12/16
125	132.5	139.7	280	19.31	13.79	6/12/16
150	160.3	168.3	315	25.07	20.18	6/12/16
200	210.1	219.1	400	38.03	34.67	6/12/16
250	263.0	273.0	500	55.19	54.33	6/12/16
300	312.7	323.9	560	71.07	76.80	6/12/16
350	344.4	355.6	630	82.91	93.16	6/12/16
400	393.8	406.4	670	99.92	121.80	6/12/16
450	444.6	457.2	710	110.80	155.25	6/12/16
500	495.4	508.0	900	145.90	192.75	6/12/16
600	595.8	610.0	1000	183.20	278.80	6/12/16
700	695.0	711.0	1100	226.50	379.37	6/12/16
800	795.4	813.0	1200	273.60	496.98	6/12/16

Thermal conductivity of the PUR insulation is $\lambda_{pur} = 0.026\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the jacket is $\lambda_{jacket} = 0.4\, Wm^{-1}\, K^{-1}$

10.2.4 Coolmant

Rigid pipe with an inner polyethylene cylinder, an insulation layer of PUR and a jacket / casing of polyethylene. Made by Brugg Pipesystems.

Type	d_i (mm)	D_i (mm)	D_o (mm)	Fluid volume (l/m)
SDR11 125/225	102.2	125	218.0	8.203
SDR11 140/225	114.6	140	218.0	10.315
SDR11 160/250	130.8	160	242.2	13.437
SDR11 180/280	147.2	180	271.2	17.018
SDR11 200/315	163.6	200	305.2	21.021
SDR11 225/315	184.0	225	305.2	26.590
SDR11 250/355	204.6	250	343.8	32.878
SDR11 280/400	229.2	280	387.4	41.259
SDR11 315/450	257.8	315	436.0	52.198
SDR17 125/225	110.2	125	218.0	9.230
SDR17 140/225	123.4	140	218.0	11.960
SDR17 160/250	141.0	160	242.2	15.610
SDR17 180/280	158.6	180	271.2	19.760
SDR17 200/315	176.2	200	305.2	24.380
SDR17 225/315	198.2	225	305.2	30.850
SDR17 250/355	220.4	250	343.8	38.150
SDR17 280/400	246.8	280	387.4	47.840
SDR17 315/450	277.6	315	436.0	60.520

Thermal conductivity of the inner pipe is $\lambda_{pipe} = 0.4\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the PUR insulation is $\lambda_{pur} = 0.024\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the jacket is $\lambda_{jacket} = 0.33\, Wm^{-1}\, K^{-1}$

10.2.5 Coolflex

Flexible pipe with an inner polyethylene cylinder, an insulation layer of PUR and a jacket / casing of polyethylene. Made by Brugg Pipesystems.

DN	Type	d_i (mm)	D_i (mm)	D_o (mm)	Fluid volume (l/m)
20	25/76	20.4	25.00	74.0	0.327
25	32/76	26.2	32.00	74.0	0.539
32	40/91	32.6	40.00	88.6	0.835
40	50/91	40.8	50.00	88.6	1.307
50	63/126	51.4	63.00	123.0	2.091
65	75/126	61.4	75.00	123.0	2.961
80	90/162	73.6	90.00	157.0	4.254
100	110/162	90.0	110.00	157.0	6.362
125	125/182	102.2	125.00	176.0	8.200

Thermal conductivity of the inner pipe is $\lambda_{pipe} = 0.4\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the PUR insulation is $\lambda_{pur} = 0.0234\, Wm^{-1}\, K^{-1}$

Thermal conductivity of the jacket is $\lambda_{jacket} = 0.33\, Wm^{-1}\, K^{-1}$