IEA Heat Pumping Technologies Annex 47

Heat Pumps in District Heating and Cooling Systems

Task 3: Review of concepts and solutions of heat pump integration

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Project website
https://heatpumpingtechnologies.org/annex47/

March 2019

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Synopsis

The present report contains methods for the selection, design and integration of heat pumps for district heating systems. The data and information were determined by means of literature research and expert interviews. This document is a summary of the investigations of heat pumps in district heating networks in general and describes the status and technical solutions for their integration. In addition, a decision-support tool will be presented, which will allow a preliminary assessment of feasibility and cost-effectiveness.

1. Introduction

Driven by an increasing share of fluctuating power sources such as PV and wind energy and simultaneously falling electricity prices and the widespread use of heat networks, large heat pumps have been used in heat networks since the 1980s in Sweden and later also in Norway [1]. Currently, about 7% of the heat supplied to district heating in Sweden is provided by heat pumps (HP) [2]. In these district heating networks, mainly waste water, sea and river water but also industrial waste heat is used as sources for the heat pumps. For instance, in Stockholm, in addition to conventional generators, Europe's largest district heating network is operated by means of many decentralized heat pumps. These heat pumps use waste heat from a wide variety of processes. Often, data centers and supermarkets, which have a year-round need for cooling, use heat pumps to satisfy their cooling needs and at the same time to feed-in the heating network. For this purpose, a separate online marketplace for waste heat feed-in was created, in which prices and quantities are traded daily, see [3]. In addition to the heat input, it is also possible to supply the (excess) cold provided by the heat pumps to the district cooling network. Figure 1 shows the concept of so-called "Open District Heating" in Stockholm.

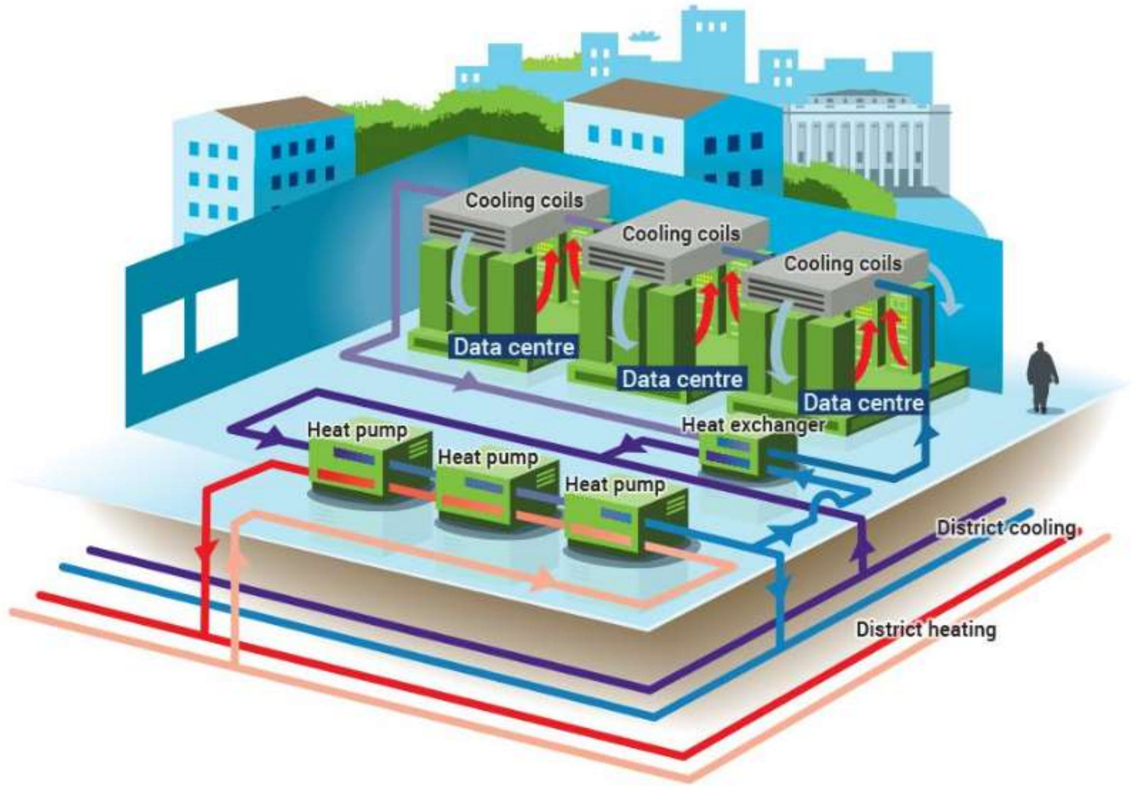


Figure 1 - Schema Open District Heating [1]

Heat pumps can be operated more economically, with small temperature difference between the heat source and the heat sink (see Section 2.5.2). In order to achieve optimum operating conditions, the supplied heat network should be operated with the lowest possible flow temperatures. In Sweden, the network temperatures are significantly lower than in Austria (compare Figure 2 and Figure 3), which enables a high proportion of heat pumps.

If the required network temperatures exceed the application limit of the heat pump or if efficient operation due to an increased temperature difference between heat source and heat sink cannot be guaranteed, heat should be provided by additional heat generators. Networks with low system temperatures such as secondary networks may be preferred for the installation of a heat pump.

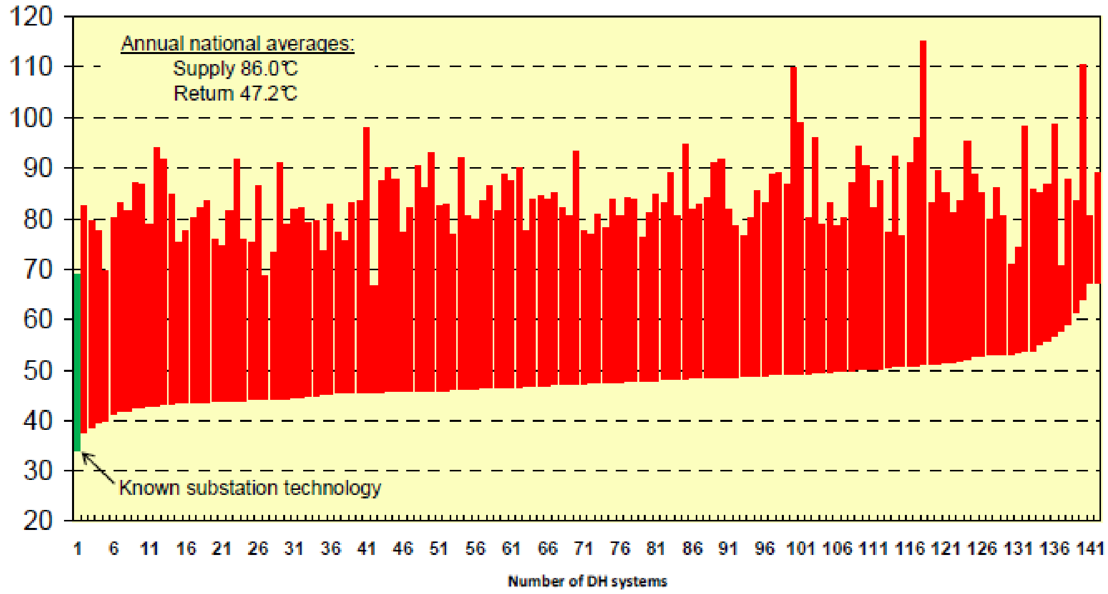


Figure 2 - Flow and return temperatures of various district heating networks in Sweden [5]

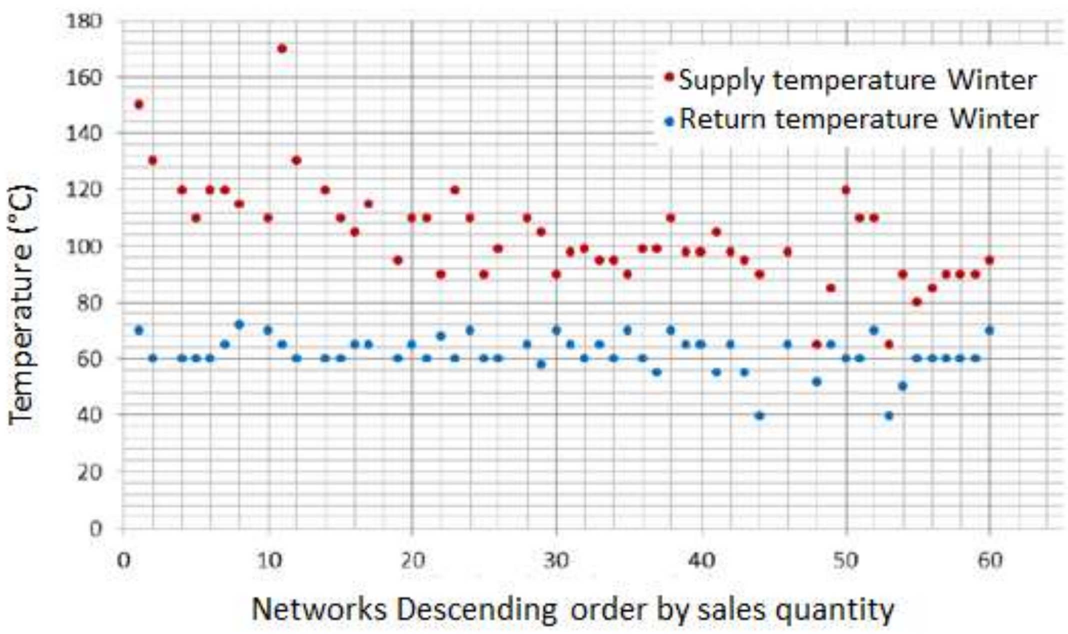


Figure 3 - Flow and return temperatures of various district heating networks in Austria [6]

Subsequently, relevant aspects for the integration of heat pumps into district heating networks will be discussed and general guidelines for selection and design will be described. These are based on technical and energetic analyzes and not on economic parameters. During various preliminary projects, investment costs and target price quotations for heat pumps for district heating systems were repeatedly obtained. The results showed that these costs vary greatly (sometimes with deviations over 300%) and always have validity only for specific applications. As a result, this report does not include the economic analysis and focus on the technical guidelines for the integration of heat pumps. Nonetheless, economic factors are included in the Excel tool described later (Section 3.1), but this requires prior cost research by the user for the specific case.

2. Method for selection, design and integration of heat pumps

2.1 Heat pump market in Europe

The European heat pump market, starting in 2005, shows a similar course to the Austrian market (see Figure 4). In total, around 8.3 million units have been sold since 2005. These are mainly heat pumps of small capacity for the heating of single and multi-family houses. Heat pumps for the use in thermal networks, as they exist in the Scandinavian countries, are an exception.

For further see task 1 of the Annex 47 project, which contains information on country level.

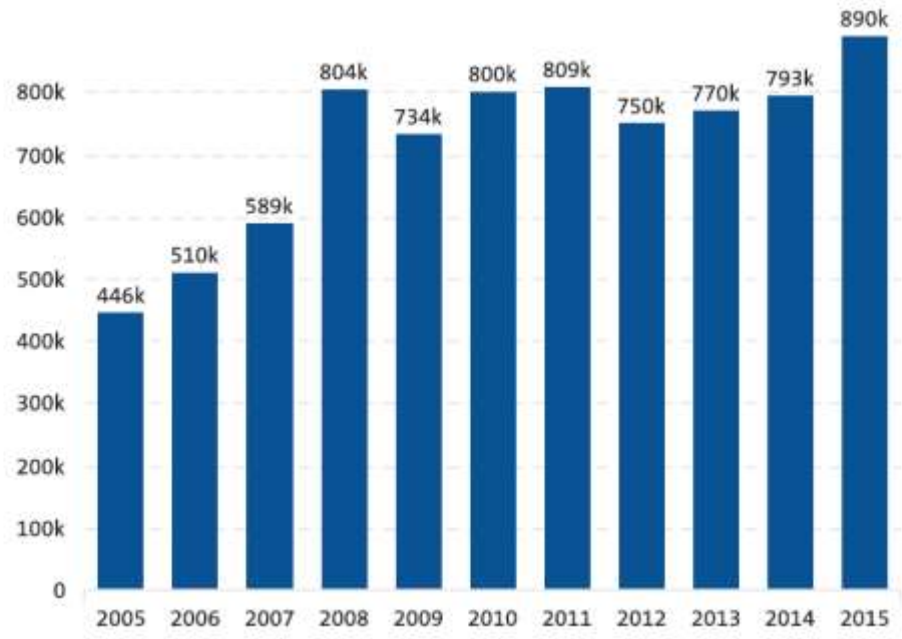


Figure 4 - Heat pump market in Europe; Sold devices per year

2.2 Heat pumps current fields of application in thermal networks

The Austrian technology roadmap for heat pumps [8] shows that current fields of application exist mainly in thermal networks in which the flow temperature, even at peak times, is not significantly above 100 °C. In Austria, the flow temperatures are often higher and are in a range up to 120 °C, in Vienna even in a range up to 150 °C, which makes the integration of heat pumps difficult. Flow temperatures of 120 °C are basically achievable with today's heat pump technologies. Even source temperatures of up to 55 °C can be used today. However, these high-temperature heat pumps are currently not widely available on the market.

The use of heat pumps in this temperature range is therefore mainly a business decision. With lower flow temperatures in domestic thermal grids, the potential for the feed-in of waste heat from the industry as well as environmental heat can be significantly increased. If the operating temperature of a thermal network is below the required temperature of the consumer (for example for hot water generation), a decentralized temperature increase has to be provided locally. For this purpose, so-called booster heat pumps are used which use low-temperature networks with flow temperatures below 60 °C as a source in order to provide temperatures of more than 60 °C as necessary. Internationally, the first devices of such heat pumps are currently finding their way onto the market.

A study by the DHC + technology platform (see [9]) confirms that, especially in the Scandinavian region (particularly in Sweden), large heat pumps are already heavily integrated in district heating systems and sometimes take over a significant part of the heat supply. Table 1 lists the results of the market analysis in 13 European countries, including the installed capacity and number of systems as well as HP units.

Table 1 - Large heat pumps in thermal networks [9]

Country	*Total Thermal Output (MW_{th})*	*Number of plants*	*Number of HP units*
Norway	84.5	8	15
Sweden	1,022.3	13	43
Denmark	45	9	11
Finland	154.6	4	9
Italy	36.6	5	9
Switzerland	35.4	9	13
Austria	10.1	2	3
Lithuania	15	1	1
Slovakia	1.8	1	1
Czech Republic	6.4	1	1
Poland	3.7	1	2
France	5.5	2	3
Netherlands	1.2	1	1

Looking at the heat sources of heat pumps, around 30% of the plants use various urban wastewater (sewage). Around 25% are accounted by industrial waste heat and flue gas condensation and another 25% of the facilities use sea, river or lake water. Geothermal sources or storage systems (such as seasonal storage) are also used as heat source but they represent just a small part, see Figure 5. The sink temperature or heat output temperature of the heat pump depends on the integration of the heat pump into the respective district heating network and thus varies widely. According to the above study, most systems vary the flow temperature between 61 and 90 ° C, see Table 3. Depending on the temperatures of the source and sink, the heat pump systems can be operated with different efficiency. [9] shows that the average number of COP in the plant is 3.74 and, depending on the system (e.g. use of high temperature sources or increase in the return temperature in the network), a COP between 5.4 and 6.5 can also be achieved.

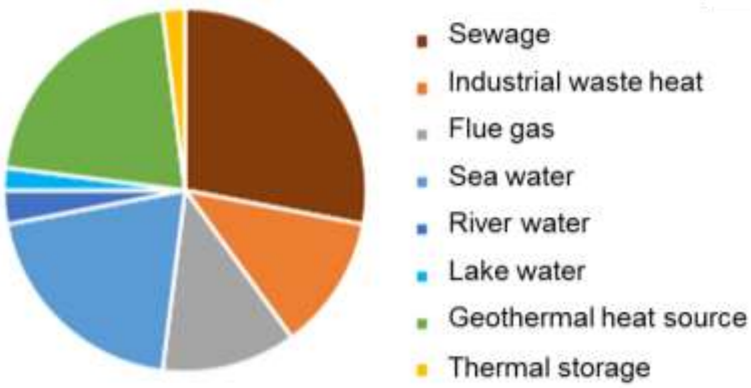


Figure 5 - Distribution of heat sources [9]

Table 2 - Temperature ranges of different heat sources [9]

Temperature range	2-9 °C	10-20 °C	11-40 °C	14-46 °C	10-40 °C	15-75 °C
Heat source	See, River, Lake water	Waste water	Flue gas	Industrial waste heat	Thermal storage	Geothermal heat source

Table 3 - Number of heat pumps per temperature range [9]

Supply temperature	41-50 °C	51-60 °C	61-70 °C	71-80 °C	81-90°C	91-100 °C
Number of HP units	1	2	22	36	35	1

2.3 Best Practice Examples

Below is a selection of relevant reference systems:

Table 4 - Selected heat pump systems in heating networks. Source: [10], supplemented and partially corrected

Location	Utilization	Number of HP	Total heating power HP	Refrigerant fluid	Inlet temp. Cond.	Outlet temp. Cond.	Heat source	Inlet temp. Evap.	Outlet temp. Evap.	Additional heat sources
Amstetten (Austria)	Heating	1	210 kW	R134a	30 °C	40 °C	Waste water	11 °C	8 °C	Nothing
Hallein (Austria)	Heating	1	7 MW	H_2O/LiBr	60 °C	90 °C	Flue gas condensation	60 °C	40 °C	Nothing
Bergheim (Austria)	Heating	2	9.5 MW	R134a	50 °C	60 °C	Flue gas condensation	50 °C	30 °C	Solar thermal energy, cooling energy of a biogas cogeneration plant
Lehen (Austria)	Heating	1	160 kW	R134a	45 °C	55 °C	Solar thermal energy	37 °C – summer; 20 °C - Winter	No data	Nothing
Flachau (Austria)	Heating	1	1.2 MW	R134a	57 °C	64 °C	Flue gas condensation	No data	25 °C – summer; 40 °C - winter	Nothing
Krumpendorf (Austria)	Heating	1	245 kW	R134a	48 °C	65 °C	Flue gas condensation	23 °C	No data	Solar thermal energy
Tamsweg (Austria)	Heating	1	8.8 MW	R236fa	55 °C	62 °C	Return flow	50 °C	40 °C	Nothing
Klagenfurt-East (Austria)	Heating	2	20 MW	H_2O/LiBr	60 °C	70 °C	Flue gas condensation	45 °C	35 °C	Nothing
Värtan Ropsten (Sweden)	Heating	6	180 MW	R134a	57 °C	80 °C		See water	2,5 °C 0,5 °C	Biofuel plants (base load), Oil boiler (peak load)
Helsinki (Finland)	Heating, Cooling	5	84 MW	R134a	50 °C	62 °C		Waste water	10 °C 4 °C	Gas CHP
Lund (Sweden)	Heating	2	47 MW	R134a	60 °C	81 °C		Geothermal	22 °C 4 °C	Biofuel CHP, Gas- and Oil boiler, electric boiler

Continuation Table 4

Location	Utilization	Number of HP	Total heating power HP	Refrigerant fluid	Inlet temp. Cond.	Outlet temp. Cond.	Heat source	Inlet temp. Evap.	Outlet temp. Evap.	Additional heat sources
Lund (Sweden)	Heating, Cooling	3	28 MW	R134a	60 °C	81 °C	Cooling network (and if necessary geothermal energy)	10 (22) °C	4 °C	Biofuel CHP, Gas- and Oil boiler, electric boiler
Dalian (China)	Heating, Cooling	3	25 MW	No data	55 °C	65 °C	Waste water and seawater	7,5 °C	2,5 °C	No data
Sandvika (Norway)	Heating, Cooling	2/1	21 MW	R134a	57 and 70 °C	78 °C	Waste water and/or Cooling network	9/8 °C	4/2 °C	3 Oil boilers (peak load)
Malmö (Sweden)	Heating	2	19 MW	R134a	50 °C	60-70 °C	Flue gas condensation	34,2 °C	24,3 °C	Waste CHP
Oslo (Norway)	Heating	1	18,4 MW	R134a	No data	90 °C	Waste water	9,6 °C	No data	Waste incineration, Biomass boiler
Rolfsbukta (Norway)	Heating, Cooling	2	16 MW	R1234ze	65 °C	80 °C	Cooling network (and if necessary geothermal energy)	4,9 °C	2,5 °C	HP-Facilities Fornebu Telenor and Lysaker, Oil boiler
Drammen (Norway)	Heating	3	13,5 MW	R717	60-65 °C	75 - 120 °C	See water	8-9 °C	4 °C	Biomass boiler (base load), Gas boiler (peak load)
Marstal, (Denmark)	Heating, seasonal Storage discharge	1	1,5 MW	R744	33-40 °C	75 °C	Seasonal Storage	33-40 °C	No data	Solar thermal, Biomass
Créteil (France)	Heating	1	8,7 MW	No data	67,2 °C	89 °C	Return from heat network	55 °C	37,5 °C	Geothermal energy
Le Plessis-Robinson (France)	Heating	2	6,8 MW	No data	45 °C	70 °C	Geothermal energy	38 °C	14 °C	Gas boiler
Fornebu Telenor (Norway)	Heating, Cooling	2	5,4 MW	R134a	50 °C	75 °C	See water	5 °C	2,5 °C	HP-facilities Rolfsbukta and Lysaker, Oil boiler
Boulonge-Billancourt (France)	Heating, Cooling	1	5 MW	No data	50 °C	80 °C	Cooling network (and if necessary geothermal energy)	10 (12) °C	5 °C	Gas boiler
Fresnes (France)	Heating	1	3,4 MW	No data	42 °C	61,6 °C	Return from heat network	42 °C	31,7 °C	Geothermal energy, No data

2.4. Description of DH Systems with different temperature levels

District heating companies, consultants and researchers have discussed definitions of district heating temperature levels intensively in recent years.

In [2], the following indicative temperature levels and definitions were specified:

- DH High Temperature System (HT), 100/50 °C (supply/return temperature)
- DH Low Temperature System (LT), 80/40 °C
- DH Very Low Temperature System (VLT), 60/30 °C
- DH Ultra Low Temperature System (ULT), 45/30 °C
- Thermal Grids (TG), 28/8 °C
- District cooling systems (DCS), 10/15 °C

The HT/LT definitions of this work were based on Euroheat & Power's guideline on district heating substations [39].

Thermal grids (TG) is defined as grids where energy is transported and exchanged between different consumers/prosumers. Some buildings are extracting heat from the system and other extracts cooling from the grid. The main problem is to balance the loads in the grids.

Other references [4] are using different generations (periods in time) of district heating to define the temperature level:

- 1st Steam systems, steam pipe in concrete ducts
- 2nd Pressurized hot water system, Heavy equipment, Large "Build on site" stations
- 3rd Generation District Heating (3GDH): Temperature level < 100 °C
- 4th Generation District Heating (4GDH): Temperature level < 50-60 °C (70) °C. The brackets indicate the temperature can go up to 70 °C in the winter.

This implies that both the VLT and the ULT definitions introduced above are contained in the definition of 4th generation district heating.

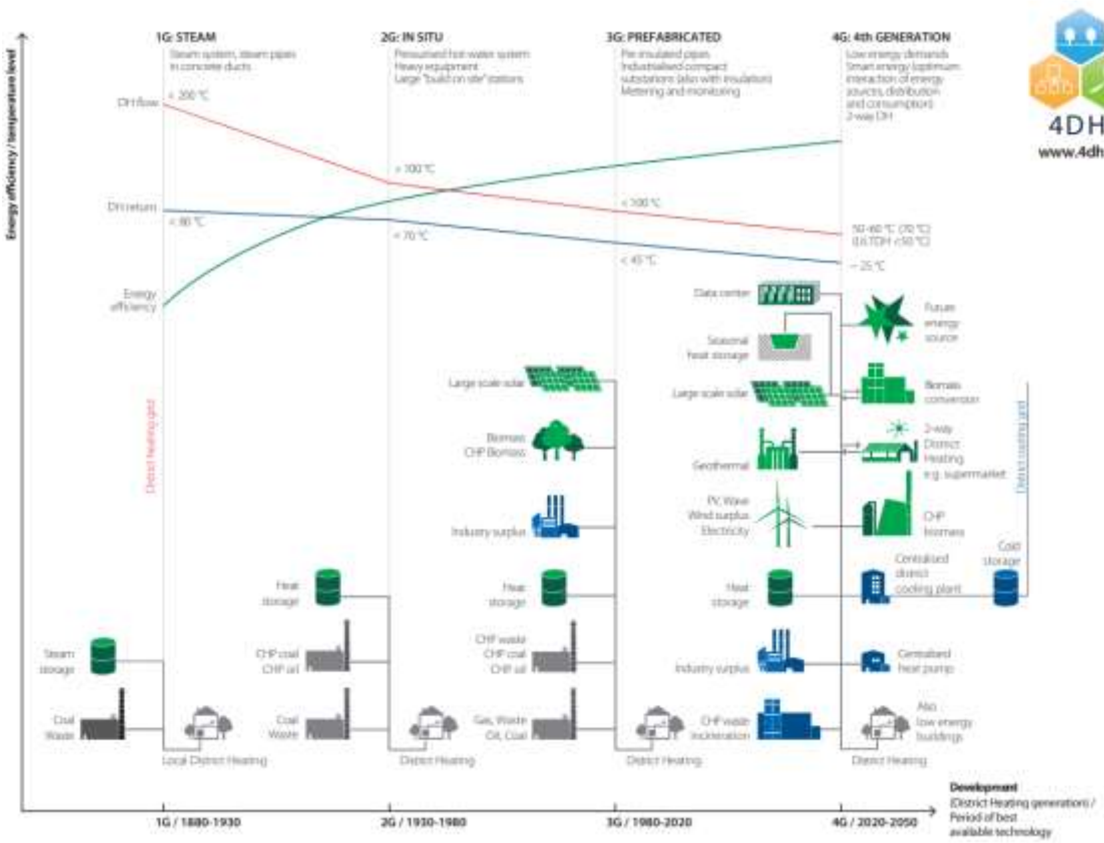


Figure 6 - District heating technology evolution (source 4DH research centre, www.4dh.eu)

2.4.1 Low Temperature District Heating

Low temperature district heating for future energy systems has advantages for both the demand side and the generation side. Especially in connection with buildings that require only low temperatures for space heating, which means the required indoor temperature in most building types residential and non-residential buildings generally low (below 23 °C). In combination with well-designed low temperature heating systems with supply temperatures between 35-40 °C is it possible to use VLT or ULT directly for heating. This enables the possibility to increase the integrate the implementation of renewable and waste energy sources like solar thermal collectors or heat pumps or excess heat. In ULT systems which has temperatures between 45 °C and 30 °C is it necessary to boost the temperature for Domestic Hot Water DHW with electrical heaters or with a booster heat pumps.

2.4.2 Grid losses

Thermal losses in the district heating grids is especially an issue in older grids which runs at temperature levels up to 100 °C (e.g. 3rd generation DH). In low temperature district heating is the network temperature reduced to about 50 °C or even less in 4th generation district heating. If the grid is well designed and insulated it means that the grid losses can be reduced by up to 75%.

Statistical data on the heat losses provided from the branch organizations on national level does not give a good indication how operation and temperature levels may contribute to the distribution losses. The grid losses is often a function of the linear heat density and consumption in the grid, which means that the losses usually are lover in cities where the population and building mass is more dense than in rural areas where the grid length is longer and the linear density is lower see figure 7. A general overview on the German national heat losses indicated that the heat losses have been in the range of 13 to 14% [5]. A challenge for the district heating grid is that the implementation of heat saving in the building stock reduces the heat demand density.

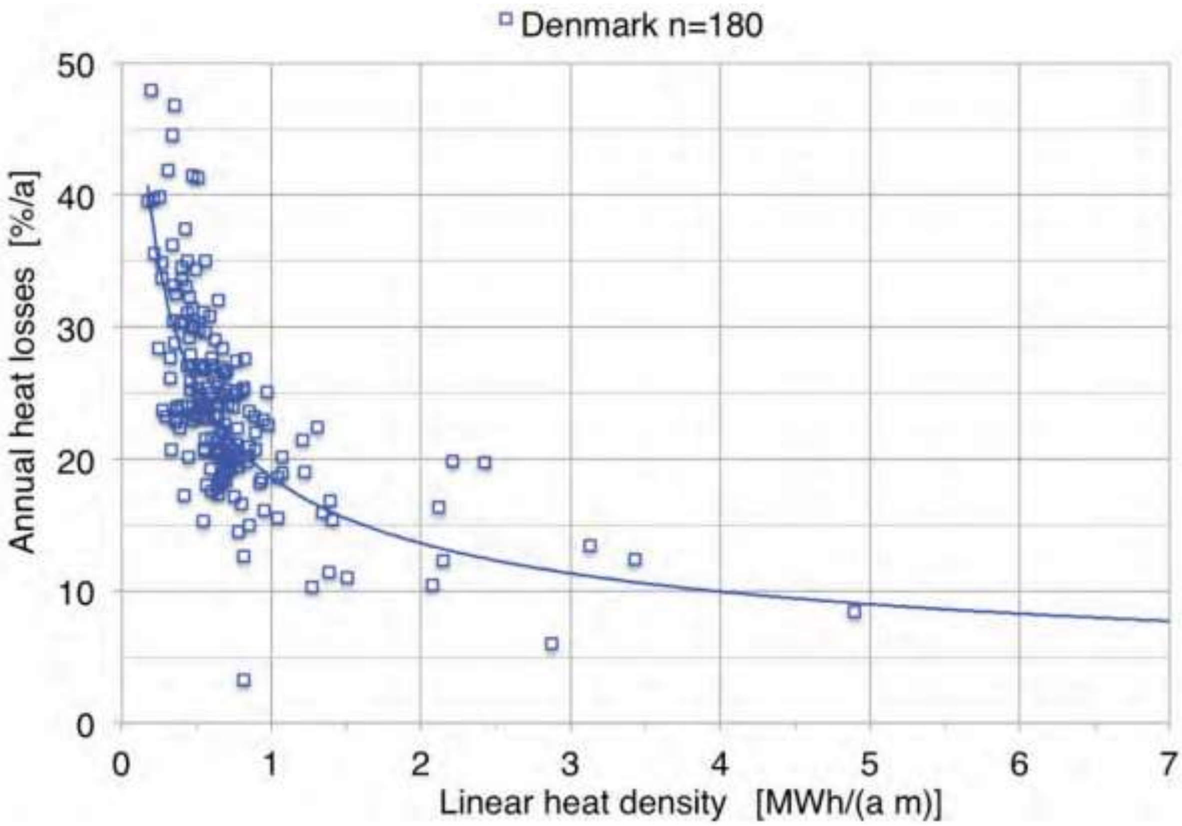


Figure 7 - Heat distribution losses as function of the linear heat density in Denmark for plants [41]

In the Danish city Middelfart is the district heating system delivering heat to 5,000 customers. The company has during a project reduced the supply and return temperature from an average of 80,6 / 47,6 °C in 2009 to an average of 64,6 / 40,0 °C in 2015, this means that the network heat losses has been lowered by almost 25% [5].

2.4.3 Domestic Hot Water production in Low Temperature District Heating

Low temperature district heating reduces the network forward temperature to the threshold value of ensuring hygienic DHW supply. One of the key issues is to supply DHW at greatly reduced temperature without the risk of Legionella bacteria. To ensure safe supply, many countries regulate the minimum DHW supply temperature and recirculation temperature. A well designed and functioning DHW system must fulfill the requirements for hygiene, thermal comfort and energy efficiency. In general, the bacteria risk is present when DHW temperatures are below 50 °C.

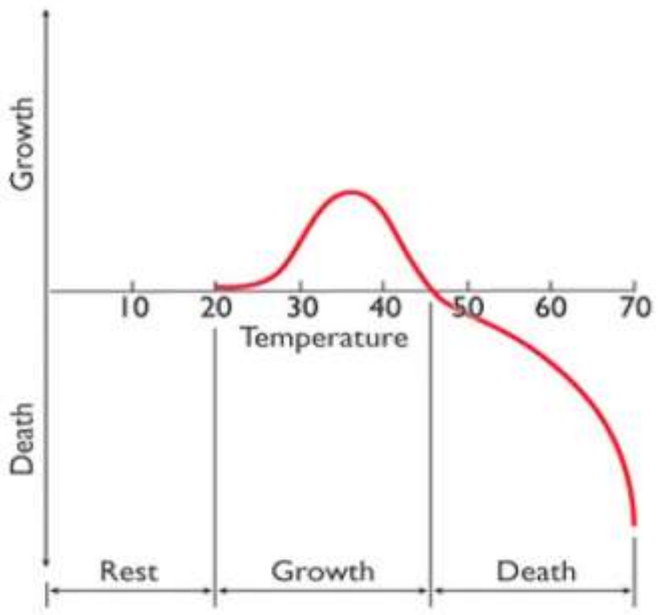


Figure 8 - Legionella bacteria growth/decay rates as function of temperature [41]

For all practical purposes is 50 °C high enough for human hygienic needs, e.g. dissolving food fats in dishwashing Legionella growth at 50 °C is confined. This temperature is also low enough to avoid scalding of human skin, which can occur at temperatures above 65 °C.

The problem of legionella in DHW systems needs to be addressed before the implementation of VLT or ULT DH. One solution could be to add local supplementary heating devices, so that the temperature of DHW can be boosted and the circulation temperature kept at 50 °C. Another method is to limit the total volume of DHW in use and heat DHW locally and instantaneously, and thereby reduce the risk growth as much as possible.

Table 5 - District heating temperature levels

	HT	LT	VLT	ULT	TG
Typical Temperature Supply/Return	100 °C/50°C	80 °C/40°C	60 °C/30°C	45 °C/30°C	28 °C/8°C
Domestic Hot water production type	Tank/ Instantaneous heat exchanger unit	Tank/ Instantaneous heat exchanger unit	Tank/ Instantaneous heat exchanger unit	Micro Booster Heat pump/electrical heater/Gas or oil	Micro Booster Heat
Heating system usable	Radiator/floor heating	Radiator/floor heating	Radiator/floor heating	Floor Heating/ Air coils	Floor Heating

2.4.4 Micro booster heat pumps

In large buildings or buildings with DHW circulation is it necessary to keep the circulation temperature at 50 °C, which requires a district heating supply temperature of more than 55 °C. This problem can be avoided by using a micro-booster heat pump to heat the circulation loop to 50 °C. The booster heat pump is relevant in combination with ULT District Heating and in VLT district heating grids. Studies shows that supply temperatures at 45-40 ° is sufficient for heating 80 % of the year and this means that the grid losses can be reduced with 25% on a yearly base.

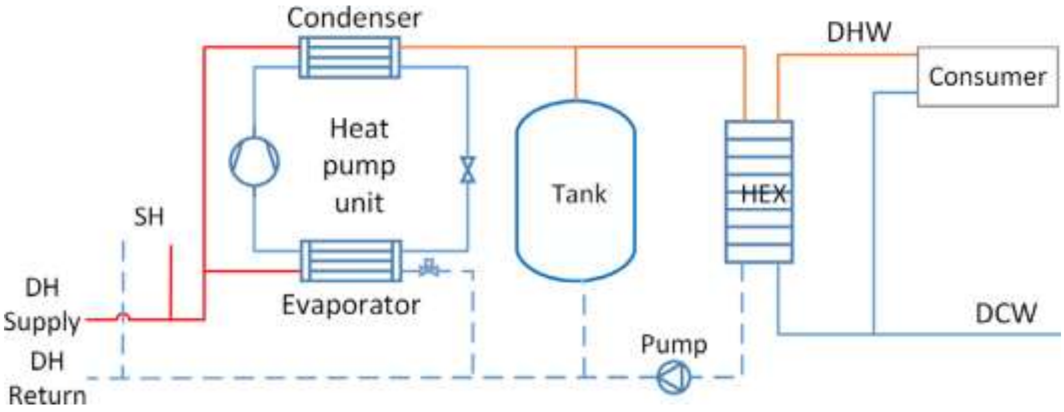


Figure 9 - DHW system installing a central heat exchanger combined with heat pump [42]

2.4.5 Electrical heater and micro tank

Instantaneously heat exchanger combined with a micro tank with an electrical heater is an option to boost the DHW tap water temperature and reduce the risk of legionella, by keeping the DHW volume to a minimum. Since the temperature of the preheated water is lower than the comfort requirement, one stream of the preheated DHW is further heated and stored in a micro tank. To meet the requirement of Legionella prevention, the DHW in the tank is heated to 60 °C by the electrical heater.

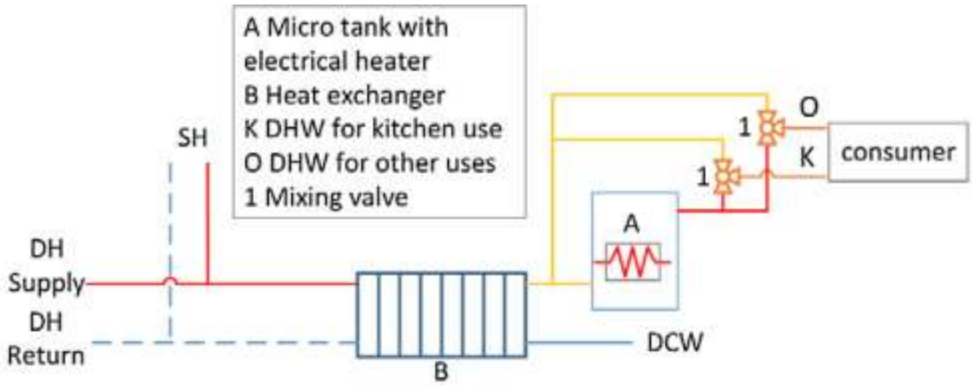


Figure 10 - DHW system with electric heated micro tank system [42]

2.4.6 Thermal Energy Grids

Thermal grids (TG) is defined as grids where energy is transported and exchanged between different consumers/prosumers. This grid type is used different names like Anergy networks or the brand name Ectogrid. The concept is that some buildings is extracting heat from the system and other extracts cooling from the grid, the temperature in the grid is usually between 28 °C and 8 °C depending on the load and usually close to the temperature of the surrounding soil. This means that the grid is working both as a heating and cooling grid. The main problem is to balance the loads in the grids. The grid is usually installed without insulation or a low degree of insulation, this means that the installation costs can be comparable to horizontal to ground source collectors.

2.5 Technical solutions for the integration of heat pumps

As previously mentioned, heat pumps are used to utilize the heat stored in low temperature sources. Usually, heat from the environment (soil, water, air), industrial processes, wastewater, or other low-temperature sources is used and raised to a level required for heating purposes using electrical energy [11]. Depending on whether there are additional generators in the system, there are three types of operation, which will be briefly discussed below.

2.5.1 Operation of heat pumps

Depending on the environment, heat pumps are used differently. A distinction is made between monovalent, mono-energetic and bivalent (parallel and alternative).

Monovalent operation
In this mode, the heat pump has to provide the entire heat supply even on the coldest days. The heat source must be able to supply the required amount of heat to the evaporator. If the temperature of the heat source is influenced by the ambient temperature the maximum heating capacity has to be provided at low source and high sink temperatures which means at low efficiency. In monovalent operation, therefore, on the one hand the source should be able to supply sufficient energy, on the other hand the building standard should be correspondingly high to limit the required flow temperatures.

Mono-energetic operation
In mono-energetic operation, the heat pump provides most of the heating load. At particularly low outside temperatures, an additional electric heater is turned on, thus helping the heat pump to reach the required temperatures. As a result, only one form of energy (power) is used, from which the name derives.

Bivalent operation
If this mode of operation is selected an additional heat generator is required, in addition to the heat pump. The heat pump either completely takes over the heating (bivalve alternative - at low outside temperatures) or is operated in addition to the heat pump (bivalent parallel). This corresponds to the most frequently used operation of heat pumps in thermal networks, since the heat pump is used for base load coverage, while being operated constantly and with higher efficiency and the peak load is provided by other heat generators. In addition, the investment costs of the heat pump can be reduced because it must not cover the entire heat load.

2.5.2 General hydraulic feed-in options

For the decentralized heat supply into a network, there are in principle three different options shown in *Figure* 11*11*, depending on the circuit arise for the district heating operator and the supply advantages and disadvantages. These are briefly described below.

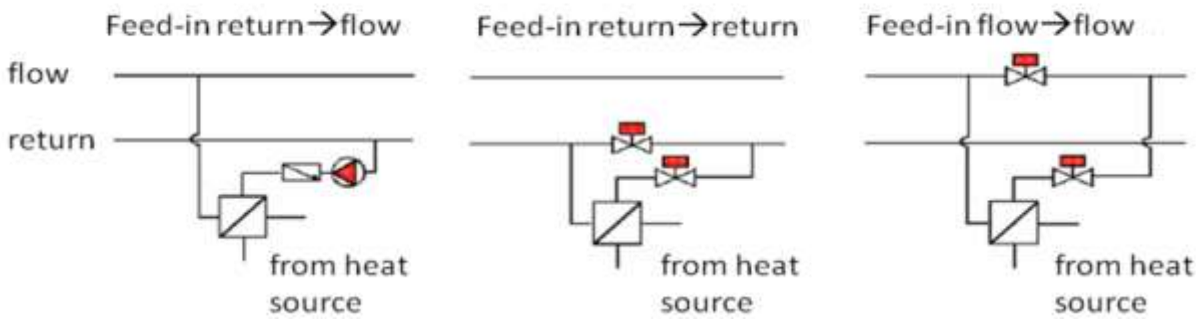


Figure 11 - Hydraulic circuits for heat feeders in existing networks [13]

Extraction from the district heating return and supply into the district heating process
This option corresponds to the classic integration of a producer into a district heating network. The respective return temperature and the required flow temperature in the network give the supply temperature difference. The supplied power or the volume flow is variable and is regulated according to the required flow temperature. To overcome the pressure difference between the supply and return pipe, a pump is necessary. The required pumping energy depends on the pressure difference between the district heating return and the district heating flow or the extracted volume flow. This type of feed-in is most preferred by district heating operators, as the return temperature does not increase, and a part of the pumping costs is assumed by the heat source owner if it is an external company [13].

Extraction and supply to the district heating return line
Through this circuit, the return temperature is raised by the decentralized supplier. As a result, the heat is supplied at a lower temperature, which is particularly advantageous for the efficiency of heat pumps. In Figure 1111 the pressure difference of the heat exchanger and the connecting lines is overcome by the network pumps and the volume flow is regulated by two straight way valves. Alternatively, a three-way valve or an additional pump can be used. An additional pump has higher investment costs but is more efficient in operation due to lower pressure losses in the system. A disadvantage of this circuit for the district heating network operator is the higher return temperature, since the efficiencies of existing conventional heat generators may decrease due to the higher return temperature. In addition, the heat losses in the return line are increased by the higher temperature [13].

Extraction and supply to the district heating supply line
To increase the flow temperature the decentralized supply has to provide the highest required temperatures which leads to a lower efficiency compared to the two previously described circuits. The pressure difference is applied by the network pumps as in section 3.2 and covers the pressure loss of the heat exchanger and the connecting lines. In Figure 1111 the volume flow is regulated by two straight way valves but, as in section 3.2, also a three-way valve or an additional pump can be used. The heat losses up to the decentralized heat supplier are lower, since the maximum temperature is reached only after the supplier [13].

2.5.3 Hydraulic integration options for heat pumps in DH networks

As the supply options described above show, heat generators, such as e.g. Heat Pumps can be integrated in different ways depending on needs and possibilities in thermal networks. In principle, a distinction can be made between decentralized and centralized integration. If there are only one or only a few main feed-in points with a large capacity of the existing facilities it is usually a case of central integration. If a separate location for smaller generation technologies is realized or producers are distributed over many locations in the network, it is possible to talk about decentralized integration. For hydraulic integration, it usually plays a subordinate role whether a heat pump is regarded as a central or decentralized feed-in.

In the case of the central option (for example, to an already existing power plant location), in many cases the required infrastructure, such as electricity connections, pipelines, pumps, etc., is already available or only needs to be adjusted. Furthermore, it can be differentiated when integrating heat pumps, whether external or internal heat sources are used. External sources refer to heat sources that bring energy into the system from outside. These include e.g. supply of environmental heat, use of industrial waste heat or waste heat from infrastructure (sewers, tunnel systems, ...).

In contrast, internal sources do not bring additional renewable energy directly into the system. Here, the district heating network is used as a heat source, e.g. to carry out a temperature increase at the consumer (for example for the preparation of hot water) or to further cool the return flow and thus to increase the transport capacity or also for a deeper discharge of a storage. Due to the greater degree of cooling, other renewable energy sources such as solar thermal energy, waste heat, etc. can be integrated into the system easier or increasingly. The next figures show different possible integration concepts.

Figure 12 shows the integration of heat pumps with external heat source in the DH supply. Through this option, additional energy is brought into the DH system from outside. By integrating into the flow, the HP has to provide the highest temperatures, which reduces efficiency. In addition, special refrigerants have to be used with very high condensation temperatures. The advantage of this option is that it does not affect existing generation plants.

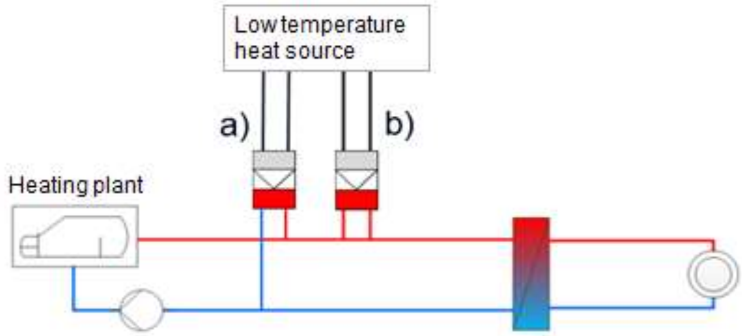


Figure 12 - Integration of heat pumps with external heat source in the DH supply line; a) parallel b) serial [10]

Figure 1313 shows the integration of heat pumps with external heat source into the DH return pipe. As a result, energy can be introduced from the outside as described above. The lower return temperature has a positive effect on the efficiency of the system. When connecting to the return line, care must be taken to ensure that existing heating plants work with higher return temperatures.

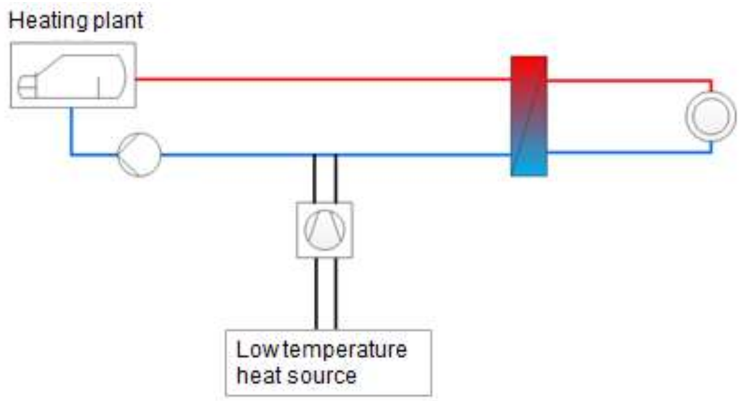


Figure 13 - Integration of heat pumps with external heat source in the DH return [10]

The concept for integrating heat pumps with internal heat source into the DH flow is shown in Figure 1414. The heat pump uses the DH return as a heat source and supplies heat to the flow. As a result, no additional renewable energy is brought into the system directly (except when using renewable electricity to drive the HP). The cooled return flow, however, makes it possible to integrate additional energy carriers, such as solar thermal energy or waste heat from flue gas condensation plants and thus additional renewable energy is brought into the system.

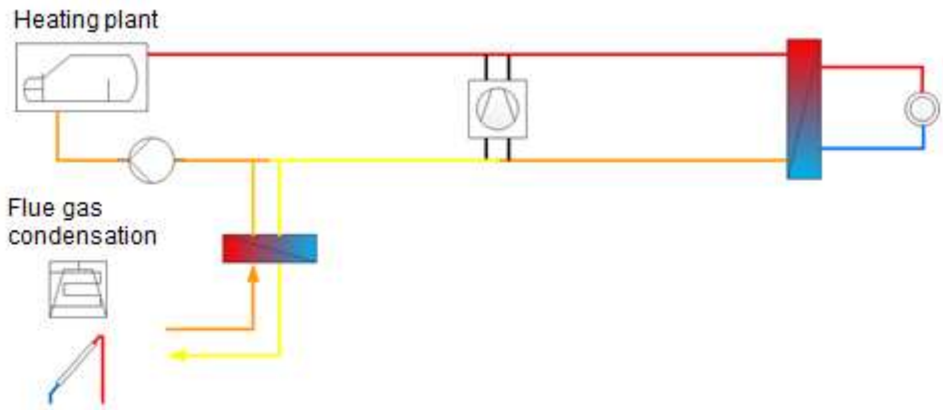


Figure 14 - Integration of heat pumps with internal heat source in the DH supply line; additional possibility for (renewable) energy sources due to reduced temperatures, e.g. here flue gas condensation or solar thermal energy [10]

Figure 1515 shows the integration of heat pumps with internal heat source in the DH return. As mentioned before, the possibility for the use of additional energy sources is created. At the same time, the HP does not have to provide heat at the maximum network temperatures of the DH flow and can therefore operate more efficiently. Existing heating plants can thus be used e.g. for peak load coverage.

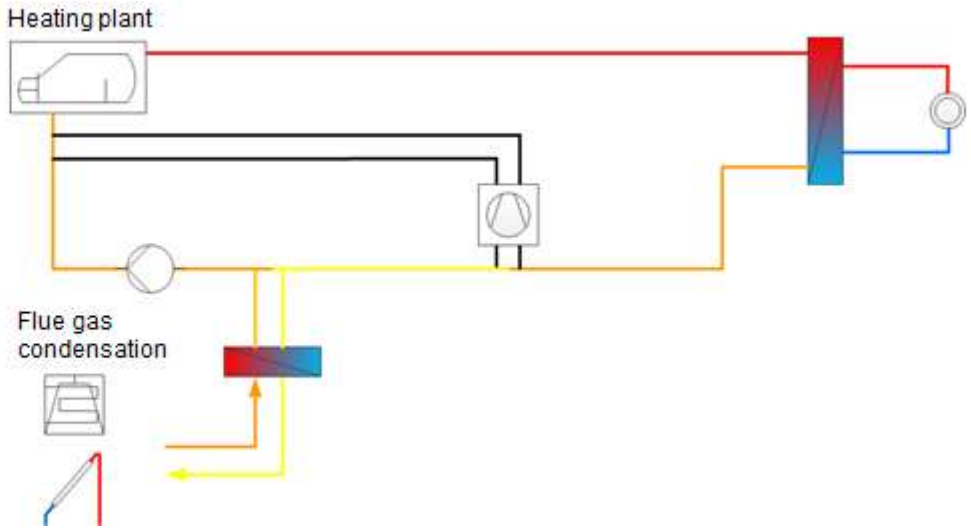


Figure 15 - Integration of heat pumps with internal heat source in the DH return; additional possibility for (renewable) energy sources due to reduced temperatures, e.g. here flue gas condensation or solar thermal energy [10]

Another option of the use of internal heat sources is the integration of heat pumps for the discharge of seasonal, stratified thermal storages. For this, the lower layers of the storage tanks are used as a heat source and the upper layers are used as a heat sink this a higher temperature difference within the storage is achieved. This increases the capacity of the storage and creates potential for additional energy sources. An example for the integration of a seasonal thermal storage, a solar thermal system and a heat pump into a district heating network is shown in Figure 16. The heat pump cools the bottom of the thermal storage to temperatures below the return temperature of the district heating network. The lower temperature at the bottom leads to lower return temperatures for the solar thermal system and thus to a higher solar yield.

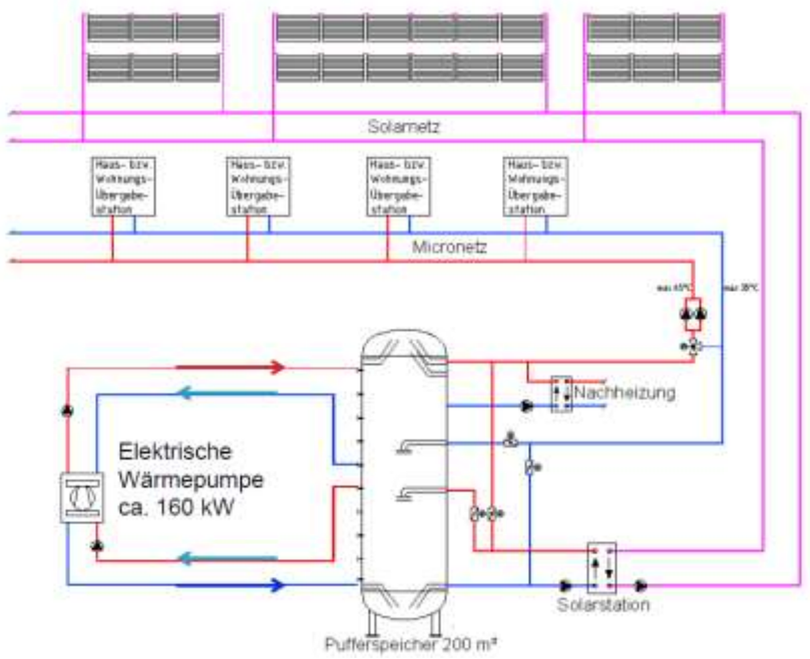


Figure 16 - Integration of a seasonal stratified thermal storage, a solar thermal system and a heat pump into a district heating network [40]

To increase transport capacities, heat pumps can be integrated as shown in Figure 17. The graphic shows two options for increasing the temperature in certain network sections, here for secondary networks. Thereby, e.g. remote areas are supplied with DH, without having to increase the temperature throughout the network. This decentralized temperature increase allows additional consumers to be connected to the grid. Possible network bottlenecks can thus be bypassed or prevented. Figure 18 shows a similar concept for the integration of heat pumps for the supply of secondary networks. Here, the supply of the secondary network is entirely covered by the heat pump, which uses the DH return as heat source.

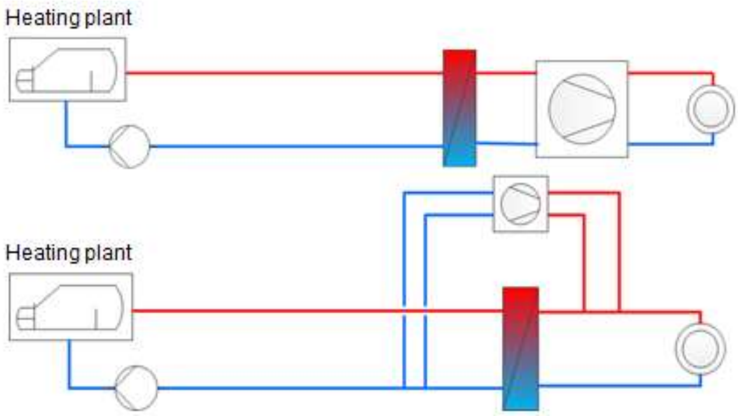


Figure 17 - Integration of heat pumps with internal heat source to increase transport capacity; e.g. Temperature increase in secondary networks [10]

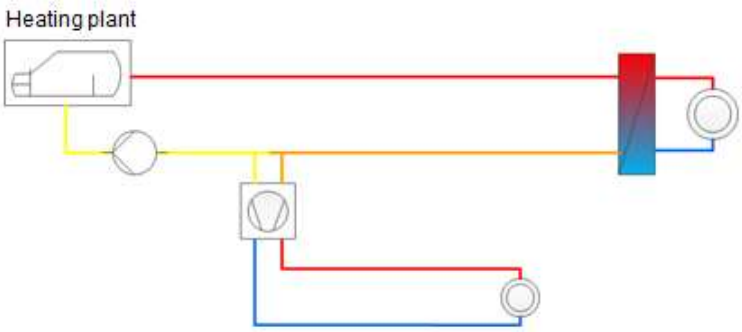


Figure 18 - Integration of heat pumps with internal heat source to increase transport capacity; e.g. Supply of secondary networks [10]

2.5.4 Commercially available large heat pumps

The use of heat pumps is mainly limited by the usability of the selected compressor in combination with the refrigerant used. The limits of use are particularly dependent on the type of compressor, the type of refrigerant circuit (single-stage, multi-stage, intermediate injection, etc.) and the refrigerant. These can be taken from manufacturer documentation. Figure 19 lists various refrigerants and assesses their properties. The possible critical temperatures and the maximum evaporation and condensation temperatures were evaluated as a function of the maximum pressure. Depending on the requirement of the heating network or depending on the available heat source different refrigerants can thus be used. According to the study [9], currently the refrigerant R134a is mainly used for the application of large heat pumps in thermal networks. Further properties of relevant refrigerants can be found in (Figure 21).

Reine Fluide	T_{krit} > 95 °C	T_{krit} > 125 °C	p_s < 7 bar p_s ($t_{v,max}$ = 45 °C)	p_s < 25 bar p_s ($t_{Ko,max}$ = 90 °C)	p_s < 25 bar p_s ($t_{Ko,max}$ = 120 °C)
R11	1	1	1	1	1
R113	1	1	1	1	1
R114	1	1	1	1	1
R123	1	1	1	1	1
R141b	1	1	1	1	1
R142b	1	1	1	1	1
R236ea	1	1	1	1	1
R245ca	1	1	1	1	1
R245fa	1	1	1	1	1
R600	1	1	1	1	1
R600a	1	1	1	1	1
R718	1	1	1	1	1
R365mfc	1	1	1	1	1
R717	1	1	0	0	0
R124	1	0	1	1	1
R236fa	1	0	1	1	1
RC318	1	0	1	1	1
R12	1	0	0	0	0
R22	1	0	0	0	0
R134a	1	0	0	0	0
R152a	1	0	0	0	0
R227ea	1	0	0	1	1
R290	1	0	0	0	0
Gemische					
R401a	1	0	0	0	0
R401b	1	0	0	0	0
R401c	1	0	0	0	0
R405a	1	0	0	0	0
R406a	1	0	0	0	0
R409a	1	0	0	0	0
R409b	1	0	0	0	0
R411a	1	0	0	0	0
R411b	1	0	0	0	0
R414b	1	0	0	0	0
R500	1	0	0	0	0
R501	1	0	0	0	0

HFCKW
FCKW
HFKW
FKW
natürlich

Figure 19 - Evaluation of refrigerant for district heating applications 1 = suitable, 0 = unsuitable, [14]

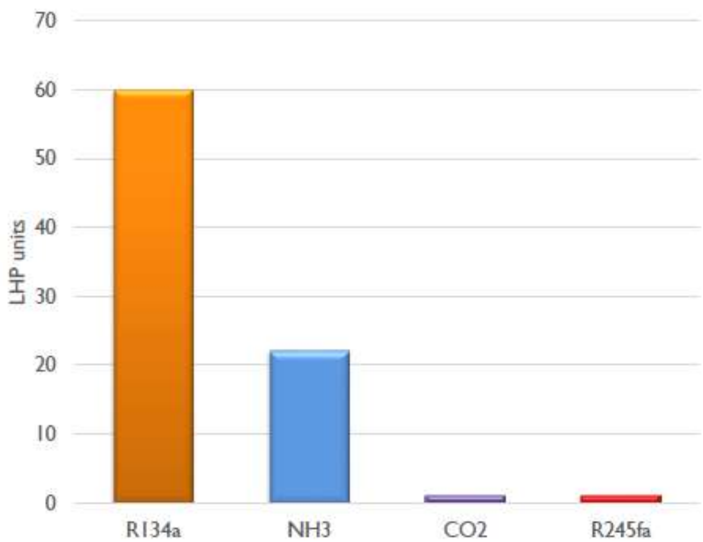


Figure 20 - Number of heat pumps according to used refrigerant [9]

				kritische Daten			Daten Normalsiedepunkt											
Kältemittel	chem. Formel	Mol-masse	Erstar-rungs-tempera-tur	kritischer Druck	kritische Tempera-tur	kritiche Dichte	Siede-tempe-ratur	Wärme-kapazität Flüssigkeit	spez. Volumen Flüssigkeit	spez. Volumen Dampf	Verdamp-fungs-enthalpie	Isen-tropen-exponent	TLV-TWA	Sicher-heits-gruppe	Atmos-phärische Lebens-dauer	POCP	ODP	GWP_{100}
		[kg/mol]	[°C]	[Mpa]	[°C]	[kg/m³]	[°C]	[kJ/(kg*K)]	[m³/kg]	[m³/kg]	[kJ/kg]	[-]	[ppm]	[-]	[Jahre]	[-]	[-]	[-]
R-744	CO_2	44,0	-56,6	7,377	30,98	467,6	-78,4					1,308	5.000	A1	> 50		0	1
R-1270	C_3H_6	42,1	-185,2	4,555	91,06	230,1	-47,6	2,193	0,00164	0,42407	439,0	1,172	660	A3	0,001		0	ca. 20
R-290	C_3H_8	44,1	-187,6	4,251	96,74	220,5	-42,1	1,440	0,00172	0,41382	425,6	1,150	2.500	A3	0,410	200	0	ca. 20
R-1234yf	$CF_3CF{=}CH_2$	114,0	-53,1	3,382	97,40	478,0	-29,5							A2L			0	4
R-134a	CH_2FCF_3	102,0	-103,3	4,059	101,06	511,9	-26,1	1,281	0,00073	0,19011	217,0	1,132	1.000	A1	14,000	0,5	0	1.430
R-227ea	CF_3CHFCF_3	170,0	-126,8	2,925	101,75	594,3	-16,3	1,079	0,00065	0,11784	131,8	1,084	1.000	A1	34,200		0	3.220
HFO-1234ze-E	$CF_3CH{=}CHF$	114,0		3,636	109,40		-19,0							A2L		6	0	
R-152a	CH_3CHF_2	66,1	-118,6	4,517	113,26	368,0	-24,0	1,625	0,00099	0,29620	329,9	1,172	1.000	A2	1,400	4,5	0	124
R-236fa	$CF_3CH_2CF_3$	152,0	-93,6	3,200	124,92	551,3	-1,4	1,205	0,00069	0,13997	160,3	1,094	1.000	A1	240,000		0	9.810
E 170	$CH_3\text{-}O\text{-}CH_3$	46,1	-141,5	5,341	127,15	277,0	-24,8	2,229	0,00122	0,42334	461,6	1,174	1.000	A3	0,015		0	1
R-717	NH_3	17,0	-77,7	11,333	132,25	225,0	-33,3	4,448	0,00147	1,12440	1369,5	1,327	25	B2L	0,010		0	< 1
R-600a	C_4H_{10}	58,1	-159,4	3,629	134,66	225,5	-11,8	1,547	0,00168	0,35379	365,1	1,118	800	A3	0,019		0	20
R-600	C_4H_{10}	58,1	-138,3	3,796	151,98	228,0	-0,5	1,641	0,00166	0,36910	386,0	1,119	800	A3	0,018	300	0	ca. 20
R-245fa	$CHF_2CH_2CF_3$	134,1	-102,1	3,651	154,01	516,1	15,1	1,302	0,00073	0,16773	196,0	1,419	300	B1	7,600		0	1.030
HFO-1336mzz-Z	$CF_3CH{=}CHCF$	164,0		2,903	171,30		33,4							A1	0,066		0	9
R-245ca	$CH_2FCFCHF_2$	134,1	-273,2	3,925	174,42	523,6	25,1	1,332	0,00072	0,17432	201,0	1,314			6,200		0	693
R-365mfc	$CF_3CH_2CF_2CH$	148,1		3,250	186,90		41,3				186,1	1,058					0	825

Quellen:

Quellen:	CAS-Nummer:	/EPA 2010/
	Molmasse:	/Lemmon et al. 2002/, /Lemmon et al. 2007/, /UNEP 2011/
	Erstarrungspunkt:	/Lemmon et al. 2002/, /Lemmon et al. 2007/
	kritische Daten:	/Lemmon et al. 2002/, /Lemmon et al. 2007/
	Daten Normalsiedepunkt:	/Lemmon et al. 2002/, /ASHRAE 2010/
	Isentropenexponent:	/Lemmon et al. 2002/
	TVL-TWA:	/ASHRAE 2010/, /UNEP 2011/
	Sicherheitsgruppe:	/ASHRAE 2010/
	atmosphärische Lebensdauer:	/IPCC 2007/, /UNEP 2011/
	POPC:	/Buchenwald et al. 2009/
	ODP:	/Calm, Hourahan 2007/, /UNEP 2011/
	GWP_{100}:	/IPCC 2007/, /UNEP 2011/

Figure 21 - Properties of relevant refrigerants [15]

Apart from the refrigerant, the characteristics of the compressor are also of great importance for the use of heat pumps under conditions prevailing in district heating networks. It is important to distinguish between different compressor types. Depending on the delivery volume flow, possible pressure ratio and controllability, different types may be of interest for the application. Table 1 shows the most common and widely used types and their characteristics.

Table 6 - Characteristics of different compressor types [15]

Compressor design	Piston	Scroll	Screw	Turbo
Driving principle	displacement	displacement	displacement	turbomachine
Compression	static	static	static	dynamic
Swept volume	geometric	geometric	geometric	depending on backpressure
Production	pulsing	constant	constant	constant
Volume flow	up to 1,000 m³/h	up to 500 m³/h	100 - 10,000 m³/h	100 - 50,000 m³/h
Heating capacity	up to 800 kW	up to 400 kW	80 – 8,000 kW	80 – 40,000 kW
Typical pressure ratio (single-stage)	up to 10	up to 10	up to 30	up to 5
Controllability at constant speed	Stages	Difficult	continuously variable	continuously variable
Speed control	possible	possible	possible	possible
Sensitivity to liquid slugging	High	low	low	low
Vibration causes	Yes	no	no	no

For heat pumps in district heating networks, some manufacturers have developed products that, on the one hand, can cover a wide range of performances and, in addition, meet the specific requirements placed on the equipment. The following table shows a selection of different manufacturers and their heat pumps, which are suitable for district heating purposes. In addition to the product name, the refrigerant used is specified as well as the maximum possible flow temperature in °C. The table is not replicating the entire market.

Table 7 – Excerpt of some available heat pumps for district heating applications[1] [16]

Refrigerant	*max. supply temperature [°C]*	*Manufacturer*	*Product*	*Nominal power range [kW]*
R717	65	Cofely Refrigeration	L7 GH [..] PP Screw	111-1102
R717	65	Cofely Refrigeration	L7 GH [..] PP Piston	50-750
R134a	65	Cofely Refrigeration	L1 KH	150-700
R134a	65	Cofely Refrigeration	Spectrum	160-570
R134a	80	Combitherm	HWW	50-5000
R245fa	100	Combitherm	custom-made product	20-300
diverse	90	Friotherm	Unitop	20-30000
R717	82	GEA Refrigeration	Screw-WP 52 bar	500-10000
R717	90	GEA Refrigeration	Screw-WP 63 bar	2000-4500
R717	80	GEA Refrigeration	Piston-WP 50 bar	230-450
R717	115	Hybrid Energy	Hybrid Heat Pump	250-2500
R410a	52	Johnson Controls	YCWL/YCRL	188-580
R134a	70	Johnson Controls	YLCS	400-2000
R134a	65	Johnson Controls	YVWA	650-1250

[1] no guarantee for completeness

R245fa	105	Johnson Controls	YMC²	700-1800
R134a	80	Johnson Controls	SHP	700-3000
R134a	70	Johnson Controls	YK	1000-9000
R134a	70	Johnson Controls	CYK	2500-7000
R134a	90	Johnson Controls	OM	5000-20000
diverse	70	KKT Chillers	ThermoDynamixX	200-1000
diverse	80	Klima Jentzsch	custom-made product	100-2000
R410a	60	KWT/Viessman	Vitocal 350-G/W Pro	89-290
R134a	73	KWT/Viessman	Vitocal 350-G Pro	27-198
R134a	65	KWT/Viessman	Vitocal 350-G Pro Screw	223-1128
diverse	65	KWT/Viessman	custom-made product	15-2000
R717	85	Mayekawa	PlusHEAT	430-487
R717	85	Mayekawa	PlusHeat X	465-523
R134a	65	Ochsner	ISWS [..] ER2	81-493
R134a	65	Ochsner	IWWS [..] ER2	111-464
R407c	50	Ochsner	IWWS [..] ER1	165-966
R407c	50	Ochsner	ISWS [..] ER1	123-723
R134a	65	Ochsner	ISWS [..] R2	72-434
R134a	65	Ochsner	IWWS [..] R2	105-599
Öko 1	98	Ochsner	IWHS [..] ER3	65-634
R134a	98	Ochsner	IWHSS [..]R2R3	176-723
R134a	50	Ochsner	IWT 400 ER2 (Turbo)	360
R407c	50	Ochsner	ILWS 170 ER1	165
R410a	65	Oilon Scancool	CillHeat Re [..]	110-420
R134a	80	Oilon Scancool	CillHeat P [..]	150-380
R134a	67	Oilon Scancool	CillHeat S [..]	180-540
R717	90	Star Refrigeration	Neatpump	350-15000
R744	65	Star Refrigeration	Neatpump	45-200
diverse	65	Star Refrigeration	Neatpump	100-500
R744	90	thermea	thermeco2	45-1000

2.6 General Design and Integration Guidelines

The design of heat pumps essentially comprises three areas:

- Design of the heat source system
- Design of the heat pump
- Design of the heat sink system

The design and dimensioning of a respective system or device has to consider the other areas. Basically, the question whether heat output should be provided by the heat pump or delivered into the heat distribution system is at the beginning of the dimensioning. For smaller sizes the heat source system is dimensioned based on parameters such as the collector area, depth of borehole, water extraction amount, evaporator size for air and direct evaporator HP (detailed planning and analysis of the source is also a prerequisite). For larger evaporator capacities as in thermal networks the source has to be analyzed more precisely during the dimensioning phase. It has to be determined how much energy can be extracted from the respective source system and provided to the heat pump. This results in an iterative approach in which the heat source and the heat pump are matched to each other and to the heat delivery system (district heating network).

In addition to the three areas mentioned above, it is also necessary to specify how the heat pump should be operated or what proportion of the heat production the heat pump should cover.

2.6.1 Design of the heat source system

If natural, external heat sources such as water (sea, ground or river water) or soil (deep boreholes, geothermal) are used, the possible amount of heat extraction has to be examined to hold temperature limits. It is also important to pay attention to seasonal fluctuations. If non-natural external sources, as waste heat from industrial processes or wastewater heat are used, the respective temperature level and the possible extraction power have to be clarified and the system (for example heat exchangers, pumps, pipelines) dimensioned accordingly.

Interpretation of water-based sources
In order to determine the required water mass flow for the design of water-based sources, following equation can be used:

$$q_m = \frac{P_0 * 3600}{c_p * \Delta T} \qquad \textbf{Equation 1}$$

q_m Water mass flow [kg / h]

P_0 Cooling capacity of the heat pump [kW]

c_p Specific heat capacity of water (cp fresh water = 4.187 kJ / kgK; cp seawater depends on salinity)

ΔT Differential temperature (cooling of the water) [K]

If, as in some best practice examples (Section 2.4), seawater is used, the salt content has to be considered in the calculation of the specific heat capacity. Due to the lower freezing point compared to fresh water, the heat source can be cooled to lower temperatures. However, the national regulated limit values must be considered. Furthermore, special materials must be used to prevent or reduce corrosion.

Design of geothermal sources
When dimensioning depth probes and probe fields, the thermal conductivity, the temperature limits of the substrate as well as the cooling capacity and the annual operating hours of the heat pump have to be considered. The heat source should be designed that no undue cooling or heating of the substrate takes place. If the maximum permissible annual withdrawal is exceeded, damage to the pipes can be caused by frequent frost/thaw changes, which lead to a reduction in the efficiency of the system or damage to the subsoil. Alternating operation between heating in winter and cooling in summer is advantageous because it promotes better thermal regeneration of the substrate. Also, important for the dimensioning is the energy-efficient design of the circulation pump. If this is not chosen appropriately, the annual work rate can worsen blatantly.

Table 8 shows typical values for the specific extraction rate per meter of a deep hole for different soil conditions according to VDI 4640 [36].

Table 8 - Specific heat extraction capacity from the soil (extract from VDI 4640)

Soil quality	*Specific withdrawal rate per meter of depth hole*
Dry sediments	10 – 30 W/m
Shale	20 – 55 W/m
Hard rock with high thermal conductivity	40 – 80 W/m
Underground with high groundwater flow	50 – 100 W/m

For deep geothermal energy (e.g.> 400 m) other values may apply depending on the geological nature or thermal sources may also be used in the subsurface. In these systems, separate source testing and analysis (e.g., through trial drilling) is essential.

Design of non-natural sources (waste heat from industry, waste water and flue gas condensation)
When using industrial waste heat in most cases, the evaporator has to be designed to reach the capacity of the source. In some cases, the heat pump is designed to a certain condenser capacity which requires only a part of the available source capacity and the surplus heat of the source has to be dissipated to the environment by the use of cooling towers. Cooling towers are often installed parallel to the heat pumps to guarantee that the heat of the industrial process can be removed if the heat pump is out of order. The design of the heat exchanger takes place basically according to following equation or by means of software that is provided by various manufacturers.

$$A_{WÜ} = \frac{P_{WÜ}}{k * \Delta T} \quad \text{Equation 2}$$

$A_{WÜ}$ Area of heat exchanger [m²]

$P_{WÜ}$ Heat exchanger output or extraction power [W]

k Heat transmission coefficient [W / (m² * K)]

ΔT Temperature difference between heat transfer media [K]

When using wastewater heat, e.g. among other things, the sewage mass flow and the temperature must be known from the urban sewer system. By transforming Equation 1, the maximum extraction power can then be determined, and the heat pump can then be dimensioned. In any case, the specific extraction rate must also be adapted to the composition of the wastewater.

Waste heat utilization in the sewer system is possible in several ways. Above all, the type of heat exchange from the wastewater is distinguished. Conventional plate or shell and tube heat exchanger cannot be used here due to the large pollution of the wastewater (including coarse materials). Therefore, some systems have been developed in the past that meet the special needs of the sewer system.

- Sewage heat exchangers are made of high quality steel alloys and have particularly smooth surfaces. They are integrated directly into the sewer pipe and are attached to the bottom of the pipe. The wastewater thus flows through the heat exchanger surface and can deliver the heat contained to the water or the brine in the intermediate circuit [17]. Figure 22 shows the scheme of heat recovery in the duct by means of channel heat exchangers.
- Tube exchangers are similar to sewer heat exchangers attached to the channel bottom but consist of steel tubes that are cast with a special mortar. The advantage of these systems lies in the low cross-sectional reduction. However, the heat transfer is deteriorated because the waste water has no direct contact with the pipes [17].
- The PKS Thermpipe is a sewer pipe wrapped with support tubes that have been modified and can now be charged with a heat transfer medium. As a result, the entire sewer pipe becomes a heat exchanger. This technique makes it possible to extract energy from the sewage and the surrounding soil through a pipe system [19].
- The so-called Heatliner provides an alternative to sewer or tube heat exchangers. The product is a construction consisting of several thin tubes, which are combined between two "liner" layers to form a heat transfer mat [20].
- The HUBER ThermWin-System represents another way of using heat from the sewer system. Part of the waste water stream is taken from the sewer and coarsely pre-cleaned by a sieve system. Thereafter, the freed of coarse materials, waste water is passed to an above-ground heat exchanger. Through the heat exchanger, the energy is delivered to the heat transfer medium in the intermediate circuit. After thermal use, the waste water, together with the previously screened substances, is returned to the sewer system [21].

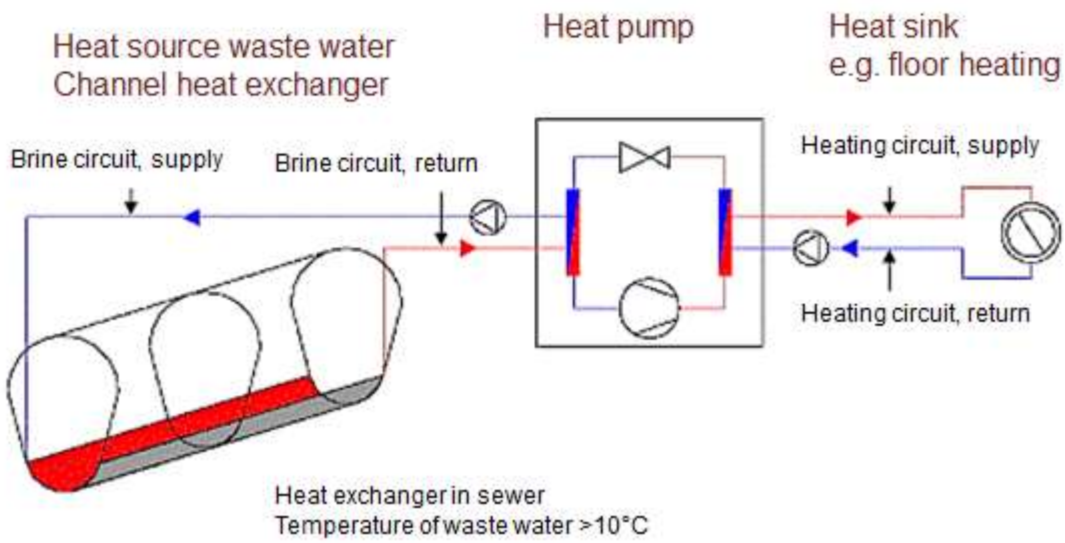


Figure 22 - Scheme heat recovery in sewer [18]

When using heat from flue gas, condensation heat is removed from the flue gas combustion processes until it comes to a condensation of the water contained in the flue gas. Condensation not only uses sensible heat (cooling down to dew point) but also latent heat (during condensation). The scheme of a possible integration is shown in Figure 23.

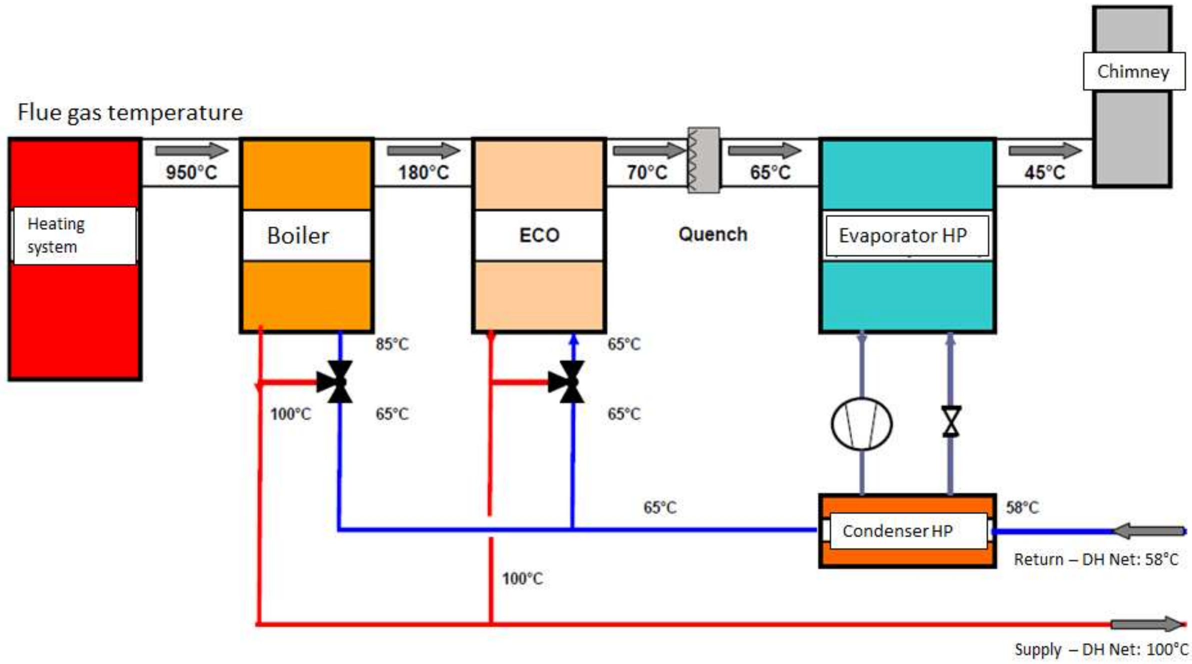


Figure 23 - Scheme of a possible WP integration for flue gas condensation [22]

The capacity of the heat pump is usually dimensioned for maximum utilization rate in accordance to the minimum boiler load, i.e. the boiler load in the summer. This ensures a high number of full-load hours for the heat pump. Furthermore, the composition and amount of flue gas (e.g., water content) has to be considered in the design of the heat pump. The acidic components in the flue gas can cause corrosion on the evaporator and other parts of the system that are in contact with the condensate. These parts of the system must therefore be made of acid-resistant material, whereby the investment costs are increased. To avoid damages of the evaporator of the heat pump, it is mostly separated from the flue gas through an additional hydraulic circuit.

2.6.2 Design of the heat pump

The design of heat pumps for thermal networks is, as described above, mainly dependent on the heat source. Depending on the possible extraction capacity, the heat pump can be dimensioned. The heat exchangers and the unit itself are designed for the respective frame conditions. Other decisive parameters are site-specific restrictions and legal issues. Furthermore, the properties of the heat distribution system (here the thermal network) such as temperature and volume flow have to be suitable for the operation of the heat pump. Since heat pumps for district heating applications are in most cases not standard or series products, the devices are designed by the respective manufacturer in cooperation with the network operator for the specific application.

As a basic rule for the design of heat pumps in thermal networks with unlimited heat sources, it can be assumed that the basic load should be covered by heat pumps. This ensures that the heat pumps have a high number of full load hours. In addition, the required installed capacity can be kept small (including lower investment costs) and still cover a significant part of the heat load. Figure 24 shows the annual duration line with base load coverage.

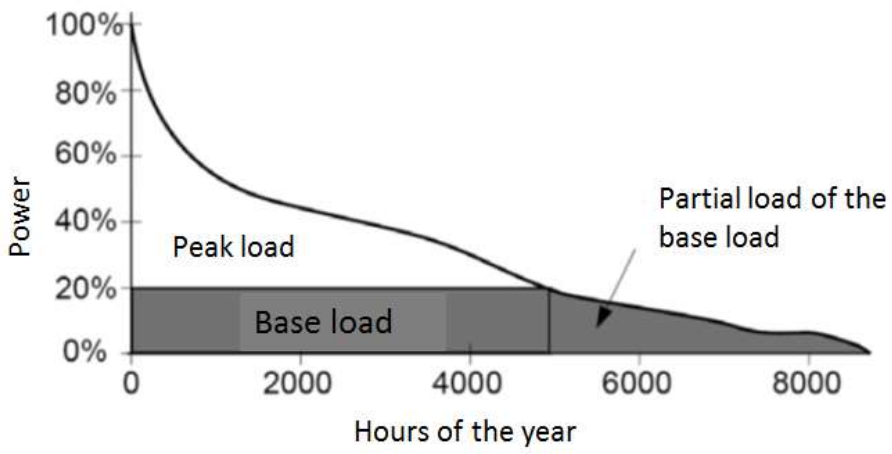


Figure 24 - Annual duration line with base load coverage

Depending on the connection option, see Sections 2.5.3 and 2.5.4, heat pumps can also be integrated into the return line to ensure high efficiency at lower temperatures. The maximum flow temperature required for some district heating networks can be reached by post-heating with e.g. combustion processes or e-boilers.

Otherwise, the design of heat pumps may be similar to that of other producers of district heating networks, taking into account such parameters as heat load, simultaneity factor, etc.

2.6.3 Design of the heat sink

When designing heat pumps in the small power range or for individual buildings, the main focus is on adapting the heat pumps to the required heating load of the building, including the supply of hot water. HP in thermal networks are in most cases not monovalent and combined with additional (peak load) producers. For this reason, the possible extraction power of the heat source is considered in the dimensioning or a cost optimum between investment costs and possible operating costs is chosen to achieve a certain amortization in a few years. Nevertheless, in a thermal network, the heat pump performance has to be adapted to the temperature decrease in the network. The heat sink is in this case the district heating system or the customers connected to it. The design of the network has to be the same as for conventional heat generators. In order to create suitable conditions for heat pumps, the temperature level of the network should be as low as possible which depends mainly on the type of customers connected. When the temperatures are suitable, the heat pump can be operated particularly efficient and cost-effective. For this reason, the heating systems of the customers should have low temperature heating systems. The distribution of heat in networks with low temperatures, due to the smaller possible temperature difference between flow and return requires usually higher volume flows and thus larger pipe diameters to transport the corresponding heat. For new networks, this fact can already be considered during planning. In the case of existing networks, this is usually not possible, which requires an early clarification as to whether low temperatures are possible or whether additional after-heating elements (for example, boilers, electric heaters, etc.) may be necessary. Furthermore, the integration of a thermal storage or the storage capacity of the network has to be taken into account. Any peak loads that require higher powers and / or temperatures can be damped thereby.

2.6.4 Storage systems

Short-term (daily) storages must be differentiated from long-term (seasonal) storages. Whereas the first type is used to balance the daily peaks, the second type is meant to store excess or renewable heat during the summer for the next winter. Short-term storages are defined by at least 20 cycles up to 500 cycles charging and discharging per year. Long-term storages are often characterized by one cycle per year.

Motivation
Storage systems can improve the efficiency and profitability of district heating systems and help having a better operation of heating systems due to increased flexibility. Despite their advantages, district heating networks rarely use them into their system. Reasons for that are higher investments, lack of space or lack of knowledge.

Short-term (daily) storages

Short-term storage systems are used to either shave peaks or increase the power of the network.

A storage system can decouple heat production from heat consumption. This is important because the electricity peaks do not occur at the same time as heat demand peaks. In a large district heating network, load balancing is demanding. The daily response in the district heating network is strongly fluctuating. These capacity changes in the network can be easily balanced by storage systems. With a daily storage, the maximum daily capacity can be reduced by about 30 %.

Storage systems can increase the power of the network, when placed decentral, i.e. at the user's location. Furthermore, decentralized storage systems fed by renewables energies can enable the network to connect more houses with the same capacity.

Long-term (seasonal) storages
The following long-term storages are considered:
ATES: aquifer thermal energy storage

Aquifers, i.e. naturally occurring self-contained layers of ground water, are used for heat storage. Heat is fed into the storage through wells and taken out by reversing the flow direction. Aquifers cannot be found everywhere. Thus, an extensive exploration program has to be passed for the building site before one can be sure that an aquifer thermal energy storage is suitable.

BTES: borehole thermal energy storage

In this kind of storage system, the heat is directly stored in the water-saturated soil. U-pipes – the so-called ducts – are inserted into vertical boreholes to build a huge heat exchanger. While water is running in the U-pipes, heat can be fed in or out of the ground. The heated ground volume comprises the volume of the storage. The upper surface of the storage is heat insulated.

TTES: tank thermal energy storage

A tank thermal energy storage is built as steel or reinforced pre-stressed concrete tank, and as a rule, partially built into the ground. The storage volume is filled with water as storage medium.

PTES: pit thermal energy storage

The usually naturally tilted walls of a pit are thermally insulated and then lined with watertight plastic foils. The storage is filled with water and a heat insulated roof closes the pit.

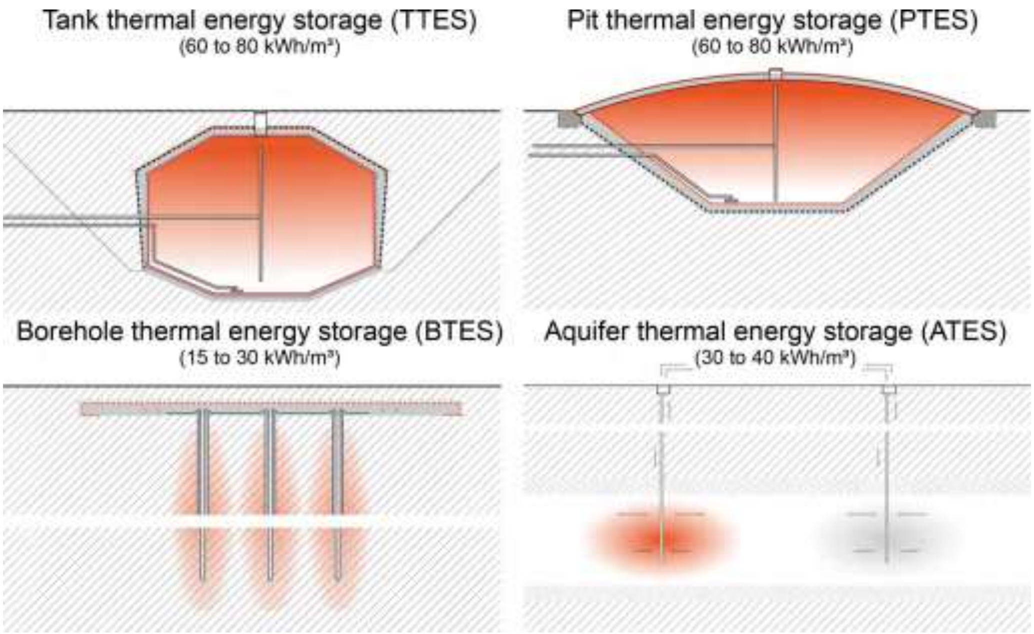


Figure 25 - Types of seasonal storages [33]

The economy of seasonal storage systems depends not only on the storage costs, but also on the thermal performance of the storage and the connected system. Therefore, each system has to be examined separately. In this context important parameters are the maximum and minimum operation temperatures of the storage and of the district heating system. Obviously, heat from the storage can only be used without a heat pump as long as the storage temperature is higher than the return temperature of the district heating system. To determine the economy of a storage, the investment and maintenance costs of the storage have to be related to its thermal performance. This quantity is equivalent to the cost of the usable stored energy. Examples show that the profitability of thermal storage systems is mostly given [34], [35].

Table 9 – Overview of types and characteristics of seasonal storages [33]

Tank	Pit		Borehole	Aquifer
Storage medium				
Water	Water	Gravel-Water	Underground / rock	Sand / Gravel-Water
Heat capacity in kWh/m³				
60-80	60-80	30-50	15-30	30-40
Storage volume for 1 m³ water equivalent				
1 m³	1 m³	1.3 – 2 m³	3-5 m³	2-3 m³
Hydrogeological requirements				
- Underground stable - Possibly no groundwater - 5-15 m deep	- Underground stable - Possibly no groundwater - 5-15 m deep		- Digging the underground possible - Groundwater is advantageous - High heat capacity - High heat conductivity - Low hydraulic permeability (kf <10^{-10} m/s) - Ground water streaming < 1 m/a - 30 – 200 m deep	- Aquifer layer with high hydraulic permeability (kf > 10^{-5} m/s) - Closing layers over and underneath - Low ground water flow - Optimal mixture of ground water by high temperatures - Aquifer-Layer > 30 m
Advantages				
- *High heat capacity* - *High charging and discharging power available* - *Good thermal stratifying possible*	- *See Tank* - *Lower investments than tank*	- *Surface usable* - *Lower investments than tank*	- *Surface usable* - *Low investment costs*	- *Big storages easily feasible* - *Low space required* - *Use of additional geothermal energy* - *Low investment costs*
Disadvantages / Risks				
- *High investment costs*	- *Temperature max. limited (ca. 90°C)* - *Big surface*	- *Temperature max. limited (ca. 90°C)* - *Big surface* - *Thermal stratifying layer lower than with water*	- *Temperature max. limited (ca. 90°C)* - *Low charging and discharging power* - *Evtl. extra components*	- *Depending on local hydrogeological characteristics* - *Evtl. conflicts with ground water use* - *Chemistry of ground water* - *Evtl. prescription due to water protection* - *Risk of finding a spot*

3. Technical and non-technical evaluation of HP integration - Key Performance Indicators

The integration of heat pumps in DH networks is particularly interesting for the implementation of different types of measures to improve the network performances. By providing more flexibility to the network supply, heat pumps facilitate demand-side management measures such as load shifting and supply-side measures such as temperature reduction. Indeed, the integration of HPs allows reducing the network supply and return temperatures and thus a higher share of renewable and fluctuating heat sources can be used and by doing this, the environmental impact of the network is reduced. Additionally, lower network temperatures decrease the heat losses and consequently better energy performance of the network is possible [24, 25, 26, 27].

To assess the integration of heat pumps in DH networks quantitatively and qualitatively, it is important to select and determine/calculate appropriate Key Performance Indicators (KPIs), for the heat pump and the overall network. The most common and widely used KPIs are:

- For the Heat Pump: COP, SPF (seasonal performance factor), payback period
- For the DH network: CO_2 emissions, Primary Energy consumption (direct consumption or energy savings), Load reduction, Return On Investment (ROI)

These KPIs are used in most of the studies because they give a good indication over the three main domains: Energy, Environment and Economy. However, to make them meaningful they have to be calculated using the same system boundaries and they have to be compared using the same reference situation. For HP, the COP is the most common KPI between manufacturers, network operators and researchers. For instance, [28] presents an in-depth comparison of HPs performances using extensively COPs.

Additionally, to these common KPIs in [27] the authors calculated the Coefficient Of System Performance (COSP). As a HP can have a high COP but also consumes power produced by a CHP for instance, it impacts the total system performance. To be more precise, a performance indicator based on the complete system is needed. The COSP is used for evaluating the complete supply scheme of a specific system. It includes all the various requirements for supplying the heat demand, and supplied by the network where $Q_{HeatSupply}$ is the heat supplied by the main heat source (or the HP), Q_{Demand} is the heat demand, $Q_{DH,loss}$ the heat losses, $\sum Q_{BoosterHP}$ the heat supplied by booster HPs if existing and $\sum W_{BoosterHP}$ their electricity consumption. $W_{HeatSupply}$ is the power consumption related to the heat supply $Q_{HeatSupply}$ and $\sum W$ the total energy consumption within the system.

$$\boldsymbol{COSP} = \frac{Q_{HeatSupply} + \sum Q_{BoosterHP} - Q_{DH,loss}}{W_{HeatSupply} + \sum W_{BoosterHP} + W_{Pump}} = \frac{Q_{Demand}}{\sum W} \qquad \text{Equation 3}$$

Although it is not extensively used, the exergy efficiency is also a KPI which assesses the performance of a well-defined system under specific reference conditions. It has been studied in detail with several examples of low-temperature DH networks including various types of renewable and waste heat energy sources in the ECBCS Annex 49 [29] and IEA EBC Annex 64 [30]. As there are many ways to calculate the exergy efficiency of a system (for example by using temperature level, or energy contents, or efficiencies and mass flows, etc.), no equation is given here, but can be found in [29] and [30].

The ECBCS Annex 49 concludes that “the method of exergy analyses has been found to provide the most correct and insightful assessment of the thermodynamic features of any process and offers a clear, quantitative indication of both the irreversibility’s and the degree of matching between the resources used and the end-use energy flows” [29]. It can be applied to any size of system and for the reasons explained at the beginning of this section, it is especially appropriate when it comes to HP integration DH networks

For a better understanding of the system performance and to be interpreted in a more comprehensive way, Exergy indicators are often used in combination with other economic and environmental indicators. Some successful conducted studies indicate a cost reduction potential for innovative low temperature heat grid community solutions based on the exergy thinking concept of about 10-18% and a CO_2 free heat delivery process [30].

Besides these quantitative KPIs, qualitative KPIs can also illustrate the system performance, through for instance user behaviors (increase/reduction of peaks in case of bad/good operation of HP) or customers feedback.

Many indicators are available; however, the calculation of Key Performance Indicators depends on the considered system and the type of information required and needed. It is critical to choose the appropriate KPIs to get a meaningful statement, which can be used for further improvements of the system.

3.1 Decision support tool for selection and integration of heat pumps

The sections above provide a first overview of parameters to observe when integrating heat pumps into district heating grids. However, an investment always requires a detailed analysis and calculation considering the respective framework conditions. Based on the general methods described here and based on so-called best practice systems, an Excel tool was developed. With its help, a first estimation can be made as to whether the integration of a heat pump into a thermal network makes sense or whether alternatives are required. For this purpose, the experience gained from existing facilities was recorded in a database and based on simplified calculations, technical, ecological and economic key figures can be determined. The tool and its functions are briefly described below.

The tool was realized with VBA and contains several tabs with different queries and the results. Green fields or drop-down menus represent input fields, white fields are used to display results, and red fields are activation buttons.

References
"References" is the first of eight tabs of the tool. This serves to present similar, already realized projects. These references are intended to serve as a guide only and to show where similar equipment is already in operation and how it works.

The following data is shown:

- Location (state, city)
- Heating power
- Flow temperature of the source and the heating circuit
- Achieved COP
- Manufacturer (company name)
- HP product name
- Figure 26 shows a screenshot of the first "References" tab.

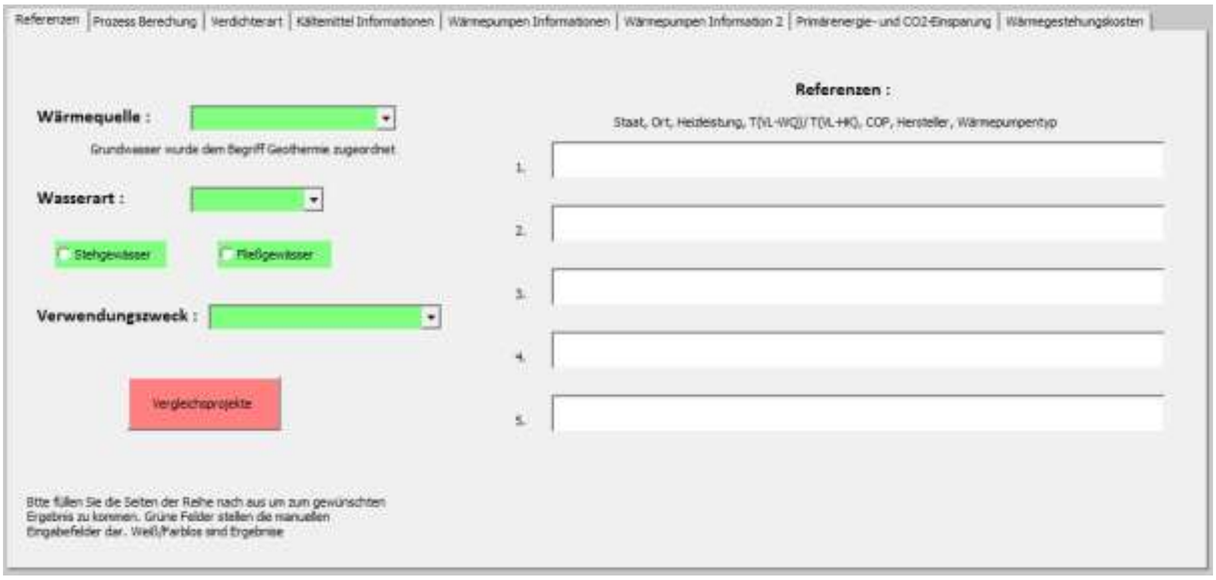


Figure 26 - First register of the tool (references)

Process calculation

Figure 27 shows the structure of the second tab of the tool. Technical details of the heat pump, such as COP, electrical and thermal power, required mass flows, etc. are determined.

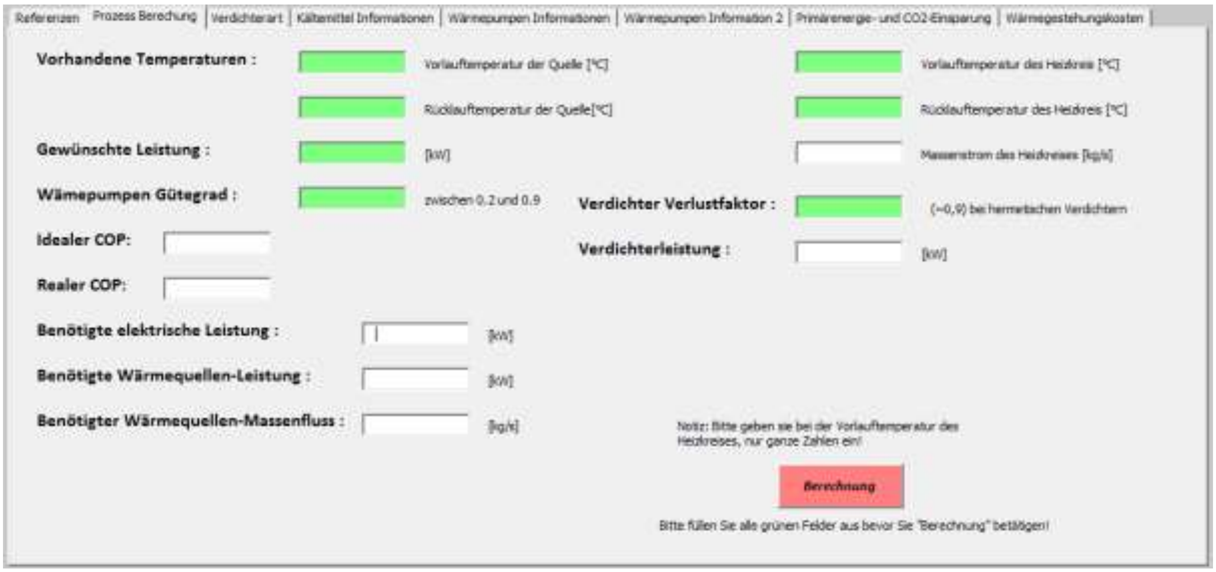


Figure 27 - Second register of the tool (process calculation)

Compressor design

In this register, the number of required compressors is determined. The calculated number represents a minimum, since a maximum possible volume flow of the respective technology is assumed. In the first step, no consideration is given to the products available on the market and the refrigerants intended for this product. The desired heat output, as well as the flow temperature of the heating circuit are taken from the first register and are used for a better overview. After selecting the refrigerant, the volumetric cooling capacities are determined with the help of stored data. An error message is generated if the selected flow temperature is not available in the data table of the refrigerant. In this case, the user is asked to change his input regarding the temperature or the refrigerant. The stored values are part of the graphic shown in Figure 28. The number of compressors is calculated by means of the required volume flow of the refrigerant and the stored technical data of the fluids.

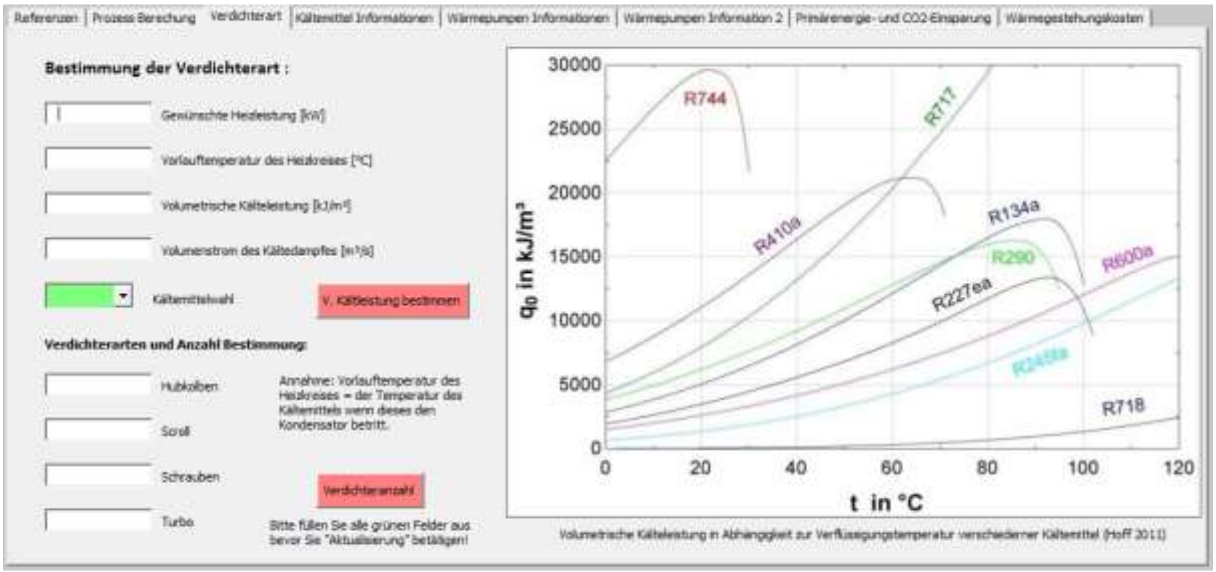


Figure 28 - Third register of the tool (compressor type)

Optional refrigerants

The fourth tab of the tool (Figure 29) deals with possible other refrigerants. After entering the flow temperature of the heating circuit, after pressing the button, all refrigerants are listed, which can also be used under these conditions. The only exception is the refrigerant CO_2 (R744), which has a critical temperature of ~ 31 °C but can also be used for higher temperatures. When using such a refrigerant, more information must be obtained from the manufacturer. For the calculation procedure for the selection of the suitable refrigerant a simplified approach was used (flow temperature of the heating circuit + 25 K). For an exact statement, detailed calculations for the respective refrigeration cycle (for example determination of the maximum cycle temperatures) would have to be made. The tool should allow a first estimate as described above. Detail calculations must be performed in a later step.

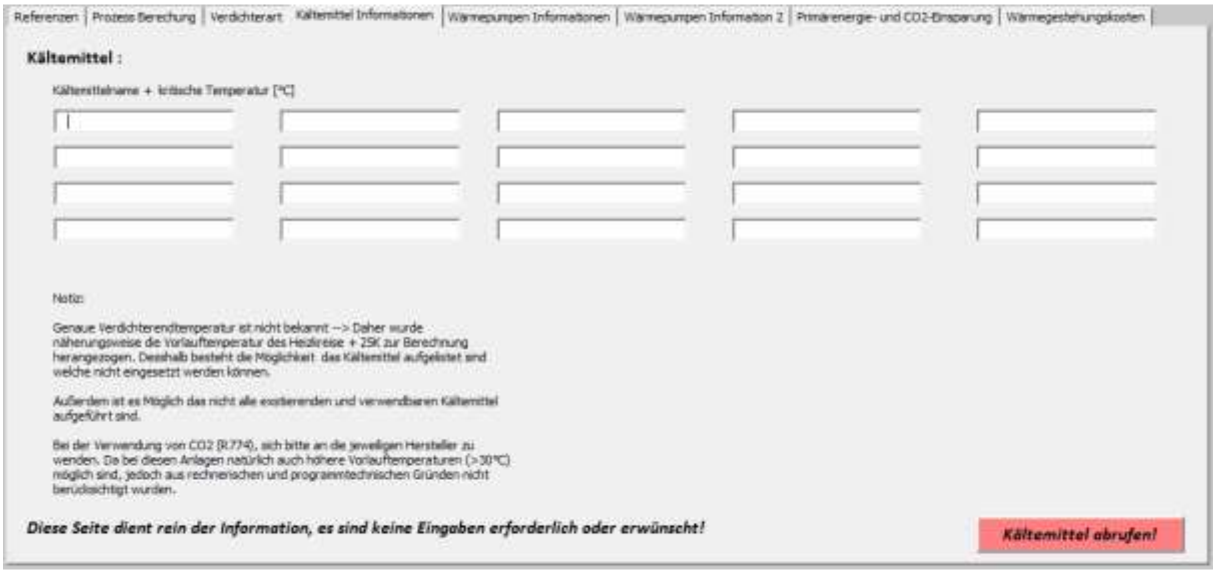


Figure 29 - Fourth register of the tool (refrigerant information)

Heat pump suggestions

In registers 5 and 6, after entering the "flow temperature of the heating circuit" and selecting a refrigerant, all the heat pumps in the database that can operate under these conditions are displayed, see Figure 30.

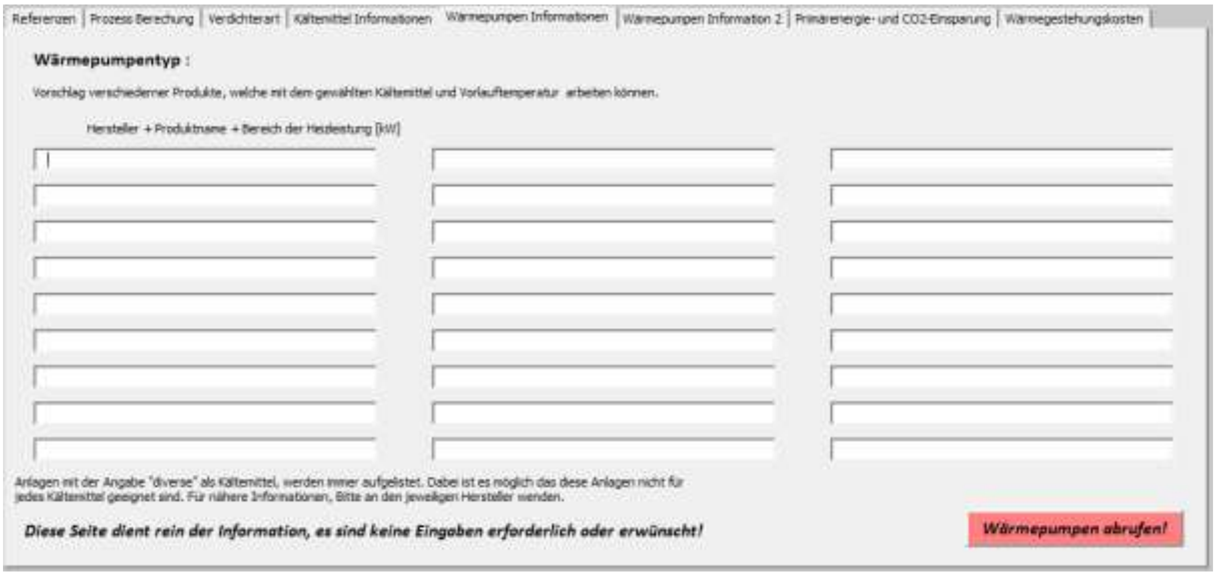


Figure 30 - Fifth and sixth register of the tool (heat pump information 1 & 2)

CO_2 and primary energy savings

Here the primary energy and the CO_2 savings are calculated (see Figure 31). Only the difference to the respective benchmark system is mapped. In the case of a positive result, the heat pump would require less CO_2 or primary energy than the comparison system. The listed values "Reference" are valid for Austria and are used for comparison with the calculated data. The selected factors can be adjusted before starting the tool in the respective Excel sheet "Primary Energy Factor and CO_2 emissions".

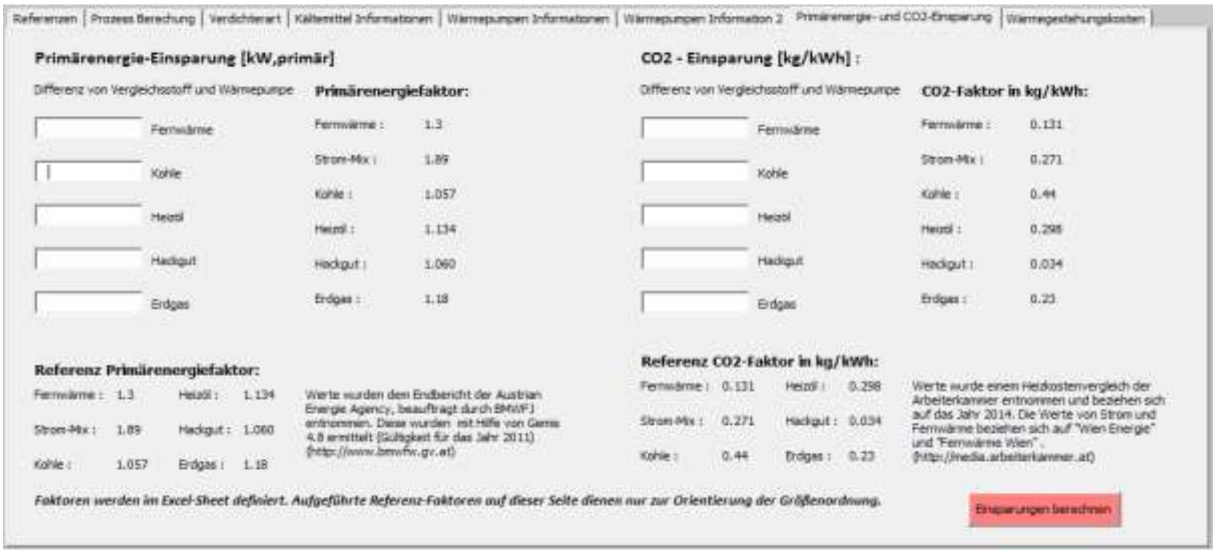


Figure 31 - Seventh register of the tool (primary energy and CO_2 savings)

Heat generation costs

In the last tab of the tool, the heat production cost of the system can be calculated. As can be seen in Figure 32, parameters are required here, e.g. investment, maintenance and personnel costs, other costs, electricity price, etc. Some of these data are not yet known in many cases, in as early a project planning phase as the basic estimation of how sensible the integration of a heat pump is. Therefore, estimated costs (e.g., percent of investment cost) can also be given here. The tool gives the user the freedom to enter and adjust the values. Thus, e.g. it can also be determined from which costs or prices a heat pump can be operated economically.

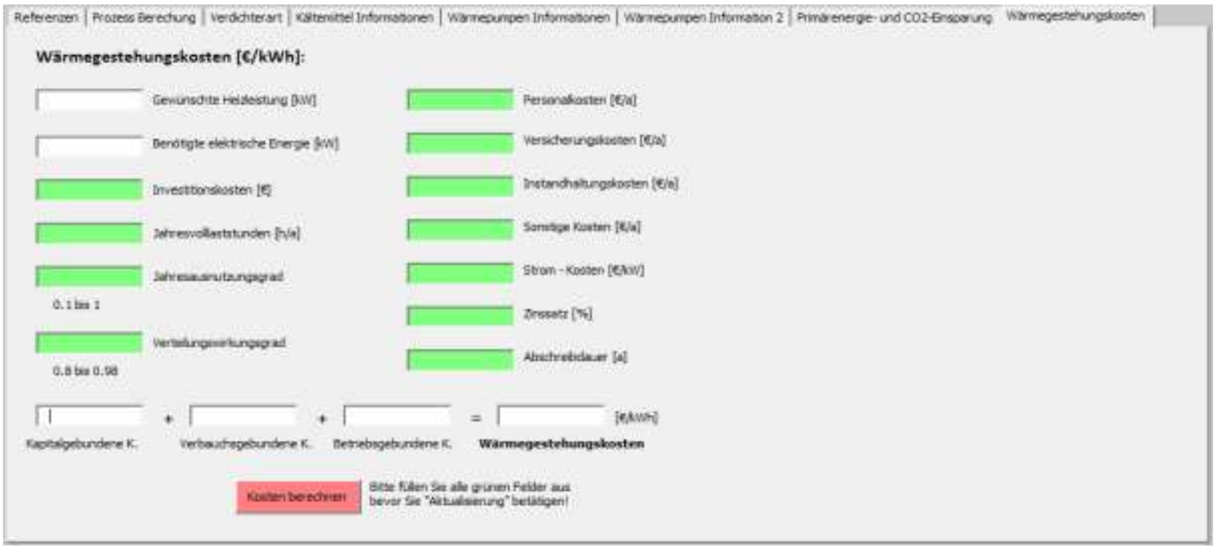


Figure 32 - Eight register of the tool (heat production costs)

3.1 Thermal driven heat pumps

Thermally driven heat pumps use, in contrast to electrically driven heat pumps, mainly thermal energy as driving source, this offers the opportunity for ecological and economic advantages if CO_2 neutral energy sources are used. There are different types of thermally driven heat pumps available on the market. Within district heating networks mainly absorption heat pumps in combination with biomass heating/cogeneration plants are used. The sections within this chapter discuss the basic functionality, a technological and economical comparison between absorption and compression heat pumps.

3.1.1 Functionality

Unlike compression heat pumps (CompHP), absorption heat pumps (AHP) use a so-called thermal compressor for compressing the refrigerant vapor. The electrically driven compressor of a compression heat pump is replaced by a solution cycle and a solvent is required in addition to the refrigerant. The solvent is used to absorb the refrigerant. Two pairs of substances have proven themselves in practice: ammonia (refrigerant) and water (solvent), as well as water (refrigerant) and lithium bromide (solvent). The solvent circuit (= thermal compressor) consists of an absorber, solvent pump, desorber, and a solvent valve. The cycle or difference to the compression cycle is described below and is shown in Figure 33.

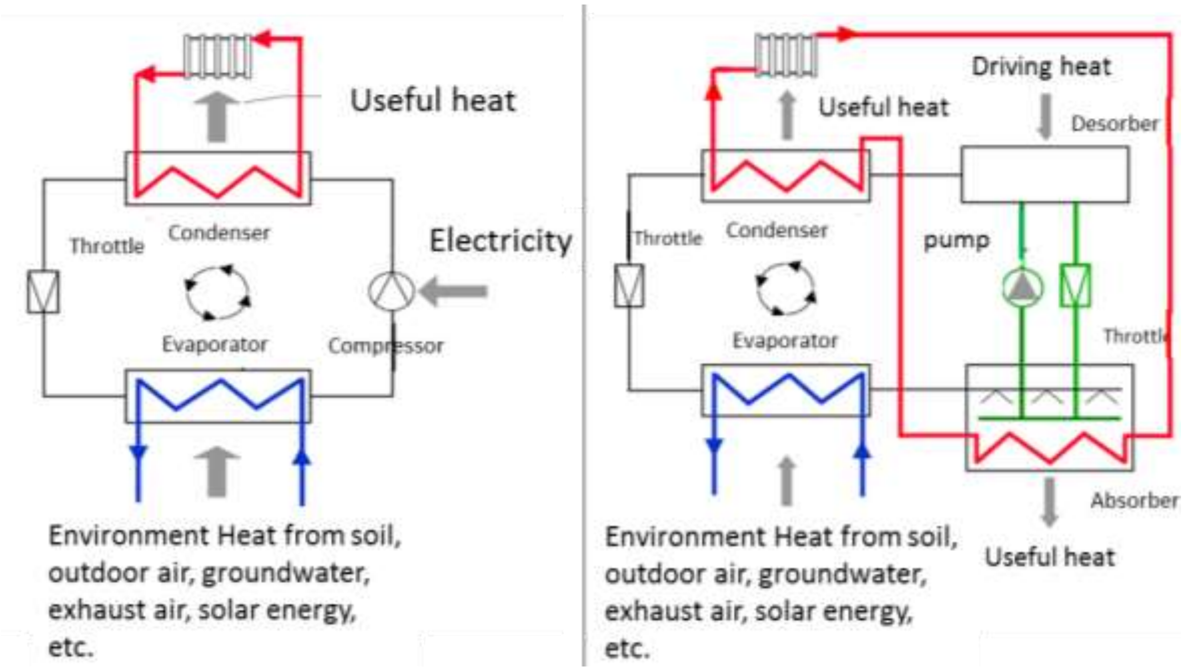


Figure 33 - Comparison between compression circuit (left) and thermally driven absorption operated circuit (right)

The process of thermally driven heat pumps operates within three temperatures:

- low temperature heat source (heat)
- medium temperature heat sink (useful energy)
- high temperature heat source (driving energy)

As with compression heat pumps, a low temperature source vaporizes the refrigerant. The vaporous refrigerant then passes from the evaporator to the absorber. There it is dissolved in the solvent and the absorption heat at middle temperature is released. The absorption heat can already be used by the user. The refrigerant-rich solution is now pumped to the desorber. The pump is powered by electrical energy. However, the required electrical power is only a fraction of the heat released by the absorber and condenser, since a liquid can be brought to a higher-pressure level with far less energy effort than gases. In the desorber, the refrigerant of the refrigerant-rich solution is separated from the solvent using heat at a very high temperature level (e.g., high temperature heating or burning natural gas). Due to the different evaporation temperatures of the two substances, the refrigerant evaporates before the solvent and is thus separated from the solvent. The now refrigerant-poor solution is expanded in the solution throttle to evaporation pressure and the cycle begins again. The refrigerant vapor, at a high pressure and temperature, is passed on to the condenser, where it releases the condensation heat (at a medium temperature level) to the heating system.

3.1.2 Technology comparison

The comparison of the energy balances shows that an AHP consumes less energy than a CompHP to provide the required heating energy. This can be beneficial in cases where little environmental energy is available. In fossil (inefficient) power production, CompHP and AHP (depending on the COP) use similar amounts of fuel energy, see Figure 34. For efficiencies of power production that are worse than shown in Figure 31, the use of absorption heat pumps can be environmentally beneficial. If, however, power is e.g. produced from a mix of renewable energies such as wind or hydroelectric power or with higher efficiencies, the CompHP performs significantly better. Otherwise, if electrical power is provided by the uses of fossil fuels an absorption heat pump is more ecological if it is driven with e.g. CO_2 neutral heat from a biomass heating/cogeneration plant. Depending on the field of application, the ecological advantages of the different technologies can be exploited.

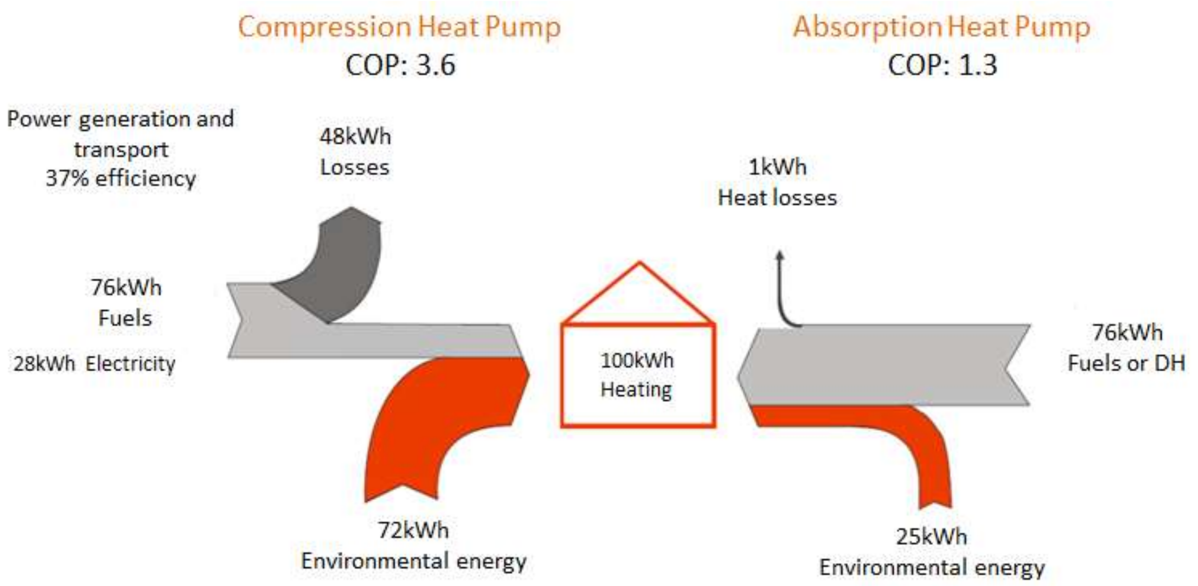


Figure 34 - Exemplary energy balances

3.1.3 Economic comparison

The decision if an operator of a district heating network installs an absorption heat pump or a compression heat pump will be based, beside the ecological point of view, on economic aspects. For economic decisions the payback period which requires the investment and the operation costs is the key deciding factor. The investment costs of absorption heat pumps are mostly higher than those of compression heat pumps. Reliable information concerning the investment costs of large capacity heat pumps are for both systems difficult to obtain and thus this chapter focuses on the comparison of the operation costs.

For the comparison of the operation costs, the heat production costs of the absorption heat pump $c_{heat,AHP}$ and of the compression heat pump $c_{heat,CHP}$ are expressed with the expected COPH of each system. The costs for electricity (c_{el}) to drive the compression heat pump and the costs for the driving heat (c_{th}) for the absorption heat pump for a certain heating capacity $\dot{Q}_{heat}$ are expressed with the following two equations.

$$c_{heat,CHP} = \dot{Q}_{heat}/COP_{H,CHP} \cdot c_{el} \quad \textbf{Equation 4}$$

$$c_{heat,AHP} = \dot{Q}_{heat}/COP_{H,AHP} \cdot c_{th} \quad \textbf{Equation 5}$$

Equation 6 is obtained if Equation 4 is divided by Equation 5. This is an equation for the ratio of the heat production costs of both systems with the cost ratio of electrical power to heat as parameter and the ratio of the COPH from absorption to compression heat pump as the variable. The results of this equation are shown in Figure 35 for different energy cost ratios and show the advantage to produce heat with absorption heat pumps at high energy cost ratios. The result of this equation is an estimation of the heat production cost ratio as the investment costs are not included, a constant COP_H is assumed and dynamic effects such as start and stop losses are also not included.

$$\frac{c_{heat,CHP}}{c_{heat,AHP}} = \frac{COP_{H,AHP}}{COP_{H,CHP}} \cdot \frac{c_{el}}{c_{th}} \qquad \textbf{Equation 6}$$

A general statement concerning the COP_H ratio between absorption and compression heat pumps emphasizes an advantage of the compression heat pump at low temperature differences between the low temperature heat source and the heat sink. This is due to the limited COP_H of single stage absorption heat pumps to a value of about 1.8 while the COP_H of compression heat pumps still increases with lower temperature differences. On the other hand, the COP_H of absorption heat pumps can be held constant, in a certain range, for increasing temperature differences between low temperature heat source and heat sink due to higher temperature of the high temperature heat source while in this case the COP_H of compression heat pumps decreases.

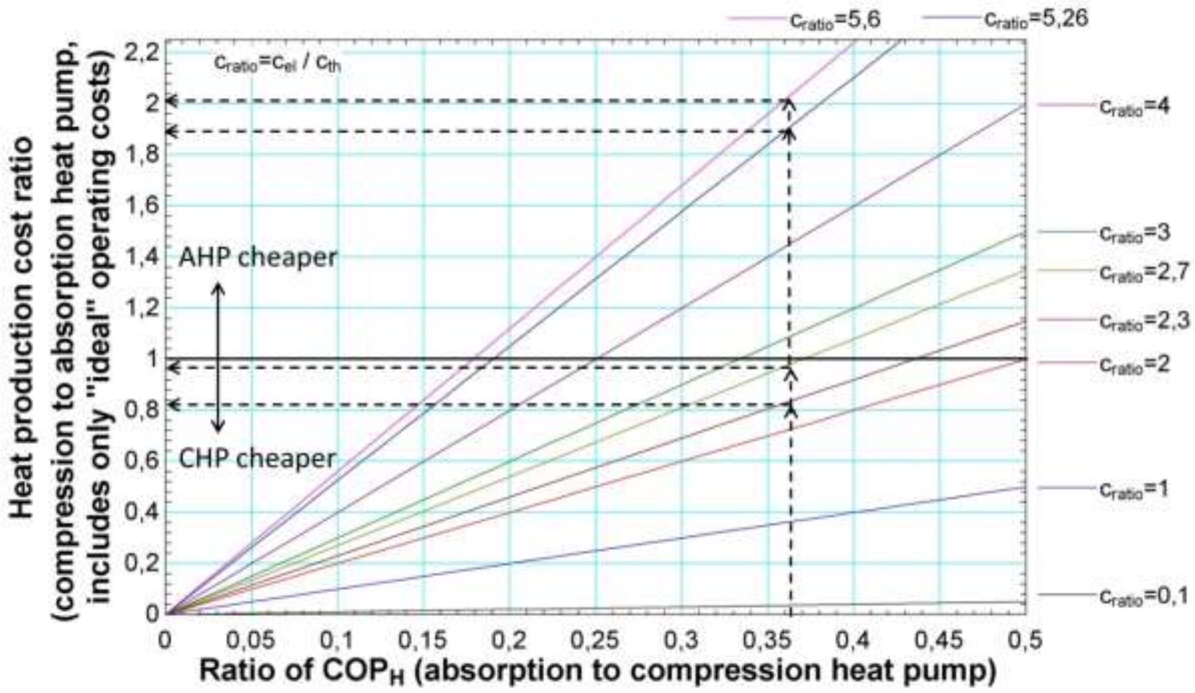


Figure 35 - Analysis of the heat production costs ratio of absorption and compression heat pumps in dependence of the COP_H ratio and energy cost ratio

Assuming the COP_H for both systems according to the previous section leads to a COP_H ratio of 0.36. For the energy cost ratio, the costs for electrical energy and natural gas are used according to the scenario calculations in [41]. This leads to an energy cost ratio of 2.7 for the year 2015, which means that the production of heat with a compression heat pump is cheaper than with a natural gas fired absorption heat pump. For the year 2050, an energy cost ratio of 2.3 can be calculated according to [41] this leads to a change in the direction of lower heat production costs with absorption heat pumps but the heat production costs with compression heat pumps are still cheaper. If other heat carriers are used, for example wood chips, the energy cost ratio for the year 2015 is, according to [41], 5.6 and decreases until 2015 to 5.26, which leads to significant cheaper heat production costs with the absorption heat pump. The expected energy cost ratio development according to [41] shows advantages for compression heat pumps due to lower electricity prices in the future.

3.1.4 Example plant

As already mentioned absorption heat pumps are already being used in district heating systems to increase the efficiency of biomass cogeneration plants using flue gas condensation. One example is the absorption heat pump integrated in the new biomass cogeneration plant Klagenfurt-East that was commissioned in November 2017. The heat pump with a nominal heating capacity of 20 MW reaches a COP_H of about 1.77 at low temperature heat source temperatures of 45/35 °C, heat sink temperatures of 60/70 °C and high temperature heat source temperatures of 130/120 °C. The high temperature driving heat is provided by the condenser of the steam cycle of the cogeneration plant, the low temperature heat is provided by flue gas condensation and the medium temperature heat is supplied into the district heating network. Furthermore, an economizer for direct flue gas condensation with a capacity of about 10 MW is used.

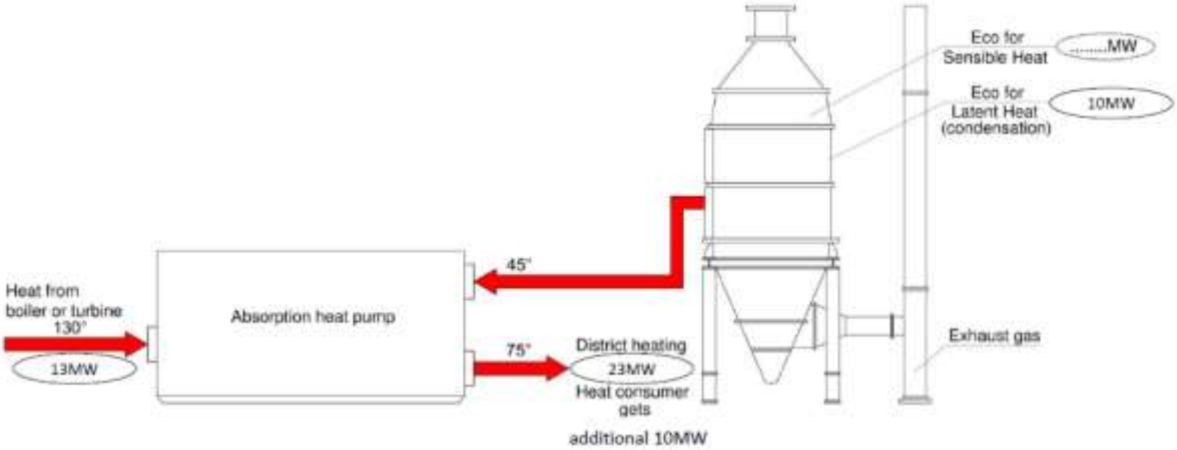


Figure 36 - Scheme of the example biomass cogeneration plant Klagenfurt-Ost [23]

A similar example for the use of an absorption heat pump in combination with a cogeneration plant is used within the district heating network in Salzburg located in Hallein. But in this case only the heat of the absorption heat pump is supplied into the district heating network. The heat of the condenser of the steam cycle is used as driving source and for industrial processes of an onsite industry. Another example is the use of industrial waste heat to supply heat into the district heating network of Innsbruck with an AHP which has a nominal heating capacity of around 6 MW.

4. Modeling and dynamic evaluation for the evaluation of operational strategies

The simulation-supported analysis of district heating systems opens the possibility of virtually depicting numerous problems and developing approaches to solving them, which can be examined for their benefits and effects. Due to the modeling, no intervention in the existing system is necessary. In this chapter, results from former AIT work is summarized based on [31], [32].

4.1 Producer-side management – centralized control strategies for district heating systems in Modelica

An automated strategy is presented using a case study. This enables the potential of using a district heating network as a thermal store to reduce the operating times of peak load boilers during peak load periods and, consequently, their energy supply. In this investigation, the potential of the district heating network with the existing pipelines was investigated, an optimization by means of installation of additional heat storages was not investigated. The methodology developed is divided into the following parts:

1) development of a temperature controller which can be used to influence the flow temperature on the heater side to control the heat supply capacity.

2) to develop an automated analysis for an arbitrary time series to determine the potential of using the district heating network as thermal storage to reduce the operating times of a peak load boiler during peak load periods.

The influence of using network models of different sizes on the results of the potential analysis will be investigated in the next period. In addition, the potential of using the district heating network as a thermal storage in direct connection with the quality of the load curve forecast is also investigated.

4.1.1 Temperature controller development

The first step in the development of this methodology was to develop a temperature controller which allowed the heat output of the heating plant to be controlled by directly influencing the feed temperature on the heating plant side.

Thus, it was possible with this controller to control the heating plant thermal output to a constant value over a limited period by increasing or reducing the feed-in temperature despite fluctuating mass flows.

4.1.2 Temperature controller design

The temperature controller influences the flow temperature on the heater side. Figure 37 shows the structure of the temperature controller using a flow diagram.

The control loop of the temperature controller was implemented by a PI controller. The setpoint value of the heating plant thermal output and the current actual value of the heating plant thermal output are transferred to this as input values. In addition, limits can be set for the output of the PI controller to influence the supply temperature of the heating plant only within a defined range. A correction value for the supply temperature of the heating plant serves as the return value of the control. Figure 38 below shows a graphical representation of the control loop.

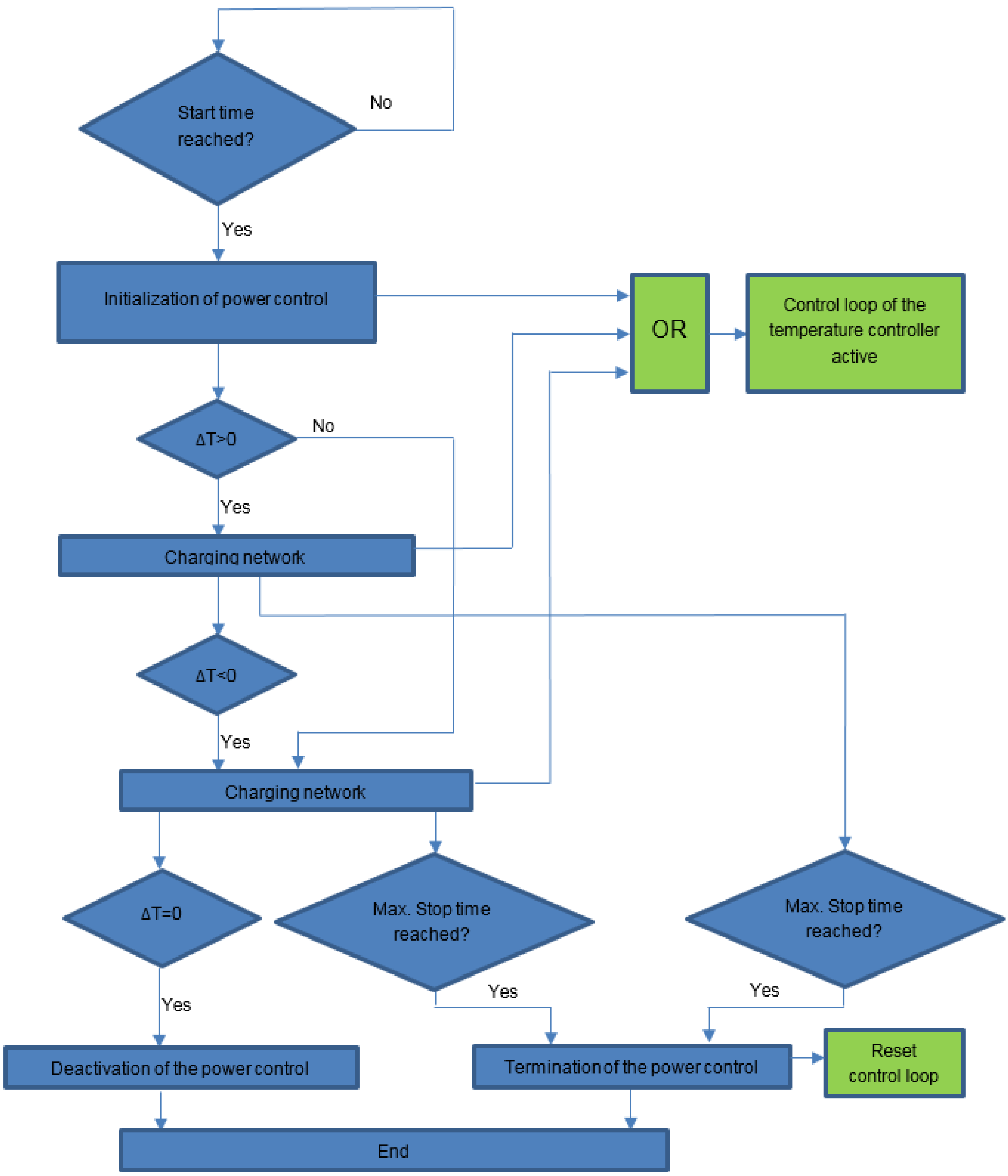


Figure 37 - Flowchart of the temperature controller

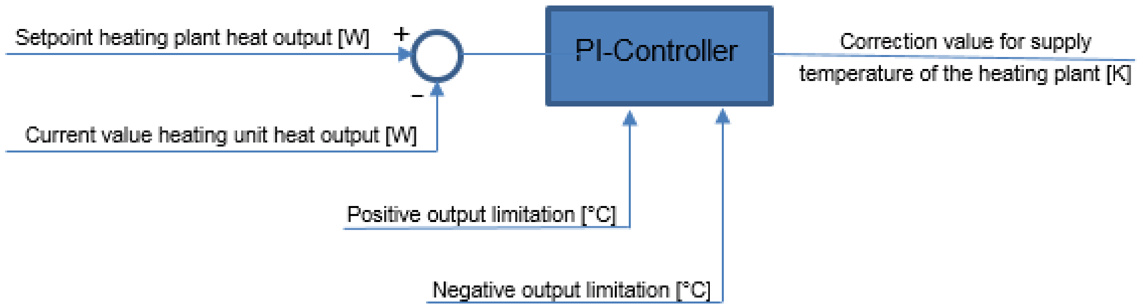


Figure 38 - Control circuit of the temperature controller

4.1.3 Inputs and outputs of the temperature controller

The following list contains all inputs required for operation of the temperature controller and a more detailed description of these signals:

- Start time: Time of day from which power control is activated by the temperature controller. From this point on, the temperature controller tries to control the actual value of the output of the heating plant to the set value.
- Max. Stop time: Time of the day on which the output of the temperature controller must be reset to zero at the latest, if the controller was not previously able to automatically reset the output to zero.
- Power setpoint: desired heat output of the heating plant.
- Actual output value: current measured heat output of the heating plant.
- Positive limitation: Temperature limitation of the output in a positive direction so that it only influences the flow temperature of the heating plant within a defined range.
- Negative limitation: Temperature limitation of the output in a negative direction so that it only influences the flow temperature of the heating plant within a defined range.

A correction value is generated as output of the temperature controller, which influences the flow temperature set in the power plant and thus has a direct effect on the heating plant thermal output.

4.1.4 Functioning with sufficient network storage capacity

Figure 39 below explains how the temperature controller works when there is sufficient storage capacity in the network, using an example of a day's heat supply to the heating plant. As shown in the figure, the dotted, grey and non-optimized original heat supply reaches a maximum value of 88 MW in the morning hours. Assuming that the use of base-load and medium-load boilers can cover a heat supply of 85 MW, it can be concluded that part of the required heating plant thermal output must traditionally be provided by activating a peak-load boiler to cover this morning peak load. An optimized heat supply of the heating plant aims at minimizing the amount of heat generated by the peak load boiler. Thanks to the green, continuously drawn optimized heat supply, the activation of the peak load boiler for this day can be completely dispensed with the case mentioned above. By early charging the district heating network to a higher temperature level, which begins shortly after approx. 03:15, a constant heat output of 85 MW is transferred from the heating plant to the district heating network. This charging of the network is achieved by increasing the flow temperature at the feed point and amounts to a maximum of 4.5 K in the case shown. The positive limitation of the temperature controller, which is set to a value of 15 K in the case shown, is thus never reached. During charging, the network acts as a thermal storage medium using the water circulating in the district heating network.

The optimum switch-on time for activating the temperature controller is determined by a parameter variation which is used to vary the switch-on time within defined limits. One limit is the earliest permissible switch-on time of the temperature controller. This parameter is to be defined by the user and was set in the example shown to a value of six hours before the morning peak load occurs. The other limit is given by the point in time at which the original heat demand exceeds a value of 85 MW and the grid can no longer be charged. By applying a parameter variation, an attempt is made to avoid the use of thermal energy from the peak load boiler and to ensure that the charging of the network does not start too early and thus unnecessary heat losses occur in the network.

Up to approx. 04:45, more energy is supplied from the heating plant to the grid than is simultaneously drawn from the grid by the consumers and by heat losses from the grid. The amount of heat stored in the district heating network can be seen from the green hatched area. In this way it is possible to cover the increased heat demand between 04:45 and 09:00 with the reserves stored in the district heating network and to leave the heat supply of the heating plant constant at 85 MW.

While the original heat supply drops back to a value of less than 85 MW from 09:00, with the optimized version it will remain constant at 85 MW for approx. 45 minutes. This happens since the temperature controller not only increases the flow temperature for charging the network but also has additional functionality for limited discharge of the network. By adjusting the flow temperature below the traditionally set temperature, the heat output can also be limited to 85 MW for a limited period despite increased demand.

The temperature curve shown in Figure 39 shows that this is the case from approx. 07:30 and that the feed-in flow temperature of the heating plant is below that of the original heat supply. In order to compensate for this limited discharge of the network from 07:30, the district heating network is recharged to a higher energy level during the last 45 minutes of active power control by the temperature controller. The amount of heat stored in the district heating network can be seen from the blue hatched area. Finally, at 09:45 the control by the temperature controller is finished.

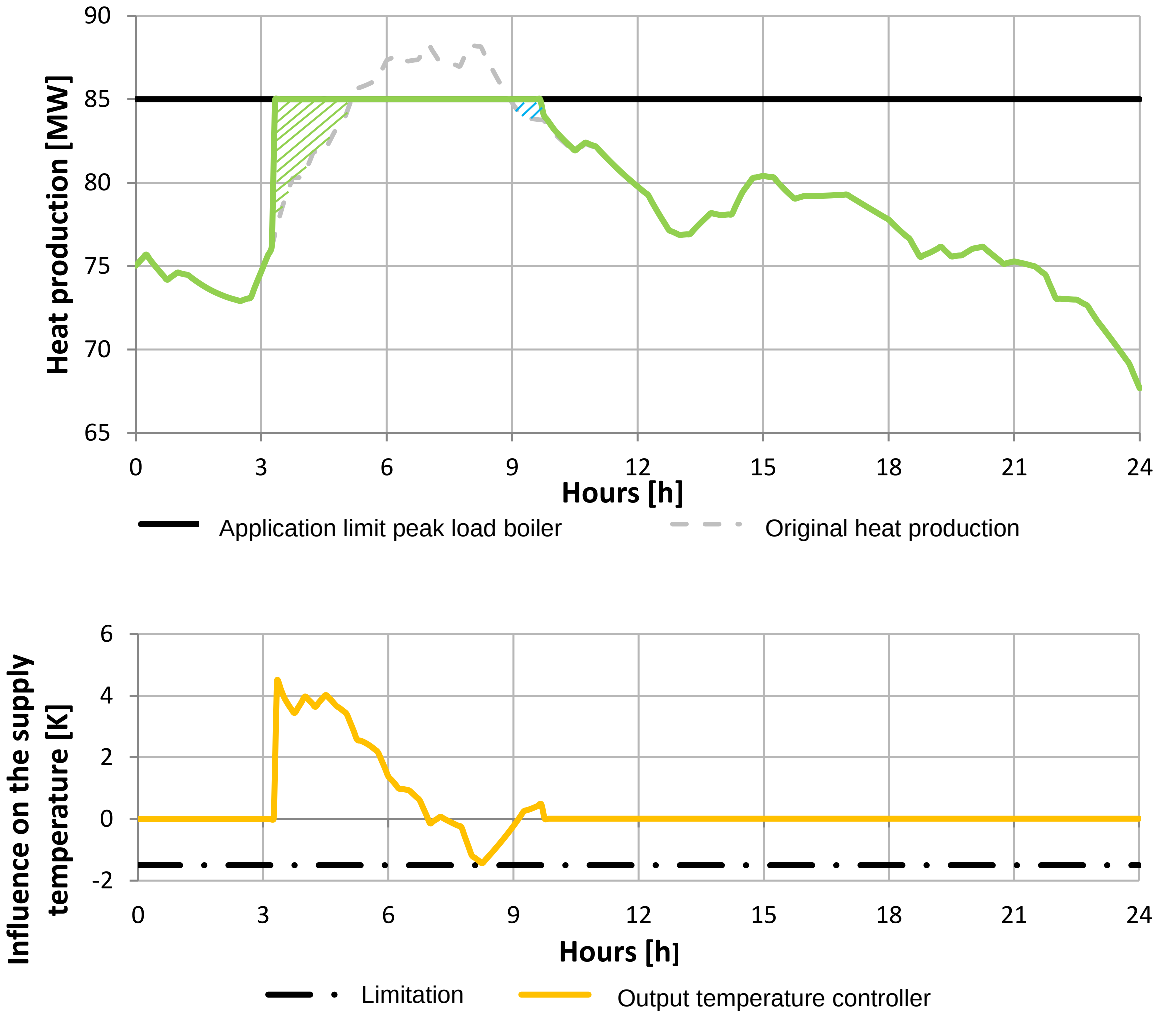


Figure 39 - Optimized heat supply with sufficient storage capacity of the network

4.1.5 Operation in case of insufficient storage capacity of the network

In the optimization of an exemplary heat supply load curve shown above, an example was used by which the optimization applied for the load curve of the day under consideration resulted in a complete elimination of the peak load boiler. The question of whether the use of the peak load boiler can be completely avoided by using the network as thermal storage for the day's heat supply depends on several factors:

- Height and form of non-optimized heat supply.
- Amount of nominal heat output that can be provided by base and medium load boilers.
- Earliest possible switch-on time of the temperature controller from which control may take place.
- Maximum permitted switch-off time of the temperature controller up to which the controller must have automatically reset the output to zero. If this is not the case at this time, the controller is reset using thermal energy from the peak load boiler.
- Maximum permitted temperature increases of the flow temperature of the heating plant by the temperature controller.
- Maximum permissible temperature reduction of the flow temperature of the heating plant by the temperature controller.

In addition to these basic conditions mentioned above, the storage capacity of the network itself plays an important role. The storage capacity is directly proportional to the volume of water circulating in the district heating network.

Figure 40 below shows the heat supply load curve of an optimized day for which a complete renunciation of the use of thermal energy from the peak load boiler was not achieved.

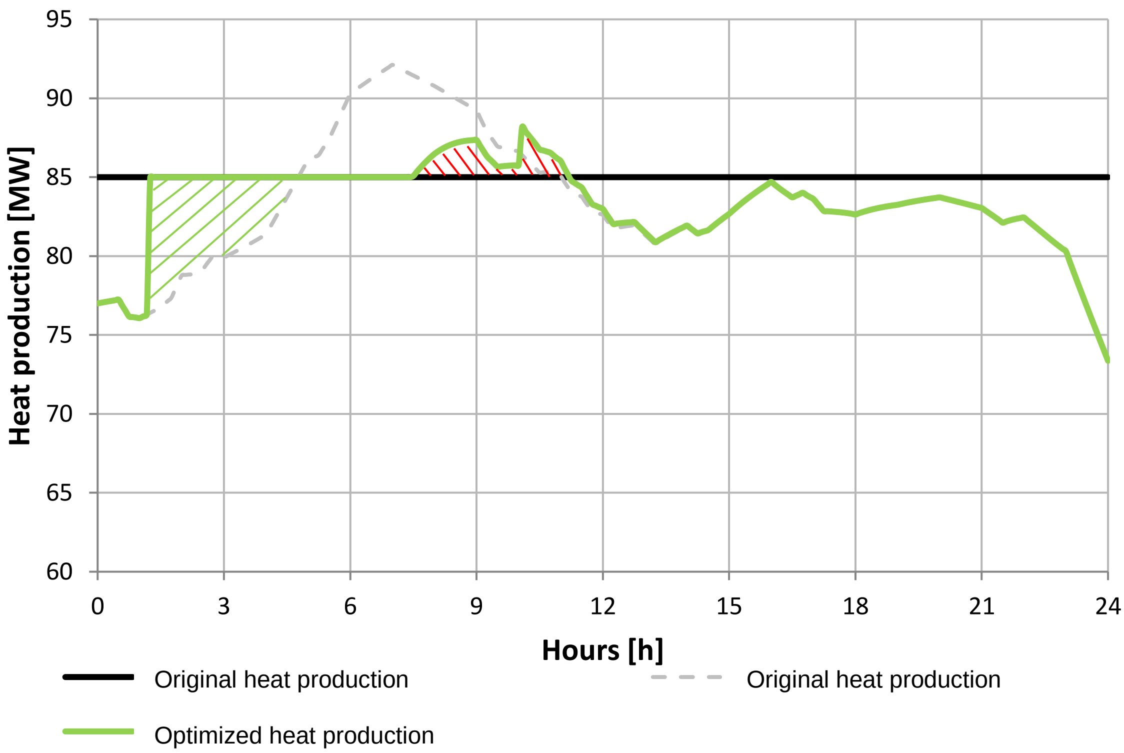
As in the example above, the original, non-optimized heat supply curve is shown in the dashed grey line in the diagram above. The green continuous line shows the course of the heat supply after the optimization has been carried out. In this example, the charging of the network starts at the earliest permissible switch-on time of the temperature controller. This parameter is to be defined by the user and has been set here to a value of six hours before the morning peak load occurs.

By charging the network, which starts at approx. 01:15, the heat supply of the heating plant can be kept at a constant value of 85 MW until approx. 07:30. At this point, however, despite the increased heat demand of the consumers, no further discharge of the district heating network is possible, since the temperature controller has already reduced the feed temperature of the heating plant as far as permissible and the negative output limitation of the controller has been reached. This is also shown in Figure 40 during the output of the temperature controller.

When the negative output limitation of the temperature controller is reached, there is no further reduction of the feed-in temperature and thus no further regulation of the heat output to a value of 85 MW. This means that the peak load boiler must be used to cover the additional heat requirement. Since the moment the maximum stop time of the temperature controller, which was set to a value of 10:00, is reached, it has not already automatically regulated the control, the controller is reset, and the controller output is abruptly reset from -1.5 K to 0 K. This causes a peak load in the heat supply at time 10:00, as can also be seen clearly in Figure 40.

The green hatched area in the figure represents the additional heat energy of the base and medium load supply compared to the original heat supply curve. The red hatched area shows the amount of heat still required by the peak load boiler.

In the example in question, the premature charging of the grid with 18.49 MWh of additional heat supply due to base and medium load production can prevent peak load energy of 17.93 MWh from being drawn down. 5.59 MWh must still be provided by activating the peak load boiler. Compared to the original heat generation, however, a reduction of 76% has been achieved for heat that has to be provided by the peak load boiler.



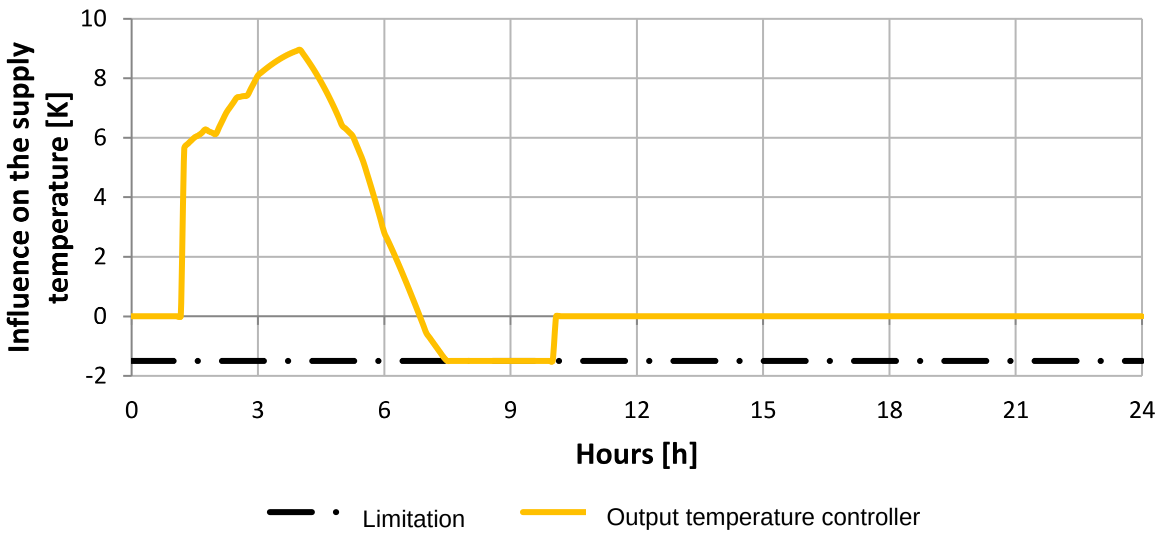


Figure 40 - Optimized heat supply in case of insufficient storage capacity of the network

4.1.6 Automated analysis of a time series

With the developed temperature controller, it was possible to optimize the heat supply process daily-based. To automate this process, a script was created that performs an analysis over an arbitrary time series. This script contains the following steps (Figure 41 shows the appropriate flowchart):

Step 1: Simulation run for any day with deactivated temperature controller. This detects the height and shape of the morning heat supply peak.

Step 2: Deciding whether an early charging of the network is necessary for the day under consideration or whether this is possible at all:

- A low morning heat supply peak, which is below the rated output of the base and medium load supply, does not require early charging of the network.
- An early charging of the network may not be possible for a daily cycle that generally shows a high level of heat supply at any time of the day due to bad weather conditions or other influences since the heat supply process is constantly above the nominal output of the base and medium load supply.

Step 3: Based on the previous analysis, a possibility of reducing the operating time of the peak load boiler and the amount of heat produced by it was determined for the day in question, several simulations run with the temperature controller activated are carried out:

- A parameter variation is used to determine the optimum start time for the district heating network charging process.
- As an optimization criterion for parameter variation, the heat generation costs for the heat quantities originating from base and medium load provision and those from peak load provision are used.

Step 4: After the best possible start time for the activation of the temperature controller has been determined by means of the parameter variation, a data backup of these settings is performed in a further simulation run with the temperature controller activated and the start time set to the optimum.

Step 5: This step determines how uncertainties in the prediction of the heat supply process will affect the results. Because the heat supply process of the entire day is analyzed to determine the optimal start time for charging the network, this analysis can only take place using a heat supply forecast. The actual heat supply process is not known in advance. Since these forecasts always contain a forecast error, the effects of forecast errors of different amounts on the results are evaluated in a final step.

For this purpose, simulation series with varying total deviations in the heating loads of the consumers are carried out for each day for which a possibility of network charging to reduce heat supply by the peak load boiler was determined in step 2. These variations have a range of ±5% and reflect the forecast uncertainties in the prediction of the heat supply process. The simulations are carried out with an active temperature controller and the optimum start time of network charging determined in step 3 based on the heat supply forecast.

Step 6: The characteristic values obtained in this way for the day in question are saved in a text file. This text file contains all data for all days of the time series under consideration and is used for subsequent evaluation of the results.

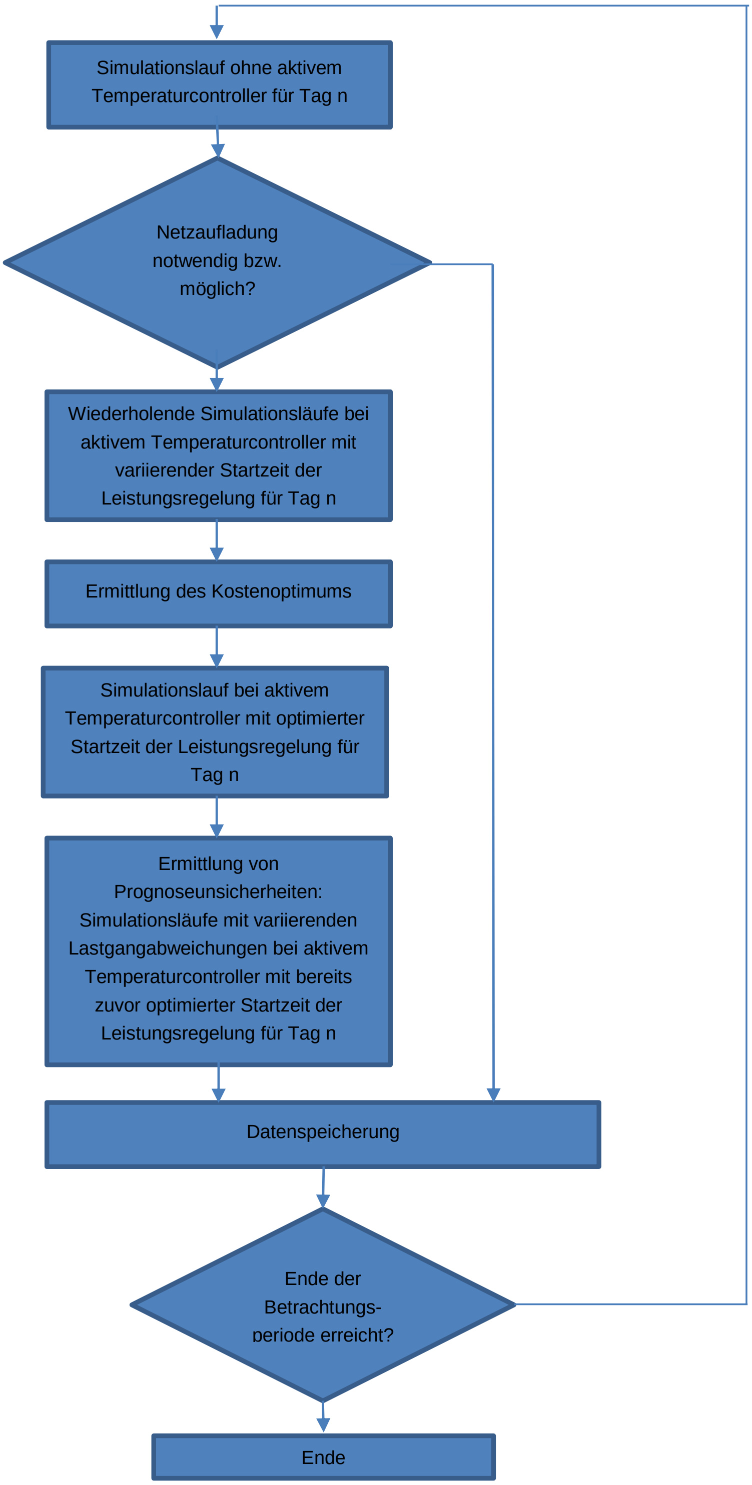


Figure 41 - Flowchart of the automated analysis script

4.2 Customer management – centralized and individualized control strategies for district heating systems in Modelica

The work for the Demand Side Management (DSM) part followed the following research question: "Does demand side management increase the efficiency of heat pumps in a district heating network?" A representative district heating network for Austria, consisting of a biomass boiler for base load and an oil boiler for peak loads, was considered as an example. The existing generators will then be replaced by a heat pump in combination with a heat storage. The following scenarios were investigated:

Baseline
The basic scenario for the heat pump with a heat storage includes a simple operating strategy. This concentrates on the charge/discharge cycles of the thermal storage. The heat pump is switched on when the storage tank is completely discharged (top layer temperature is below the setpoint) and switches off when the tank is fully charged (bottom layer temperature is above the setpoint). There are no other factors that affect the operation of the heat pump and the storage.

Dynamic Pricing
The heat pump is not only loaded and unloaded based on the temperature level, but also depending on fluctuating electricity prices. This process is based on hourly spot price changes (€/MWh). The heat pump runs at full load when electricity prices are low and is switched off when prices are high. This scenario was tested for full load and partial load operation.

Demand Side Management
This scenario continues the concept of Demand Side Management (DSM) and combines it with the concept of price-dependent operation. In contrast to a fixed setpoint depending on the outdoor temperature, a variable setpoint is introduced for room heating in the building. This scenario is intended to analyze the heat capacity of buildings and consequently the savings potential through DSM strategies. The developed strategies are documented in the following sections.

4.2.1 Dynamic pricing

Demand response in the electricity sector refers to changes in the electrical loads of end consumers (in this specific case from a district heating network operator operating heat pumps) in response to changes in electricity prices or subsidies. Here, the simplicity of control, the so-called "Dynamic Pricing", as a strategy for an improved economy of a heat pump in a district heating network is investigated. The "Dynamic Pricing" concept considers the volatility and dynamics of the wholesale electricity market. The control strategy receives the hourly wholesale prices for electricity, as input to the controller, designated as €i, where i denotes the hours from 1 to 8760 (annual considerations). Based on this input, two limits, $€_{ON}$ and $€_{OFF}$, apply based on the historical frequency of occurrence of these limits. The control concept shown in Figure 42 is summarized below:

- $€_i < €_{ON}$ Heat pump ON (@Full load)
- $€_i > €_{OFF}$ Heat pump OFF
- $€_{ON} < €_i < €_{OFF}$ Heat pump ON/OFF/Part load (Depending on Storage charge status)

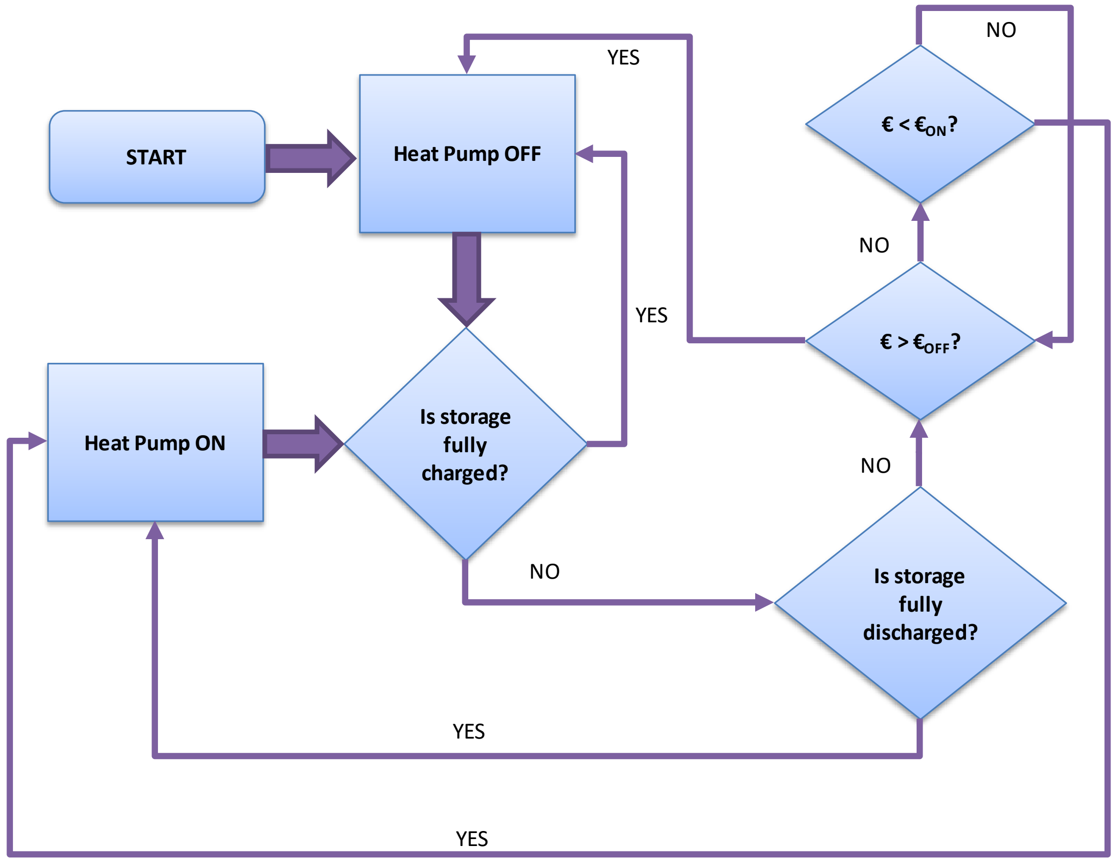


Figure 42 - Flowchart "Dynamic Pricing" Scenario

4.2.2 Demand side management

Together with Dynamic Pricing, which provides load management on the producer side (in cooperation with Task 3.1 on producer side measures), Demand Side Management (DSM) proposes control concepts on the consumer side, i.e. by influencing the customer systems. This considers two effects:

1) **Load shift**. This includes shifting most of the heating load at times when electricity prices are below the lower limit. This means that the room temperature setpoint is varied as follows:

$$€_i < €_{ON} \rightarrow T_{room-setpoint} = T_{room-setpoint} + 1°C$$
$$€_i > €_{OFF} \rightarrow T_{room-setpoint} = T_{room-setpoint} - 1°C$$

This strategy increases heating demand at low prices and consumption decreases during periods of high prices. This shifts the load towards low electricity prices.

2) **Building thermal mass**. This factor refers to the time it takes to heat and cool a building down to a certain temperature point depending on the outside temperature. This can be explained with the following two concepts:

   a) Building time constant: The building time constant can be represented as follows:

   $$\tau = \frac{c_m A_{cref}}{H_{tr} + H_{ve}}$$

   c_m Building Heat capacity (Wh/m²K)
   A_{cref} reference building area (m²)
   H_{tr} Heat transfer coefficient through transmission (W/K)
   H_{ve} Heat transfer coefficient through ventilation (W/K)
   This results in:

   b) Building Cooling/heating time: The time it takes to heat a building or cool it down to the temperature T_{in} with initial temperature Tin and outdoor temperature T_{out}:

   Based on the "Dynamic Pricing" control strategy, the DSM concept was developed (Figure 43): as soon as electricity prices are low, the load is heard (shifted by pre-heating the building). The cooling of the building is delayed by the time constant. In times of high prices, this can be used to operate the heat pump in partial load (or, if possible, to switch it off) and thereby reduce operating costs.

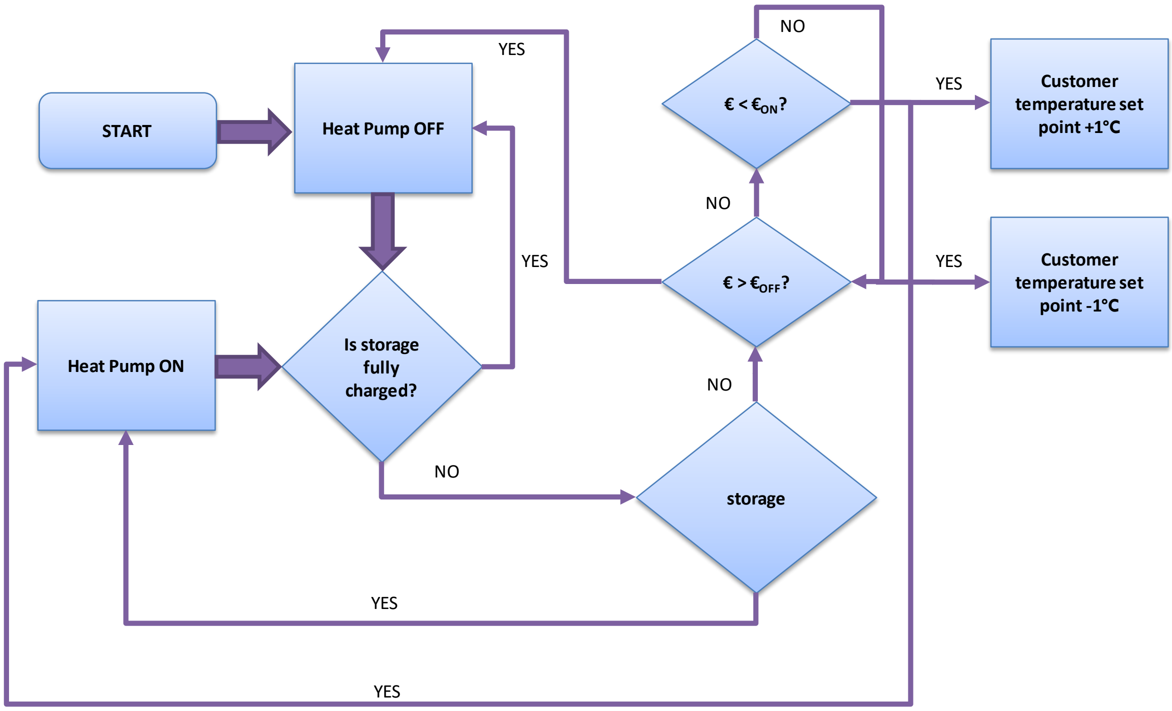


Figure 43 - Flowchart "Demand Side Management" Scenario

Figure 44 shows exemplary results of the control concepts "dynamic pricing" and "demand side management". Spot market prices and limits P_{ON} P_{OFF} are shown. The dashed blue line shows the normal heat consumption and the dark blue line shows the heat consumption when load shift is applied. The figure shows that the total load is shifted to the region where the price signal is below P_{ON}. With the help of the strategies described, the consumer and the producer can save operating costs compared to operation without load shift, while maintaining the same amount of heat.

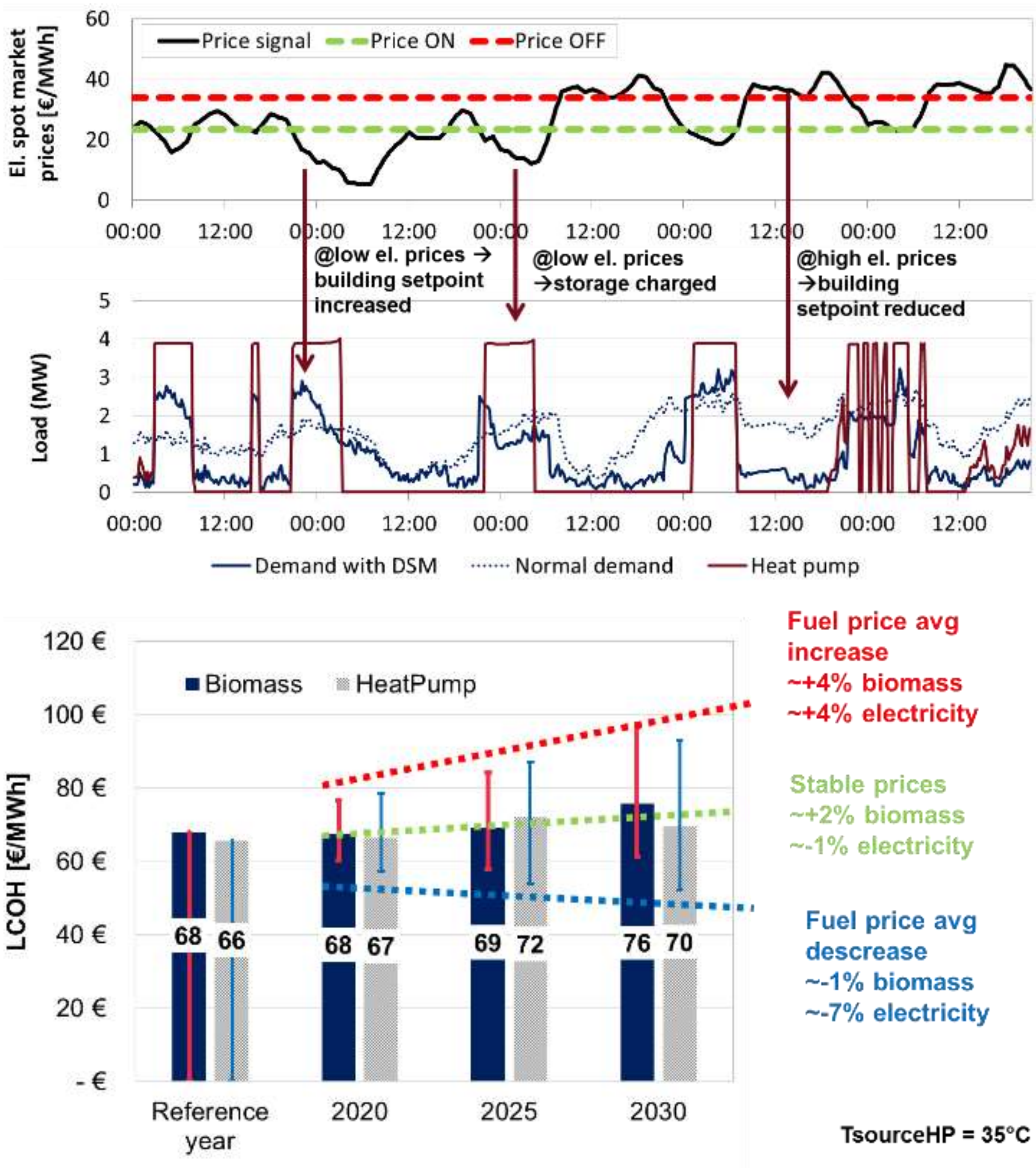


Figure 44 - Exemplary results of "dynamic pricing" and "demand side management" (up), heat generation costs for future scenarios (down)

4.3 Outlook: Participation in the balancing energy market

A new field of application for heat pumps in district heating networks could be the support of the power grids. The massive development of renewable electricity generation capacities in Austria has led to an installed capacity of 2,800 MW of wind and PV systems. These generate massive challenges in the power grids due to the stochastic production characteristics. Accordingly, flexibility options will increasingly be needed in the electricity markets such as the day-ahead or spot market, but also in the balancing energy market. On the other hand, the district heating market in Austria is confronted with many small and medium sized biomass plants. Many of these systems have reached the end of their technical life. Additional challenges are changing market conditions (in particular energy prices and falling demand for heat), which results in a reduced profitability of investments and an uncertain future outlook. Heat pumps can create a connection between the electricity and heat sectors, thus counteracting high costs for the expansion of electricity grids and at the same time increasing the efficiency of existing heating grids. Although technical solutions are available on the market and have already been successfully demonstrated, only very few examples have been realized in Austria. The aim of the just starting project fit4power2heat (FFG No. 861726) is to develop and evaluate innovative business models for the economic integration of heat pumps in small and medium-sized urban heating networks, regarding synergies from the heat and electricity market. The project includes the analysis of technical integration concepts, including the identification of suitable business models, based on the coupling of heat and power markets, e.g. by pooling several systems to participate in the balancing energy market, in the foreground. If applied successfully, this represents an essential component for significantly increase the system efficiency and the share of renewable energies in heat networks.

5. Conclusion

The research shows that, especially in the Scandinavian region, driven by an increasing share of fluctuating power sources such as PV and wind energy and at the same time decreasing electricity prices as well as the widespread use of heat networks, large heat pumps have been used in heating networks since the 1980s. Currently, Sweden is a forerunner using heat pumps in district heating and cooling networks. Approximately 7% of the district heating demand is produced by heat pumps. In other countries, e.g. Austria, the heat pump market consists mainly of devices for the supply of single and multi-family houses. Because of high system temperatures prevailing in a lot of heating networks, adapted concepts are needed in order to be able to guarantee the cost-effectiveness of the systems. Aim of current research projects such as fit4power2heat, it is therefore, to establish heat pumps by participating in various energy markets, as an attractive alternative. It has to be mentioned, that especially in the last years a lot of effort happened whole over European countries to foster heat pump integration in district heating and cooling networks.

Above all, the basis for economical operation is the correct design and hydraulic integration of the systems. As described, advantages can be achieved through different modes of operation. Instead of a monovalent operation, an additional generator for peak load times can save a large part of the investment costs.

Furthermore, different circuit options can be used in order to achieve the optimum for the operation of the system. Depending on which framework conditions exist, it is possible to exploit considerable potentials in terms of efficiency and therefore also in terms of costs. The correct design of the heat source system and the heat sink play as much a role as the dimensioning of the heat pump itself.

As a first clue, the AIT internally developed Excel based tool that can be used to pre-estimate feasibility and cost-effectiveness. With the help of simple calculations and against the background of already realized plants, first conclusions can be drawn thereby. The more detailed information about the planned project, the more accurate the initial assessment can be. Through the conversion into Excel by means of VBA and the database integrated in the tool, as well as the user interface, the calculations can be carried out relatively easily and without special software prior knowledge. The quick and easy adaptation of the underlying database is therefore also guaranteed.

In addition to the current-driven compression heat pumps, occasionally also thermally operated heat pumps are used. Depending on the field of application, the advantages of the different technologies can be used.

With reference to the results achieved by the mentioned investigations, the importance and contribution of heat pumps in district heating networks were pointed out. In addition, recommendations for "best practice" strategies for the operation of heat pumps in combination with a central storage unit are presented:

- Heat pumps with dynamic pricing and demand side management´ are more resilient to market risks, as dynamic operation counteracts fluctuations in fuel and electricity prices.
- Savings in heat generation costs of up to 9% with the "dynamic pricing" application and 11% to 15% possible with DSM in residential blocks or in the overall network
- Heat pumps increase the flexibility of district heating systems by expanding the heat generation portfolio, which enables higher reactivity through fast commissioning and low start-up costs and takes advantage of the volatility of the electricity market and thermal batteries.
- Heat pumps can be used to increase renewable heat generation. In addition, low-temperature heat sources and alternative heat sources can be used.

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