Challenges and potentials for low-temperature district heating implementation in Norway

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Abstract

Current district heating (DH) systems with high temperatures are facing many challenges that may decrease its competitiveness. Some of the challenges are decreased heat demands due to energy efficient buildings and high return temperatures that decrease possibilities for utilization of renewable heat sources. Low temperature DH (LTDH) systems have opportunities for utilization of waste heat and renewables and lower distribution losses. Therefore, the aims of the study were to analyze the challenges in the transition to LTDH and to estimate the increased competitiveness in low heat density areas. Since the heating density is an important factor for the DH competitiveness, the high and the low heat density area were analyzed. A building area consisting of the passive house and low energy buildings in Trondheim, Norway, was analyzed. The hourly DH network model was developed included both thermal and pressure losses. The results showed that the heat loss could be reduced by lowering the supply temperature from 80°C to 55°C. Analysis of the return temperature showed that LTDH could provide a lower return temperature than the existing DH system, regardless of the faults. Competitiveness of LTDH might be decreased for the heat densities lower than 1 MWh/m.

Keywords: Low-temperature district heating, Low energy buildings, Heat density, Distribution losses

Nomenclature

L $[m]$ – pipe length

L_{tot} $[m]$ – the total pipe length of the supply pipeline

R $[Pa/m]$ – pressure drop per pipe length

$T_{in,i}$ $[°C]$ – temperature at inlet of pipe

$T_{out,i}$ $[°C]$ – temperature at pipe outlet

T_1 $[°C]$ – supply temperature

T_2 $[°C]$ – return temperature

T_g $[°C]$ – ground temperature

T_s $[°C]$ – supply temperature to the radiator

T_r $[°C]$ – return temperature out of the radiator

$T_{r,s}$ $[°C]$ – return temperature in the secondary side of heat exchanger

$T_{s,p}$ $[°C]$ – supply temperature in the primary side

$T_{r,p}$ $[°C]$ – return temperatures in the primary side

$T_{s,d}$ $[°C]$ – design supply temperature

$T_{r,d}$ $[°C]$ – design return temperature

T_i $[°C]$ – indoor room temperature

T_m $[K]$ – mean temperature of the supply and return temperature

ΔT_m $[K]$ – mean arithmetic temperature difference

H_p $[Pa]$ – pressure rise over the circulation pump

$U_i\ [W/mK]$ – overall heat loss coefficient

$U_{11}\ [W/mK]$ – heat loss coefficient in the supply pipe without thermal influence of return pipeline

$U_{22}\ [W/mK]$ – heat loss coefficient in the return pipe without thermal influence of supply pipeline

$U_{12}\ [W/mK]$ – heat loss coefficient due to thermal influence of return pipeline

$U_{21}\ [W/mK]$ – heat loss coefficient due to thermal influence of supply pipeline

$p_1\ [Pa]$ – minimum permitted pressure in a DH network

$\Delta p_{cs}\ [Pa]$ – pressure drop over the customer substation

$\Delta p_{dh}\ [Pa]$ – pressure drop in the heat exchanger delivering the heat to the observed area

$\Delta p_t\ [Pa]$ – increase in pumping pressure

$\Delta p_{tot}\ [Pa]$ – total pressure drop

$\dot{m}\ [kg/s]$ – mass flow rate

$\dot{Q}_{hd}\ [W]$ – heat demand

$\dot{Q}_{hd}\ [W]$ – actual heat demand

$\dot{Q}_{d,hd}[W]$ – design heat demand

$\dot{V}_t\ [m^3/s]$ – total volume flow rate

$c_p[kJ/kgK]$ – specific heat capacity of water

$d_i\ [m]$ – internal pipe diameter

$f\ [-]$ – friction coefficient

$n1\ [-]$ – radiator exponent

$\rho\ [kg/m^3]$ – water density

$\varepsilon\ [-]$ – temperature efficiency of heat exchanger

η $[-]$ – pump efficiency

Abbreviations

cs – Consumer substation

DH – District heating

DHC – District heating and cooling

DHW – Domestic hot water

LTDH – Low temperature district heating

1. Introduction

Use of renewable energies and waste energy is highly necessary and required by national and international regulations [1]. Future district heating and cooling (DHC) systems may enable transition to a complete renewable society [2], meaning that the future DHC systems will be based on completely renewable energies such as solar, waste heat, and geothermal energy. To enable higher share of renewables in the DHC system, the temperature of the district heating system should be lowered [3]. Throughout the historical development of the district heating (DH) system, the system development has passed through four generations [4]. At the beginning, the DH temperature was decided based on energy supply technologies that were providing high temperature water. Consequently, the DH temperature was decreased due to implementation of heat pumps and a general idea to increase the system efficiency by decreasing the DH temperature [5]. Lately, due to lower heat demand in new passive and low energy buildings, the supply temperature could be decreased to 45 - 55°C [5]. The DH system operating with these temperature levels is called low temperature DH (LTDH) [3, 6]. There is no a clear limit for LTDH, but the temperature levels should be within the given range.

Current DH systems belonging to the second and third generation with the temperature of 80 to 100°C are facing many challenges [4]. In low energy buildings, there is no need for heating at high temperature levels and heating demand is decreasing. High temperature levels are unfavorable in terms of utilizing renewable energy sources and waste heat. Finally, due to higher share of the distribution losses in the total DH heat demand, competitiveness of the DH system in the low heat density area is decreasing [7]. In order to be competitive in the areas with low heat densities and low energy buildings, it is important to achieve low heat losses for a high efficiency of the DH system. In general, the heat losses in Norway are in the range of 8-15% of the delivered heat [8, 9]. The merit with the low supply temperature is that the temperature difference between the pipe and the ground is less than in the case with the high supply temperature. This facilitates that the heat losses to the ground are less and the demand for insulation can be reduced for certain DH areas. LTDH systems have better opportunities for utilization of waste heat and renewable heat sources, as well as lower distribution losses. However, on the way to the transition to LTDH, there is a problem with high return temperature and the low temperature difference between the supply and the return temperature in the network [10].

Until now, several LTDH projects have been successfully realized. Some conclusions and the most important characteristics are explained. Seven low energy apartment buildings in Lystrup, Denmark, have been connected to LTDH in an attempt to reduce distribution losses [11]. This is done by reducing pipeline dimensions, setting the distribution temperature to 55°C, and using twin pipes. In addition, booster pumps that raise the pressure in the area enables further reduction of the pipeline dimensions. In the mentioned project, two consumer substation connection types with LTDH are implemented: 1) with storage tanks and 2) with high heat output heat exchangers [11]. The use of DH storage tanks makes it possible to reduce the pipeline dimension as it reduces the peak demand. The total costs and benefits of these two alternatives for the LTDH connection are roughly the same, and both are viable solutions. The result is a reduction of energy use of 75 % compared to the traditional DH systems [11]. In Albertslund, Denmark, LTDH has been introduced for 1544 refurbished houses from the 1960's. The distribution network for DH was replaced by twin piping laid in shorter routes, and the houses were completely renovated to a standard close to today's low energy regulations. The houses were fitted with individual heat exchangers for instant heating of domestic hot water (DHW). This design requires a higher peak heat rate from the DH network, but eliminates the need for storage tanks. It is expected that this solution will result in a 62 % reduction in distribution heat losses. This is achieved at an extra cost of 20 million DKK and will result in a profit of 31 million DKK over the project lifetime of 50 years [12]. In Chalvey, England, a small scale LTDH network has been constructed, supplying ten zero emission houses. The houses are equipped with photovoltaics to cover part of their electricity demand, while DH covers the heat demand. Each house is equipped with a heat exchanger for DHW. Heat is produced using biomass, air to water heat pump, two ground-source heat pumps, and 20 m^2 of solar collectors. The DH central contains a large storage tank. The heat in the storage tank is used to cover peak load. Due to possibility to charge the storage tank by any of the mentioned energy supply technology, the flexibility of the system is increased. Heat production is optimized for a low temperature system of 55°C and LTDH is completely based on renewable energies [13]. In all these examples, the introduction of LTDH has produced good results when it comes to distribution loss savings, low temperature heat production, and customer satisfaction.

Since the issue on the distribution losses is relevant for the future development of LTDH [4], it is highly important to model properly the distribution system with belonging issues. There have been different studies dealing with steady-state analysis of new concepts for the DH distribution systems [14], detail pressure drop models for the DH system [15], the dynamic heat loss model [16], and pipe network models based on producer data [17]. The mentioned studies have not treated heat dynamics due to annual variation in heat demand and flow and pressure control at the same time. Regardless of the topic importance and future development of the smart DH, the studies on the prosumers in the DH [18, 19] have not provided a general method to be used for the performance analysis, design, and operation optimization of the DH system including distributed heat sources. In our study, the model for the pressure and thermal losses with the hourly heat load input was implemented to treat in a proper way relevant issues in the DH system.

In Norway, the new buildings are being built to high standards with reduced space heating demands, and thereby the demand density in the DH network is decreasing. This will induce that the percentage of distribution heat losses is increasing [7]. However, there are still a high percentage of the existing buildings requiring higher temperature and heat demand. The DH system is not a system where the technology and parameters may be changed at once [4], yet there is transition to come to the defined aims [3]. This means that the DH system is under continuous development. For example, once installed pipes will be used as long as they are not damaged regardless of increased heat demand [8]. Therefore, this study by analysing a DH system in Trondheim, Norway, had two aims. The first aim of the study was to estimate the challenges in the transition process from the current DH to LTDH systems, while integrating low energy and passive house buildings. The second aim was to estimate possibilities and increased competitiveness of LTDH in the low heat density areas.

The rest of the paper is organized as the following. The methodology is introduced in Section 2. The analyzed areas and issues in decreasing the DH temperature are introduced in Section 3. The results are presented in Section 4. The result section is firstly introducing the heat demand and temperature distribution profiles. Based on these, the results on the energy performance of the analyzed LTDH system together with the issues in the operation and LTDH competitiveness are provided. In Section 5, discussion and criticism on the provided results are given. Finally, the conclusions are given in Section 6.

2. Methodology

The methodology to model the DH network included the network model and the consumer substation model. This is a development of the work started in [20]. The network model consisted of two parts: thermal heat losses and pressure drops. The thermal heat loss model was necessary to explain the temperature and heat losses. The pressure loss model was necessary to estimate the energy need for the DH water transportation and the pressure level in the grid. For both thermal and pressure loss model, mass flow rate was an input. To provide a correct mass flow rate, the substation model was developed to provide the correct temperature levels. All the consisting models were developed on hourly basis. This means that the model needed an hourly input on the heat load.

2.1. DH thermal network model

To be able to keep a sufficient supply temperature to the last customer substation in the DH system, it is important to have reliable calculations on the temperature drop in the distribution network. The temperature drop in the distribution system is dependent on several parameters: temperature levels in the DH system, ground temperatures, the pipe length, the mass flow rate, and the heat loss coefficient [21]. To develop a general model for the thermal losses a part of the pipe with one inlet and one outlet node was observed. Equation (1) gives the outlet temperature of the pipe as:

$$T_{out,i} = \begin{cases} T_g + \left(T_{in,i} - T_g\right) exp\left(-\frac{U_i L_i}{\dot{m}_i c_p}\right) \\ T_g \end{cases} \tag{1}$$

where, $T_{in,i}$ is the entering temperature to the pipe, $T_{out,i}$ is the outlet temperature of the pipe, T_g is the ground temperature, U_i is the overall heat loss coefficient, L_i is the pipe length, $\dot{m}_i$ is the mass flow rate, c_p is the specific heat capacity of water. The temperature drop in the supply line is always larger than in the return line. The typical supply temperature drop for winter heat load is about 1-2 K being the difference between the supply temperature at the heat supply units and that at the average substation. The corresponding temperature drop during the summer can be in the range of 5-10 K [4]. In twin-pipes, the heat transfer between the pipes may lead to increase of the return temperature by 2 K [22].

The twin-pipe is a pipe construction where two pipes are located within a common circular insulation within an outer casing [4]. It is questionable whether the twin-pipe could facilitate the increase in the return temperature, since the pipes are located in the same coinciding temperature field. The authors in [23] state that if the return temperature drops below a predefined temperature level, the effect of the coincident temperature field will facilitate the increase of the return temperature rather than the heat losses to the surrounding. In our study, the heat loss per length of the supply pipe was calculated as:

$$q_1 = U_{11}\left(T_1 - T_g\right) - U_{12}\left(T_2 - T_g\right) \tag{2}$$

where U_{11} is the heat loss coefficient in the supply pipe without the thermal influence of the return pipeline and U_{12} is the heat loss coefficient due to thermal influence of the return pipeline, T_1 is the supply temperature, and T_2 is the return temperature.

The heat loss per length of the return pipe was calculated as:

$$q_2 = U_{22}\left(T_2 - T_g\right) - U_{21}\left(T_1 - T_g\right) \tag{3}$$

where U_{22} is the heat loss coefficient in the return pipeline without the thermal influence of the supply pipeline and U_{21} is the heat loss coefficient due to the thermal influence of the supply pipeline.

In the case of twin-pipe, the pipes are identical and placed horizontally in relation to each other. This means that $U_{12} = U_{21}$ and $U_{11} = U_{22}$. This provides Equation (4) for the heat losses of the twin-pipe:

$$q_{tot} = q_1 + q_2 = 2\,(U_{11} - U_{12}) \cdot \left(T_m - T_g\right) \qquad (4)$$

The overall heat loss coefficients, U_{11} and U_{12}, were calculated based on Wallenten's equation [24]. In Equation 4, T_m is the mean temperature of the supply and the return temperature.

2.2. Pressure drop and pumping power

The total pressure drop in the DH system can be calculated as:

$$\Delta p_{tot} = 2 \cdot R \cdot L_{tot} + \Delta p_{cs} + \Delta p_{dh} \qquad (5)$$

where $2 \cdot R \cdot L_{tot}$is the pressure drop due to friction in pipes considering the entire pipeline, Δp_{cs} is the pressure drop over the customer substation, and Δp_{dh} in the pressure drop in the heat exchanger delivering the heat to the observed area.

The pressure drop due to friction can be found by employing Darcy-Weisbach equation:

$$R = \frac{\Delta p_f}{L} = \frac{8 \cdot f}{d_i^5 \cdot \pi^2 \cdot \rho}\;\; \dot{m}^2 \qquad (6)$$

where f is the friction coefficient of the pipe, L is the observed pipe length, d_i is the pipe diameter, ρ is the water density, and $\dot{m}$ is the mass flow rate. In general, the determination of the friction losses requires complex simulations based on laminar and turbulent flows. In this paper, the simplification was made in terms of employment of constant friction value. The friction factor for DH pipes had value in the range of 0.015 and 0.04 [4]. Therefore, for calculation purposes the value was chosen equal to 0.025.

In order to calculate the statistic pressure in the DH system in a predefined location marked x, the following equation could be used:

$$p_x = p_1 + H_p - R \cdot L_x - \Delta p_{cs} \tag{7}$$

where H_p is the pressure rise over the circulation pump, and L_x is the pipe length till the observed place marked x. p_1 is the minimum permitted pressure in a DH network. In Equation (7), the part Δp_{cs} (pressure drop over the consumer substation) was included if the pressure level was estimated after the substation. In the case when the pressure level was estimated only in the supply pipe, this term was not necessary.

The pump was operated based on maximum pressure difference in the DH system for delivering heat to the last customer in the system. Finally, by combining the results on the total pressure difference for the entire system from Equation (5), the pumping power was calculated as:

$$P = \frac{\Delta p_{tot} \cdot \dot{V}_t}{\eta} \tag{8}$$

where $\dot{V}_t$ is the total volume flow and η is the pump efficiency. According to [25] the pumping efficiency is in the range of 0.8 - 0.9. For the simulation model, the constant efficiency of 0.85 was chosen.

2.3. Customer substation model

A customer substation model was developed in order to analyze how customers itself and operation of the substation could affect the return temperature in the DH system. In addition, it was necessary to develop a model due to unknown return temperature and mass flow rate in the primary loop, which is the result of the operation of the customer substation. The substation analyzed in this study was an indirect connection to the DH system with parallel-connected heat exchangers for space heating and DHW. Fig. 1 shows the layout of customer substation with the necessary flows and temperatures used for the model development. Description of all the temperatures and flows marked the substation sketch in Fig. 1 is given in Table 1.

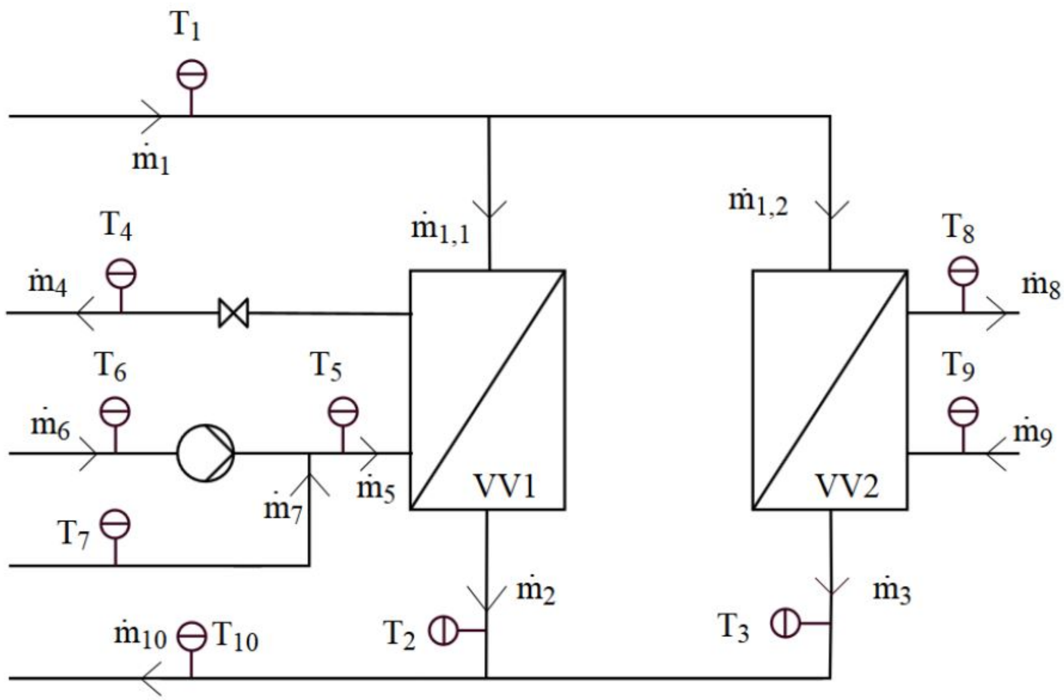

Fig. 1. Layout of customer substation

Table 1. Overview of the flows and temperatures in the consumer substation in Fig. 1

Variable	**Description**
T_1 and $\dot{m}_1$	The supply temperature and mass flow rate in the primary loop
$\dot{m}_{1,1}$	The mass flow rate in the primary loop for the DHW
$\dot{m}_{1,2}$	The mass flow rate in the from primary loop for the space heating
T_2 and $\dot{m}_2$	The return temperature and mass flow rate from the DHW heat exchanger
T_3 and $\dot{m}_3$	The return temperature and mass flow rate from heat exchanger to the space heating at the primary side
T_4 and $\dot{m}_4$	The supply temperature and mass flow rate for the DHW use
T_5 and $\dot{m}_5$	The return temperature to DHW heat exchanger at the secondary side
T_6 and $\dot{m}_6$	The temperature and mass flow rate in the DHW circulation
T_7 and $\dot{m}_7$	The supply cold water temperature and mass flow rate
T_8 and $\dot{m}_8$	The supply temperature and mass flow rate to the space heating system
T_9 and $\dot{m}_9$	The return temperature and mass flow rate from the space heating system
T_{10} and $\dot{m}_{10}$	The return temperature and mass flow rate to the primary loop

In Fig. 1, the heat exchanger for heating the DHW is marked with VV1, while VV2 corresponds to heat exchanger for the space heating. At the primary side of the customer substation the only known values are the supply temperature T_1 and the heat demand $\dot{Q}_{hd}$.

However, the challenging part is defining the return temperature and mass flow rate from customer substation including both the space heating system and the DHW system. The text below describes the calculation method for the return temperature in the DH system.

The return temperature at the primary side of heat exchanger was calculated as:

$$T_{r,p} = T_{s,p} - \varepsilon \cdot \left(T_{s,p} - T_{r,s}\right) \quad (9)$$

where $T_{s,p}$ and $T_{r,p}$ are the supply and return temperatures in the primary side, $T_{r,s}$ is the return temperature at the secondary side of the heat exchanger, ε is the temperature efficiency of the heat exchanger [26]. The efficiency of the heat exchanger may influence the return temperature, too. A sensitivity analysis to evaluate the influence of the heat exchanger efficiency on the return temperature was also performed.

The supply temperature at the primary side was estimated based on the outdoor temperature compensation. In practice, both the primary and the secondary side supply temperature are compensated. Fig. 2 shows outdoor temperature compensated curves that were used for the primary supply temperature and for the radiator heating system. These temperatures were necessary as input for the model. Further, the design supply and return temperatures for the heating system depending on temperatures in the DH grid are given in Table 2.

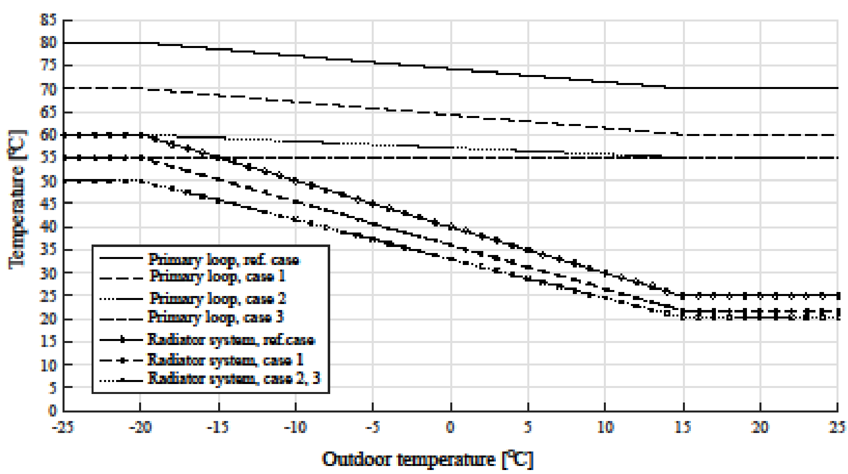

Fig. 2. Outdoor temperature compensation for supply temperature in the primary loop of DH system

Table 2. The design supply and return temperatures in the heating system depending on various DH temperatures

Supply temperature in the DH system	**80°C**	**70°C**	**60°C**	**55°C**
T_s/T_r	60/40°C	55/30°C	50/25°C	50/25°C
ΔT	20 K	25 K	25 K	25 K

The temperatures shown in Table 2 were selected based on technical considerations of the DH heat provider, which stated that the design supply and the return temperatures should have level of 60/40°C or lower [27]. The temperature in the radiator heating system would decrease with decrease of the supply temperature in the DH system. However, the previous studies showed that the supply temperature could be as low as 55°C without causing problems with the indoor comfort level. The high temperature difference between the supply and the return provides low mass flow rate that could lead to the issues with the control valves. Therefore, this issue was considered while modeling.

In order to size the heat exchangers for the heating system, the value of the return temperature from radiators was necessary. Therefore, the solution can be found by combining several equations. The radiator characteristic can be expressed as:

$$\frac{\dot{Q}_{hd}}{\dot{Q}_{d,hd}} = \left(\frac{\Delta T_m}{\Delta T_{m,d}}\right)^{n1} \tag{10}$$

where ΔT_m is the mean arithmetic temperature difference, $\dot{Q}_{hd}$ is the current heat demand, $\dot{Q}_{d,hd}$ is the design heat demand, and $n1$ is the radiator exponent.

By solving Equation (10) for the return temperature, the return temperature in the radiator could be expressed as:

$$T_r = 2 \cdot \left(\left(\frac{\dot{Q}_{hd}}{\dot{Q}_{d,hd}}\right)^{\frac{1}{n1}} \cdot \left(\frac{T_{s,d}+T_{r,d}}{2} - T_i\right) + T_i\right) + T_s \tag{11}$$

where T_s and T_r are the supply and return temperatures of the radiator, $T_{s,d}$ and $T_{r,d}$ are the design supply and return temperatures, and T_i is the room temperature. The room temperature in this study was set to be the constant value of 21°C and the radiator exponent equal to 1.3. It can be argued what may be the indoor room temperature. Therefore, analysis how the value of the indoor temperature could influence the return temperature was also performed.

When the return temperature from each substation were calculated, it was possible to calculate the final return temperature that was necessary to evaluate the heat losses in Equation (4). Further, the return temperature was also used to calculate the mass flow rate that was necessary for the pressure drop calculation.

3. Description of the low and high heat density area

The two heating networks, given in Fig. 3 were introduced to estimate the performance and future competitiveness of LTDH. The accurate selection of the network structure is essential for achieving an energy efficient and profitable low temperature heating grid resulting in low specific heat losses. The linear density is a parameter that is used to define competitiveness of a DH network compared to alternative energy supply methods. In addition, it shows how much heat is delivered per meter of the pipe length [28]. At the same time, the competitiveness is dependent on the local topology and the situation on the energy market creating various profitability threshold, for example, 0.2 MWh/m in Denmark [29] and 1.5 MWh/m in Canada [30]. The heating network A represents the case with the low heat density and is characterized by the linear density of 1.3 MWh/m. The high heat density with the linear density of 2.3 MWh/m characterized the heating network B. The linear heat density may have different values, depending on DH network development and building heat demand. For example, for small house areas it may be up to 1 MWh/m, while for small DH networks is it up to 5 MWh/m [28]. In this study, the heat density values for the low and high heat density areas were defined based on the values in [28-30].

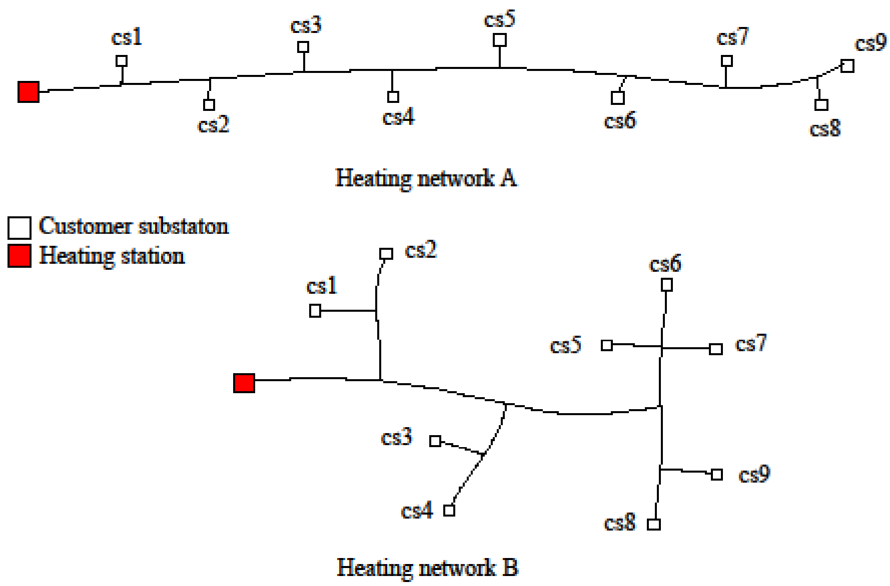

Fig. 3. Structure of the heating networks A and B

For the areas with low linear heat densities, the main challenge is in a network structure that has direct impact on profitability of the DH system. Therefore, in this study two different network structures with the different linear heat densities were analyzed. In this study, the maximum length of heating network A to the customer substation was limited to 1230 m, while for the heating network B this value was 510 m. These two lengths were chosen to be able to define two different linear heat density areas. It was decided to look at these areas from a perspective of different building types and customers that could be operated by the DH company. The information on customer types were based on the real measurement data collected for Trondheim in 2013 obtained from the DH company, Statkraft Varme AS. The customers analyzed in this study were the following: one building block built under low energy building standard, TEK10 [31], three passive house standard building blocks [32], a primary school with sport center, a kindergarten, a health and welfare center, and an office building with low energy standard, see Table 3.

Table 3. Specification of the customers

Customers	**Gross area (m^2)**	**Construction year**	**Building standard**	**Number of apartments**	**Substation number**
Building block A	2380	2011-2012	Passive house standard	26	cs1
Building block B	2160	2011-2012		25	cs2
Building block C	4750	2011-2012	TEK10	50	cs3
Building block D	1480	2011-2012	Passive house standard	13	cs4
Primary school	6900	2008-2009	Low energy building standard		cs5
Sport center	2724	2008-2009			cs6
Kindergarten	2000	2011			cs7
Health and welfare building	2696	2011		64 rooms	cs8
Office building	8600	2010-2011			cs9

Each building was equipped with a substation providing hydronic space heating and the DHW as given in Fig. 1. The designed heat demand for the substations and the pipes was chosen to be 20 % higher than the heat load in 2013 in order to cover increase in heating demand if necessary. It was assumed that the radiator was used in each room and could cover all the needs for heating. Based on available data these buildings showed low heating demand, hence, it was concluded that they could be connected to LTDH without significant changes in the network structure. LT-DH could be implemented either in existing heating networks or via development of a new DH system. Hence, it was decided to develop a network models for A and B systems and compare the results with the existing DH system. Due to different design requirements for those cases and the reduced temperature difference for LTDH, the analysis aimed to find a solution for the transition of the existing DH systems to more efficient LTDH.

3.1. Development of the DH system

For the purpose of the study, it was decided to model both the existing and the new DH systems. The reason for this was that the design conditions for the pipes are different due to different temperature differences. Information flow how development of the DH system may be made for the low energy buildings is shown on Fig. 4. The information flow chart in Fig. 4 was developed in collaboration with the DH company and was implemented to test different DH grid development scenarios.

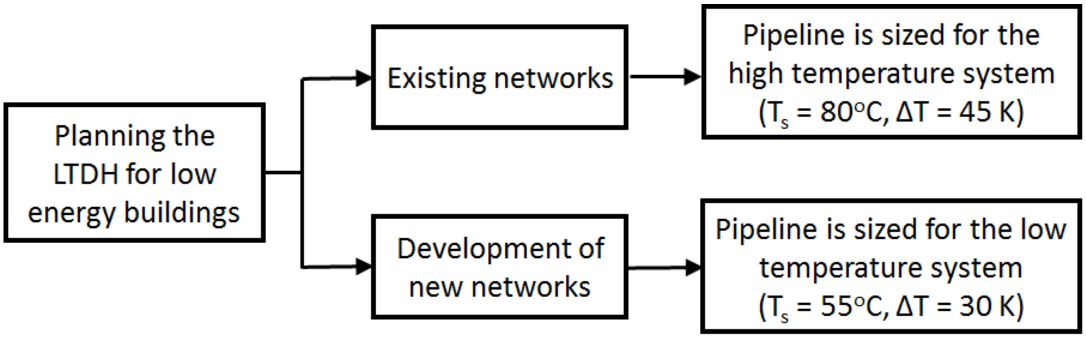

Fig. 4. Planning of LTDH for the low energy buildings

The design of the existing DH system was based on Technical specifications for the DH in Trondheim [27]. In this specification, it is defined that the DH system consists of primary and secondary loop, which are hydraulically separated by heat exchangers. For the existing DH network, the consumer substations and the DH network were designed based on the following:

- Supply water temperature at winter (at design outdoor temperature): 80°C;
- Supply water temperature at summer: 70°C;
- Design temperature difference: 45 K
- Criteria for R-value (at design outdoor temperature): 50-250 Pa/m;
- Pipe type: twin-pipe in steel casing;
- Maximum water velocity: 2 m/s.

Based on the discussion with the DH company, it was assumed that the transition from the high temperature to LTDH would be accomplished with gradual reduction in the supply temperature from year to year without pipe changes. This was assumed because there are still many existing buildings in the system with the high temperature requirement These existing buildings would undergo some improvements in the future and consequently be capable to use lower supply temperature [3, 33]. This means that the existing network design for higher temperature levels would become a LTDH with lower temperatures and the same pipe dimensions as designed at the beginning. Therefore, the simulation models included several scenarios for the temperature reduction. The summary on the temperature levels is given in Table 4. Further, two of them have been chosen for deeper analysis. The reference case shows the temperature levels in the traditional DH system, while Case 3 shows temperature levels for LTDH.

Table 4. Supply temperatures in the DH system and radiator heating system

Scenarios	**Primary loop**		**Radiator heating system**	
	Winter	Summer	Winter	Summer
Reference case	80°C	70°C	60°C	25°C
Case 1	70°C	60°C	55°C	22°C
Case 2	60°C	55°C	50°C	20°C
Case 3	55°C	55°C	50°C	20°C

The development of the new DH network implies that the designed pipes have to be able to satisfy customer demand with the supply temperature of 55°C, with the designed temperature difference of 30 K. Further, for a better energy efficiency and profitability, it is important that heat losses remained low. Therefore, the plastic pipes with diameter of 32 mm and smaller with good insulation characteristics were implemented. The decision about technical parameters of LTDH was based on several demonstration projects developed in Denmark. Sizing of the new LTDH network was based on the following:

- Supply temperature: 55°C;
- Design temperature difference: 30 K;
- Pipe type: twin-pipe in steel casing and twin-pipe in Aluflextra material;
- Maximum water velocity: 2 m/s.

For the new development of the DH network, it is important to achieve good performance and decrease the heat losses. This is achieved with improved insulation and smaller pipe diameters. The decrease in the pipe diameters will lead to change in the pressure drop in the system. Therefore, the analysis included various specific pressure drop values in the range of 200 and 800 Pa/m for development of the new heating network. The summary of the scenarios for sizing a new DH network is given in Table 5.

Table 5. Pressure drop constrains for the new development

Scenarios	**Δp1**	**Δp2**	**Δp3**
Main lines	R ≤ 150 Pa/m	R ≤ 300 Pa/m	R ≤ 600 Pa/m
Service lines	R ≤ 250 Pa/m	R ≤ 550 Pa/m	R ≤ 800 Pa/m

To calculate the pressure drop, pressure level, and the pump power, it was necessary to define certain limits for the calculation. Necessary parameters for calculation of the model introduced in 2.2 are summed-up in Table 6 based on practical constraints and values given by the DH company [27].

Table 6. Parameters for the presser drop and level calculation

Parameter	Value
Differential pressure over customer substation	$\Delta p_{ab} = 0.7\ bar$
Differential pressure over DH plant/main heat exchanger	$\Delta p_{ab} = 1\ bar$
Minimum permitted statistic pressure	$p_1 = 1.5\ bar$
Maximum permitted statistic pressure	$p_{max} = 25\ bar$
Maximum pressure drop in the heating system	$\Delta p_{max} = 8\ bar$

The minimum static pressure p_1 should be kept above the saturation pressure in order to avoid boiling and cavitation in the pipe. The saturation pressure is lower than 1 bar for the temperatures below 100°C. For security margin, the pressure could be increased up to 5 bar [34], however in this study the value was chosen equal to 1.5 bar due to the small DH system.

3.2. Issues with the high return temperature

LTDH is a paramount of the DH technology that should be achieved. The high return temperature from the customer substation is considered as one of the main issues for decreasing the supply temperature. Different type of errors causing the faults in the return temperature are identified such as: system design, heat exchangers, control, and errors outside of the substation [4, 10]. Based on the literature review and the discussion with the DH company, the following errors were introduced in the models: short bypasses, aging and fouling of the heat exchangers, indoor temperature set point errors, and fail adjustment of the outdoor compensation curve. Introduction of each error in the model is explained in brief below.

Bypass - The inspection of the DH system indicates that the bypass valves may be installed intentionally or unintentionally in the primary loop at the customer substation. Sometimes they may stay open due to fails or neglecting. Further, bypasses or some additional pipes may be installed sometimes in the system with some purpose, but this has been forgotten over time. All these may influence a short circulation, i.e. that the supply water is mixed directly with the return water and thereby the return water temperature is increased. In this study, four various situations when the water bypasses from the supply to the return line were tested, 1%, 2%, 5%, and 10% of the flow might bypass.

Aging and fouling of the heat exchangers - Over the time, fouling of the heat exchangers appears introducing the decrease of the heat exchanger efficiency. This is a minor throughput that causes high return temperatures in customer centers. However, district heating companies are experiencing an increasing amount of leaks in heat exchangers. This problem can lead to high replacement costs of the heat exchanger. It was chosen to briefly investigate how temperature efficiency can affect the return temperature, if there is a problem with the heat exchanger. Therefore, the different efficiencies of heat exchangers were tested by changing the value in Equation (9). The temperature efficiency was set to 0.85 as a reference for both heat exchangers in Fig. 1. To introduce aging and fouling the temperature efficiency was decrease from 0.85 to 0.6 [26]. In the analysis, when adjusting the temperature efficiency of one heat exchanger, the temperature efficiency of the other was kept at 0.85.

The error in the indoor set-point temperature - It is known that building occupants prefer to adjust the indoor temperature level based on their preferences rather than on design. Therefore, different set-point indoor temperatures were examined to see the change in the DH return temperature due to different settings. The different set-points were tested by changing the value of the indoor temperature in Equation (11). The results have been compared to the reference value of 21°C, which is given as a requirement in the national standard NS 3031 [35].

The error in the adjustment of the outdoor compensation curve - Adjustment of the outdoor compensation curve may lead to the change in the return temperature [36, 37]. For the analysis of this issue, three compensation curves were suggested.

4. Results

This section starts with the presentation of the heat demand of the consumers. Based on the introduced heat demand and the DH network model, the results on the temperature level are introduced. As indicated in the methodology, after the operation parameters of the DH grid were defined, general DH grid performance data, pump energy use and heat losses could be estimated. Effects of the introduced errors are introduced afterwards. Finally, results on the competitiveness of LTDH in the low heat density area are given.

4.1. Heating demand

Hourly heating demand data for different customers were obtained from Statkraft Varme AS from the direct measurement at the customers. The heating load data for 2013 are summarized in Table 7. A coincidence factor given in Table 7 is a dimensionless factor explaining that all the maximum heat loads from different users or buildings do not appear at the same time. The value of the coincidence factor is between 0 and 1, and usually lower than 1. It is important to consider coincidence factor for sizing and energy planning to avoid oversizing [38] In building blocks with different occupants and different life habits, heat use is different. Therefore, the values of the coincidence factor for the building block are lower than in the other buildings in Table 7.

Table 7. Specification of energy use for different customers

Consumers	Substation number	Maximum heat demand [kW]	Max. specific heat demand [W/m^2]	Annual heat demand [kWh]	Specific annual heat demand [kWh/m^2]	Utilization time	Coincidence factor [-]
Building block A	cs1	62	26	139 114	58	2 244	0.63
Building block B	cs2	40	19	121 136	56	3 028	0.70
Building block C	cs3	116	24	372 698	78	3 213	0.71
Building block D	cs4	64	43	116 068	112	2 595	0.84
Primary school	cs5	208	30	263 101	38	1 265	0.84
Sport center	cs6	114	42	178 006	65	1 561	0.89
Kindergarten	cs7	74	37	135 431	68	1 830	0.61
Health and welfare building	cs8	124	22	366 819	64	2 958	0.98
Office building	cs9	192	22	158 698	18	827	0.84

From Table 7, it can be seen that Building block D showed the highest maximum specific heat demand and the highest specific annual heat demand in comparison to all the buildings in spite of the fact that the building was constructed under the passive house standard. The reason for this could be explained by diverse occupancy patterns or poor operation of the customer substation. Utilization time in Table 7 describes how long the system should operate with the maximum heat rate to cover the annual heat demand. The reason why building blocks and welfare buildings showed a higher utilization time than the office buildings was due to the fact that the heating system operated longer and the use of the DHW was higher. The other buildings (office, primary school and sport center) had a high maximum heat demand with lower total heat use and thereby the utilization time was low. Table 7 shows that the coincidence factor was higher for buildings where the heat demand was dependent on the outdoor temperature and the share of the DHW use was in general low. Due to diversities in the heat use, the coincidence factor was lower at the building blocks. A summary of the specific energy demand and annual heating demand for the customers in Table 7 is shown in Fig. 5. In Fig. 5, it is possible to note that the substations cs1, cs2, cs6, cs7 and cs8 showed total annual heat demand around 60 kWh/m^2, however, the maximum specific heat rate (W/m^2) was different.

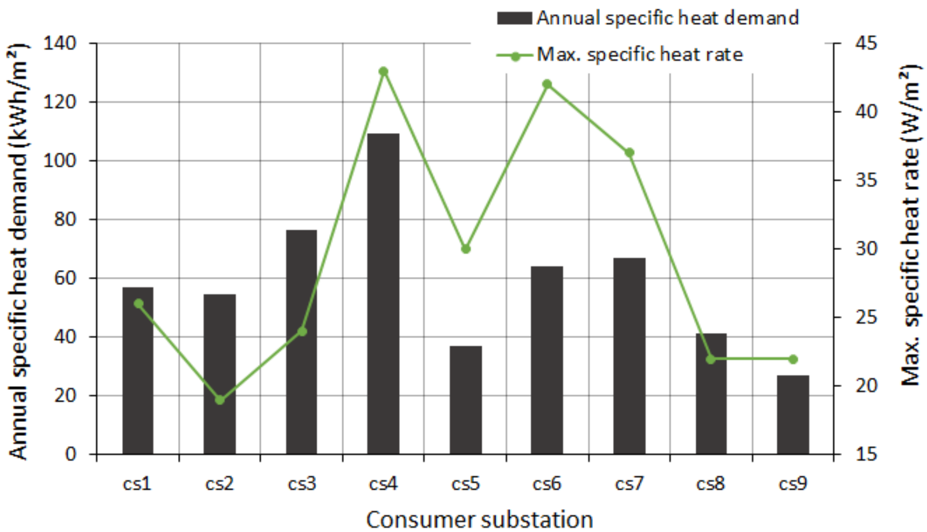

Fig. 5 Specific annual heat demand and maximum specific heat demand for customers

For the heat demand data in Table 7, the total heat load specification was as: the coincidence factor was 0.83; the maximum capacity was 791 kW, and the total annually delivered heat was 1.9 GWh. Finally, Fig. 6 shows the heat duration curve for aggregated heating load in the reference system.

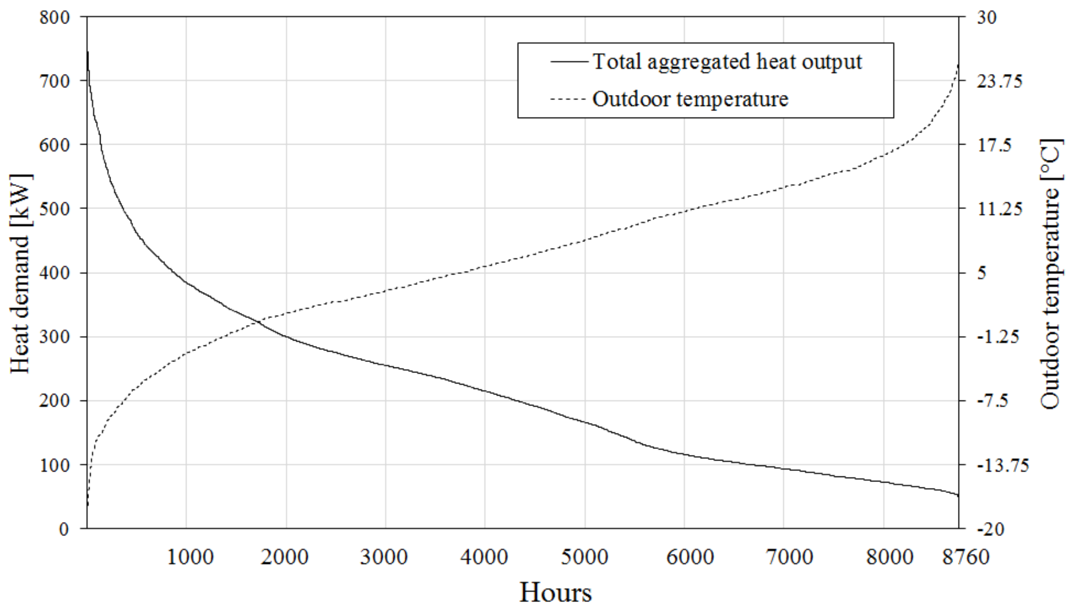

Fig. 6. Heat duration year for the reference year

4.2. Temperature distribution in the DH grid

The existing DH system was designed for delivering heat at 80°C in the supply line during the winter conditions and 70°C during the summer. For the scenario with lower supply and return it was considered to decrease the supply temperature no lower than 55°C to avoid the Legionella issue. Hence, Fig. 7 shows the hourly distribution of the supply and return temperatures for two scenarios. The results in Fig. 7 were calculated based on the methodology introduced in Section 2. To recall, please see Table 4 where all the analyzed temperature levels were introduced.

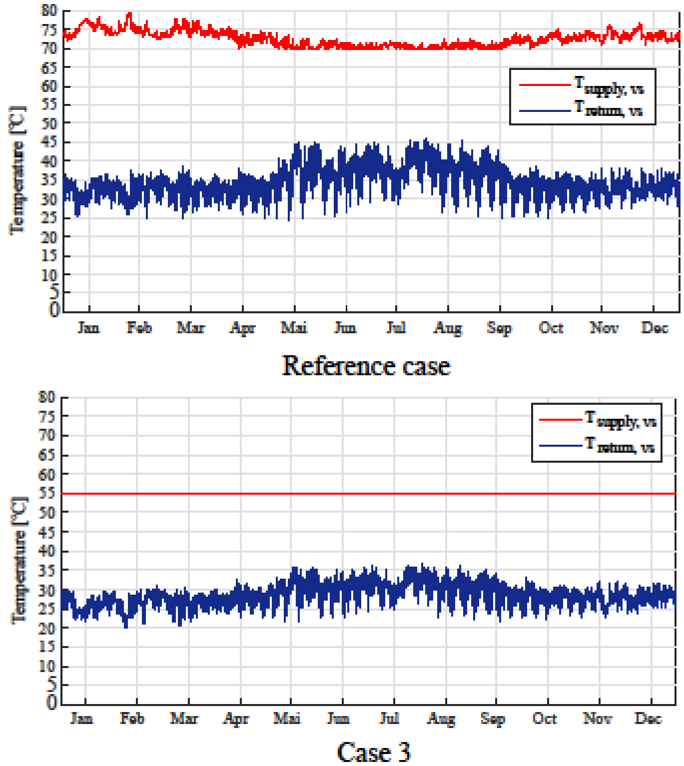

Fig. 7. Supply and return temperature distribution over the year for the high and LTDH system

From Fig. 7 it can be seen that the analyzed DH system (see Fig. 3 and Section 3) operated with the lower temperature difference during the summer time due to the lower heat demand. Further, the analysis of the reference case revealed that the average temperature difference was 45 K during the winter and 30 – 35 K during the summer. For the LTDH scenario (Case 3) the temperature difference was 30 – 35 K during the winter and 25 K during the summer. To recall, the reference case was designed for the temperature difference of 45 K, while Case 3 (LTDH) was designed for the temperature difference of 30 K, see Fig. 4. All these meant that Case 3 or LTDH managed at some extend better to maintain the design temperature difference over the year, regardless of the change in the heating load. This advantage of LTDH would enable reliable values of the return temperature over the year. This was an important conclusion with focus on the lower return temperature.

4.3. Heat loss and pump energy

The annual distribution of heat losses for the various scenarios in Table 4 are shown in Fig. 8.

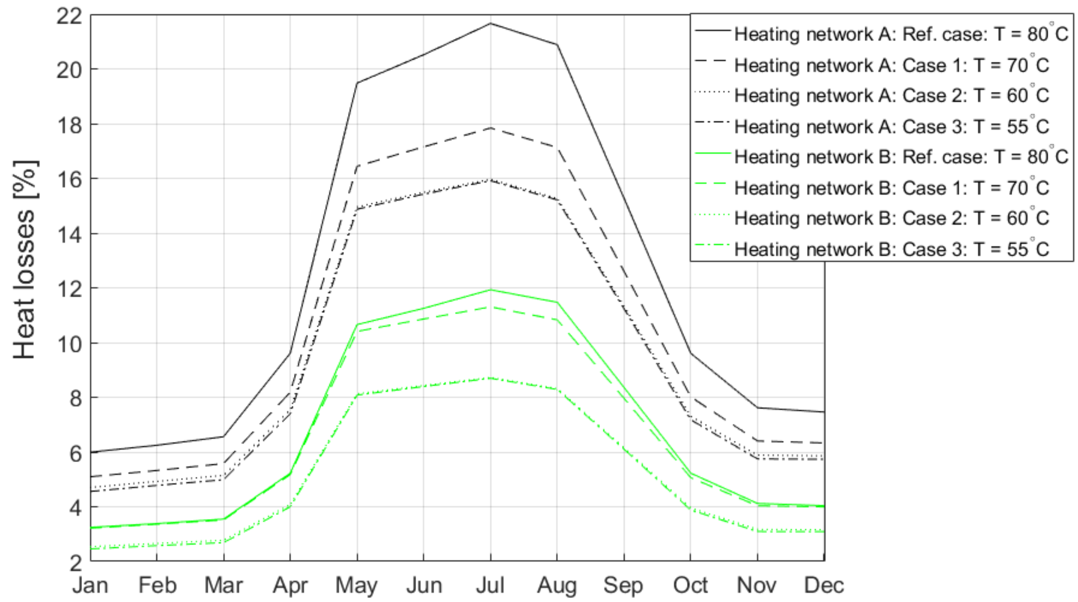

Fig. 8 Heat losses in the heating networks A and B

The results in Fig. 8 show that the heat losses percent had higher values during the summer months than during the winter months. This reason for this was that a lower heat amount was delivered during the summer, while the warm water was always circulating for the DHW use. Due to compact structure of the network B, the heat losses were lower than in the network A.

For the DH networks shown in Fig. 3, the results for the existing DH system showed that the heat losses could be reduced by 25% while decreasing the supply temperature from 80°C to 55°C for both heating network A and B with no change in pipe diameters. The maximum pumping power would increase up to 107% and annual pump electricity use would increase by 58% for the heating network A. The results for the heating network B showed values of 92% and 54% for the pumping power and electricity use, respectively. The reason for such results is that the heating networks A and B were structurally different.

Fig. 9 shows pump energy and pump power with the decrease in the temperature levels for the heating networks A and B.

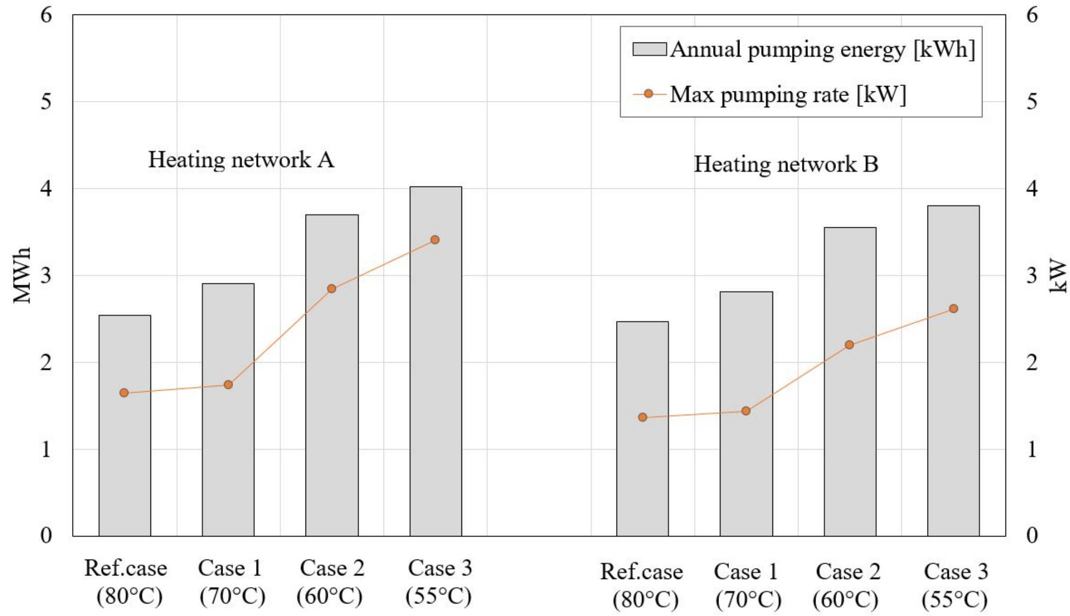

Fig. 9 Annual pumping energy and maximum pumping power with various temperature levels in the existing DH system

Fig. 9 revealed that the reduced temperature levels in heating network A lead to higher increase in both the pump energy and pump power in comparison to the network B.

Further, the new development with the LTDH system was analyzed. To recall, the new development was developed as the LTDH system with the supply temperature 55°C, see Fig. 4. Table 5 gives the overview of the pressure drop constraints for the new developments. Fig. 10 shows an overview of the pump performance results for different pressure drops in the heating networks A and B.

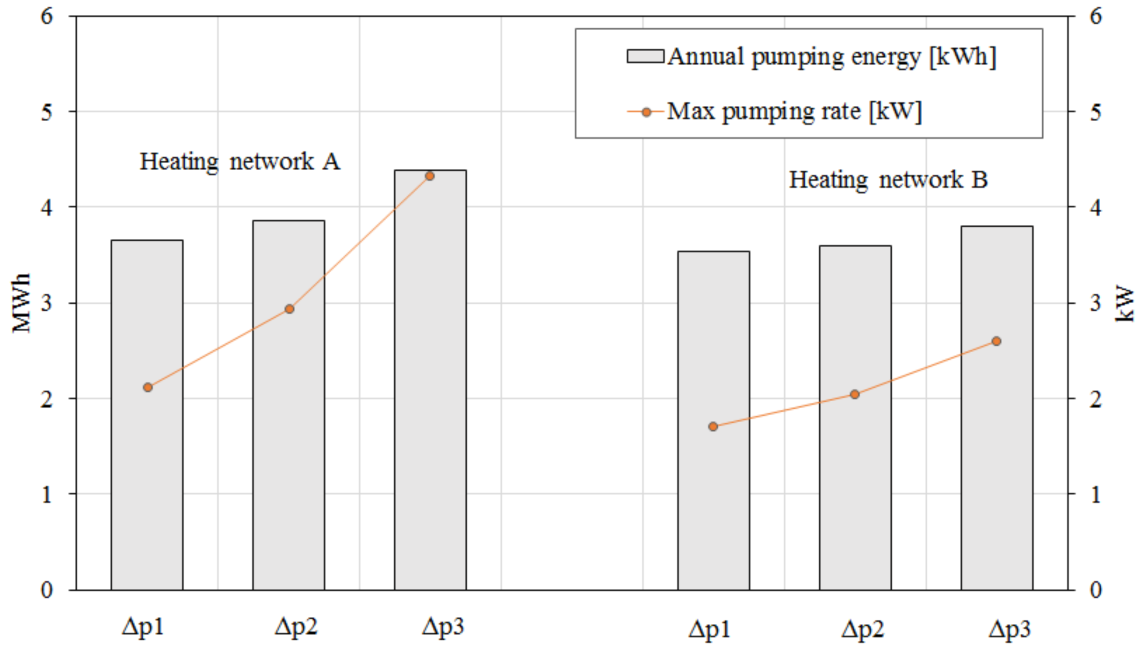

Fig. 10 Annual pump energy and maximum pump power under different conditions for pressure drop in the new heating network

It can be noted from Fig. 10 that the heating network A was especially sensitive to the increased pumping power under reduced pipe diameters. The results showed that the heating network B was less sensitive on the change in the pumping power due to higher linear density. Further, it was concluded that the reduction in the pipe diameters of about 20 % lead to increase in the pressure drop of the system. This means that the pump power increased faster than the pump energy on annual basis, and this was significantly higher for the heating network with the lower linear densities in comparison to the network with the higher linear densities.

The analysis of pump performance in the case of the new developed DH system showed that with the increase in the R-value from 200 - 800 Pa/m in the service lines and 150 - 600 Pa/m in the main line would increase the pump power by 105% and the pump energy by 20% for the heating network A. For the heating network B, these values were 53% and 7.65%. This shown that linear density plays the crucial role for the transition to LTDH if increase in R-value would be allowed. In the case of the area B with the higher heat density, the pump energy use did not increase much due to temperature decrease. This was an important conclusion regarding competitiveness of the DH system in the new areas. The heat losses in the heating network A and

B were almost not affected due to increase in R-value. This is because pipe diameters were not changed significantly with the change of R-value. The pressure drop is approximately proportional to the pipe diameter in fifth extent and thereby was affected in a higher degree than the heat loss results.

Based on the results on the heat losses and the pump energy use introduced above, a trade-off analysis between these two indicators related to LTDH systems was made. The analysis considered how the supply temeprature decrease would change the LTDH performance. A summary of the key performance indicators considering the decrease in the supply temperature is given in Table 8. Please note that the results in Table 8 are valid for a small DH grid given in Fig. 3. In Table 8, specific pump energy is introduced as a relation between the electricty use in kWhel and the heat delivery in MWhh. Here “el” is used to mark electricity and h to mark heat. To provide a general conclusion how the decreased supply temperature may change the LTDH perforamnce, a trad-off analysis between the heat losses and the specific pump power is given in Fig. 11.

Table 8. Resulting operation performance for the low and high heat density grids considering different supply temperatures

	Heating network A - low heat density			
	Reference case (80ºC)	**Case1 (70ºC)**	**Case2 (60ºC)**	**Case3 (55ºC)**
Max. pump rate (kW)	1.64	1.74	2.84	3.40
Annual pump energy (MWh)	2.54	2.9	3.7	4.02
Specific pump energy (kWhel/MWhh)	1.34	1.53	1.95	2.12
Annual heat loss (MWh)	169.3	142.2	129.9	127.5
Heat losses in %	8.91	7.48	6.84	6.71
	Heating network B - high heat density			
Max. pump rate (kW)	1.36	1.43	2.20	2.61
Annual pump energy (MWh)	2.47	2.81	3.55	3.80
Specific pump energy (kWhel/MWhh)	1.30	1.48	1.87	2.00
Annual heat loss (MWh)	92.1	77.1	70.3	68.9
Heat losses in %	4.85	4.06	3.70	3.63

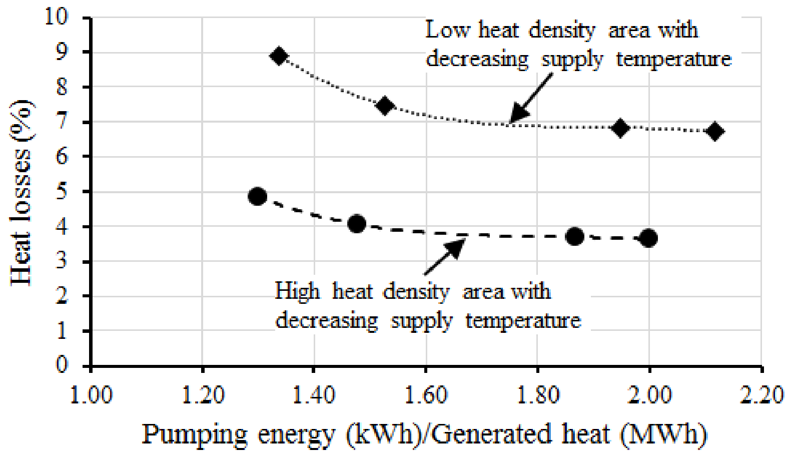

Fig. 11. Trade-off between heat losses and pump energy use considering decreasing of the supply temperature

The relative demand for electricity for pumping is about 0.5 % of the heat delivery, as noted in [4]. Therefore, the above results on the relation between the pump energy use and the heat delivery in kWhel/MWhh should be treated as valid. The results in Table 8 and Fig. 11 show that by decreasing the supply temperature, a considerable amount of the heat would be saved, while the increase in the annual pump energy use would not be high. The results of the trade-off analyses should not be treated by analyzing the percent difference between the resulting performance values, because such an analysis might not show the full advantage of LTDH. The results in Table 8 and Fig. 11 should be rather used for an analysis where the difference in the integral values are used or for an economic analysis, because in such an analysis the advantage of the decrease in the heat losses would be much higher than the increase in the pump energy use.

4.4.Issues with the return temperatures

Sufficient water cooling in the consumer substation and a proper return temperature in the DH network are a result of proper operation of the customer substation. However, this is not often the case and the return temperature can be much higher than expected. The literature review showed that the largest number of fails leading to high return comes due to inappropriate operation [39] of the customer substations.

The analyzed errors are introduced in Section 3.2 and the main finding are presented here.

Fault in the return temperature due to bypasses or short circuits

Table 9 shows increase in the return temperature during the summer and winter due to the bypass. The simulation implied use of constant percentage of the mass flow rate. The winter period was considered from October to March, and summer period was from April to September. The results are valid for both heating networks A and B.

Table 9. Average increase in return temperature due to bypassing of supply medium

Share of mass flow in bypass		1%	2%	5%	10%
Winter	Ref. case (80°C)	0.7 K	0.7 K	1.8 K	3.7 K
	Case 3 (55°C)	0.2 K	0.5 K	1.3 K	2.5 K
Summer	Ref. case (80°C)	0.4 K	0.5 K	1.2 K	2.6 K
	Case 3 (55°C)	0.1 K	0.4 K	1.0 K	2.0 K

The results in Table 9 show that increase in the average return temperature is higher for the reference case in comparison to Case 3 (LTDH). The findings for 10% could be considered as the most representative, since literature review indicated that this percentage is typical for the Swedish DH system leading to increase in the return by 4 K [4]. As it was explained previously, the bypass is used in the DH systems with low heating densities and high heat losses. This is done to avoid the risk of temperature drop below predefined minimum level. This is particularly relevant during the summer season when the mass flow rate is low and the heat losses are high. For the low energy buildings, the heating season is normally shorter. Thus, the bypass valves can have a greater impact on energy efficiency of LTDH networks associated with the low-energy buildings compared with the traditional DH systems [40]. The results showed that effect of bypassing to the return temperature was less during the summer than during the winter. The return temperature versus the outdoor temperature for different bypassing percentage is shown in Fig. 12.

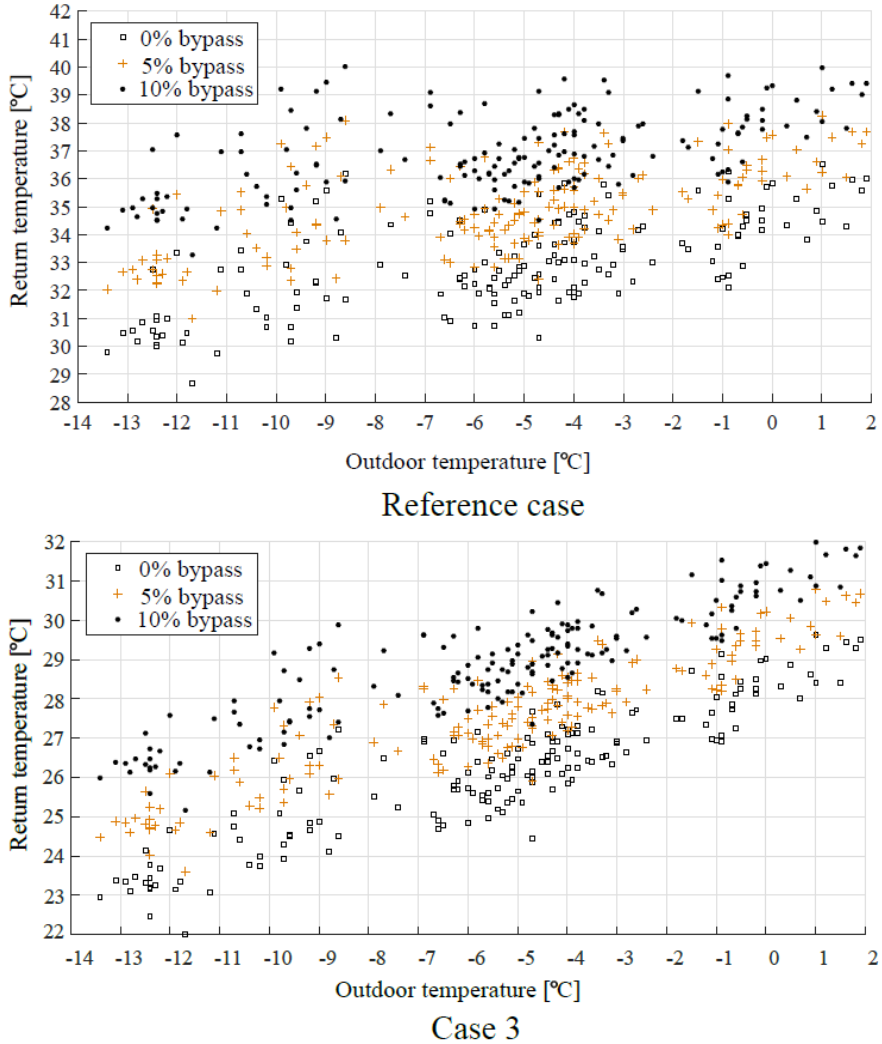

Fig. 12 The return temperature affected by bypassing for the reference and the LTDH case

From Fig. 12 it can be noticed that the return temperature increased with the increase of the bypassing percentage. The highest increase was observed for the 10% bypassing for Case 3 (LTDH). The same conclusion could be drawn for the reference case. Further, it was found that the increase in heat losses due to bypassing of 10% was in the range of +3.1% to +3.5% depending on the heating network. The heating network B showed higher values due to shorter pipes than in network A. In spite of the fact that the bypassing led to increase in the return temperature and heat losses, this can be limited significantly by employing twin pipes in comparison to single pipe solutions [11].

Fault in the return temperature due to aging and fouling of the heat exchangers

The result of changing the temperature efficiency of the heat exchanger showed that the return temperature was more sensitive to varying the temperature efficiency on the heat exchanger for the DHW than the heat exchanger for the space heating. The results showed that the average annual temperature difference was decreased by 3 - 5 K when the temperature efficiency of the heat exchanger was decreased from 0.85 to 0.6. The higher decrease was experienced for the reference system with the high temperature and for the DHW heat exchangers.

Faults in the return temperature due to the indoor set-point temperature

The low energy buildings and passive houses are designed for achieving low energy use, however, quite often it is opposite and heat energy use shows much higher measured values [41]. In Table 7 it was shown that the customers energy use varies from building to building in reality. The reason for this could be either fail in operation, an effect of the user behavior, or occupancy level in the apartments. Further, it is known that some customers set higher requirements for indoor climate and comfort level; however, this should not be an issue in the low energy buildings. In spite of this fact, the energy use data showed that buildings C and D had higher energy use in comparison to the building blocks A and B. Therefore, the analysis looked how the indoor set-point temperature could affect the return temperature in the DH system. Fig. 13 shows the effect on the temperature difference between the supply and the return temperature due to different indoor set-point temperatures.

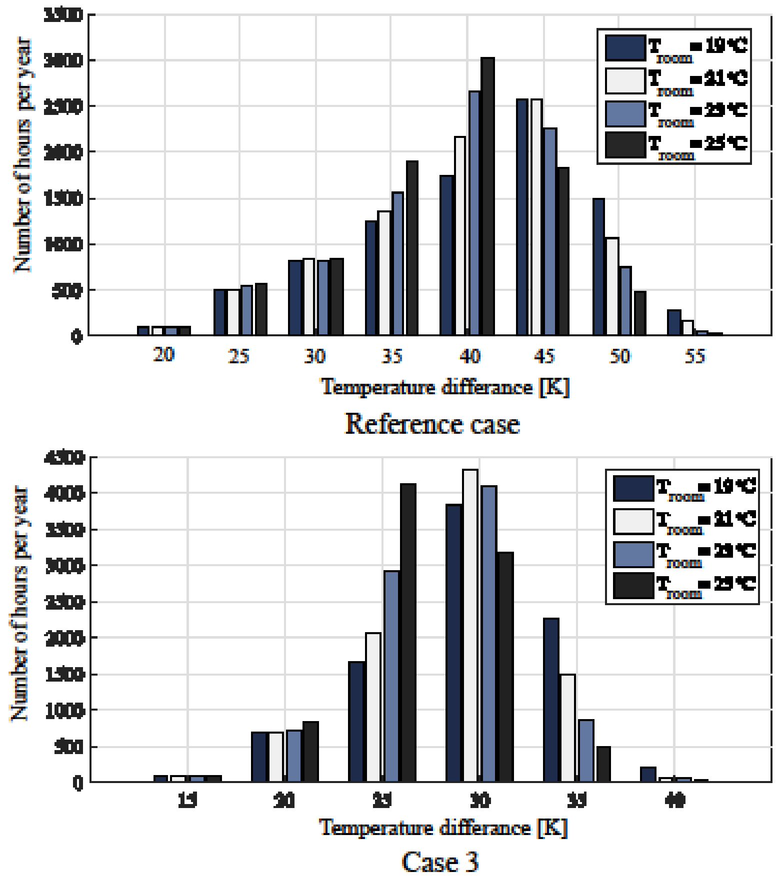

Fig. 13. Temperature difference between the supply and the return temperature at primary side of the customer substation influenced by the indoor set-point temperature

The comparison of the results in Fig. 13 showed that the reduction in the temperature difference due to higher set-point temperature was bigger for the reference case than for Case 3. To recall, the reference case was designed for the temperature difference of 45K, while Case 3 had the supply temperature of 55°C resulting in a lower temperature difference. In Fig. 13, it may be noted that the temperature difference was 40 K for the reference case a bigger part of the operation hours, while it was 30 K for Case 3 regardless of the set-point indoor temperature. The decreased set-point indoor temperature led to increase in the temperature difference, which resulted in efficient operation of the DH system. Case 3 showed a higher sensitivity in the operation parameters of the DH network due to increased set-point temperature. A summary of the set-point indoor temperature influence on the DH network operation is given in Table 10.

Table 10. Influence of the set-point indoor temperature on the DH network operation

Indoor temperature		19°C	23°C	25°C
Annual heat loss	Ref. case (80°C)	-1.0 %	+1.0 %	+2.1 %
	Case 3 (55°C)	-1.4 %	+1.5 %	+3.1 %
Annual pumping energy	Ref. case (80°C)	-3.1 %	+3.5 %	+7.5 %
	Case 3 (55°C)	-5.3 %	+6.8 %	+15.0 %
Maximum flow rate	Ref. case (80°C)	-3.2 %	+3.5 %	+7.1 %
	Case 3 (55°C)	-4.5 %	+5.8 %	+10.8 %

The summary results for the entire analyzed network in Table 10 show that both heat loss, annual pumping energy, and the maximum flow rate increased due to increasing of the set-point indoor temperature. Further, it can be noted that the annual performance were more sensitive on the increase in the set-point indoor temperature for Case 3 than for the reference case.

Faults in the return temperature due to fail adjustment of the outdoor compensation curve

Different shapes of the outdoor temperature compensation curve were tested. The results showed that the change in the outdoor temperature compensation curve influenced the return temperature and thereby heat losses and the pump energy of the DH system. In general, the best performance was obtained in the case when the water flow rate through the heat exchanges was low. The change in the average temperature difference in winter due to implementation of the different compensation curves was up to 3 K.

4.5. Competitiveness of LTDH in the low heat density areas

The long-term program aiming improvement in the building energy efficiency will facilitate decreasing of the linear heat densities (MWh/m) and heating densities (MWh/m^2). This could create new challenges for LTDH, since the profitability of the DH system is dependent on high demand and high linear density. Therefore, it was of interest to examine how linear densities affect heat losses and pumping energy in the heating networks A and B. Different linear heat densities were introduced by changing the length of the networks given in Fig. 3. Fig. 14 shows relative heat losses under various linear densities for heating networks A and B.

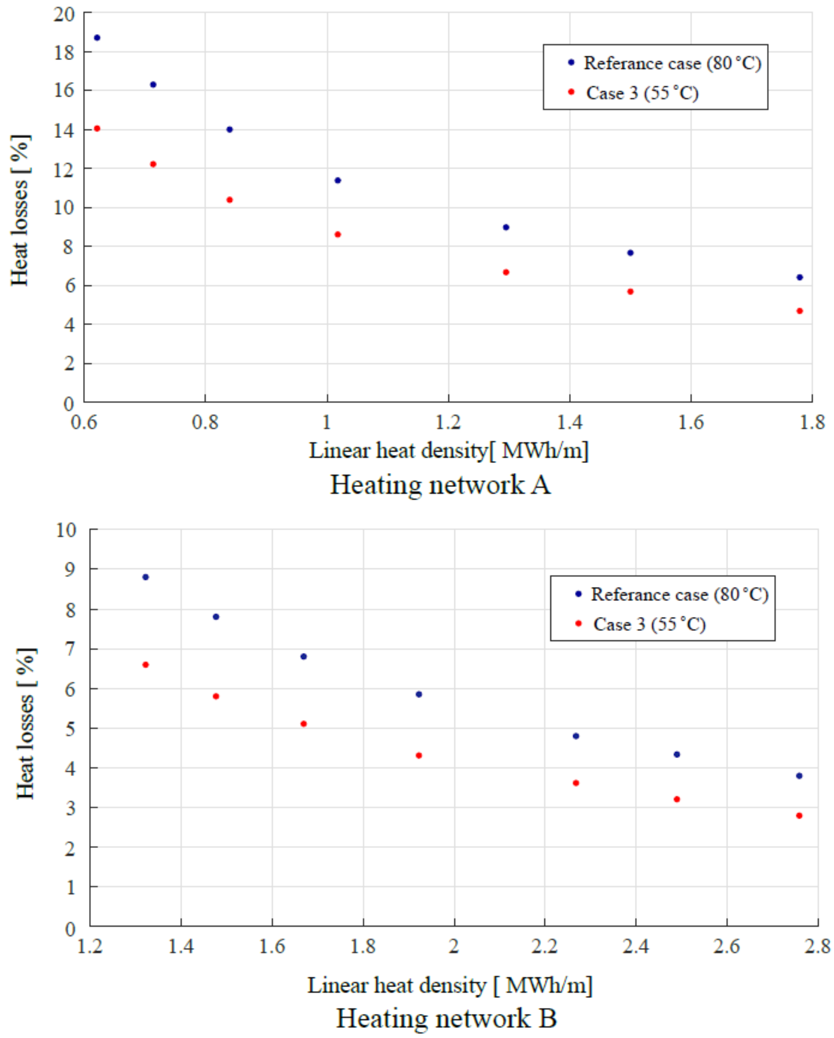

Fig. 14. Heat losses as a function of linear densities for heating networks A and B

Fig. 14 shows that the percentage of the heat losses increases with the decrease of linear densities. From the results in Fig. 14, it could be noted that the reference case and Case 3 showed a similar trend regarding the percentage of the heat losses when the heat density decreases, while Case 3 had lower percentage heat losses. This meant that LTDH might be competitive in the low heat density area while supplying the low and passive building. In general, the percentage of the heat losses was lower for the heating network with the higher heat density, the network B.

Further, the pumping energy (kWh) per delivered heat as a function of linear density in the heating networks A and B is shown in Fig. 15.

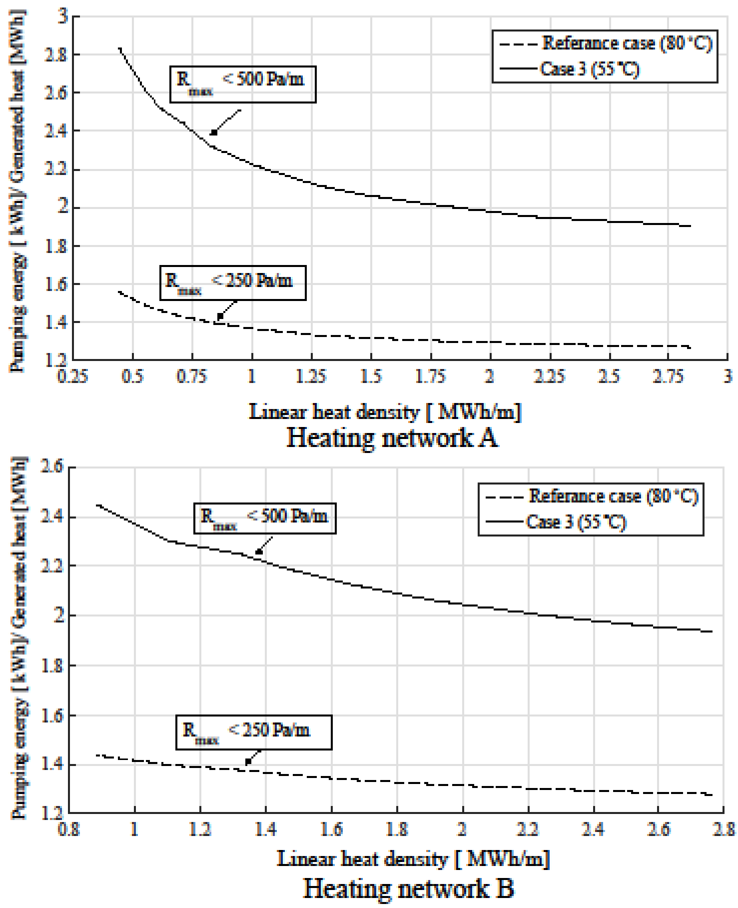

Fig. 15. Pumping power for different linear density and different pressure drop per pipe length

From Fig. 15 it should be noted that the low supply temperature of 55°C in Case 3 induced increase in pumping energy in comparison to the higher supply temperature of 80°C. This effect was more critical to for the heating network A with the low heat density. In Fig. 15, it may be also noted that the pump energy increased more for the heat density lower than 1 MWh/m for the heating network A. This result fits well with the DH profitability threshold in Denmark and Canada [29, 30]. Therefore, the results in Fig. 14 and Fig. 15 may be used to find the DH profitability threshold for LTDH with the low energy buildings.

5. Discussion

The model of the DH network was developed based on several important assumptions, therefore, it is vital to give some critics.

The heating rates that have been used as the input to the model were obtained from the existing customers of the DH system. Unfortunately, it was impossible to analyze how heating rate could be changed due to adjustment of the heating system at the customer side. With the change of the supply temperature from 80°C to 55°C at the primary system, the supply temperature to the DHW system has decreased from 65°C to 50°C. In practice, this would also lead to decrease in heating demand. By decreasing the supply water temperature in the existing system, the water flow rate is kept within the existing limits, meaning that the maximum heat rate would decrease by decreasing the supply temperature. However, this was not analyzed in this study. If heating demand is reduced with the reduction of temperature levels, then, a lower mass flow rate effecting pressure drop and pumping effect should be expected. Further, it was unknown whether the DHW system employed storage tanks. The customers with the storage tank would show different energy use pattern. In addition, the DHW storage tank would require a higher supply temperature to avoid the Legionella issue. As it was mentioned before, the supply temperature was reduced to 50°C; therefore, the measures should be taken to avoid the Legionella risk.

The analysis focused on radiator heating, however, the floor heating and ventilation system were not considered for the analysis. The floor heating, for example, is an accepted solution for heating in kindergartens. It requires lower supply temperature than radiator system and therefore, can provide lower return temperature than 30°C. This could facilitate decrease in the return temperature of the DH system and the heat losses. However, the floor heating has longer response time in comparison to the radiator system.

Another uncertainty that should be mentioned is the model of the customer substation. The customer substation model, introduced in Section 2.3, is a simplified solution for heat delivery under ideal conditions. The substation was modeled as one stage DHW heat exchanger with parallel connections for the DHW and the radiator heating. The large buildings are often equipped with two stage heat exchangers to decrease the return temperature. The modeled customer substation employed a constant temperature efficiency for the heat exchangers. In reality, the efficiency would vary according to the mass flow rate, hence the different values of the return temperature could be expected. For the more complex analysis of the change in the return temperature due to various substation design, a further analysis of the heat transfer coefficient and the heat transfer surface should be made.

The pressure drop in a distribution system is dependent on several factors like mass flow rates, pipe diameters, pipe lengths, branches and bends, valves and filtering equipment. The study employed the constant friction coefficient. However, the friction coefficient is dependent on roughness of the internal pipe surface and Reynold's number. The uncertainty goes for roughness, since its affects factors like corrosion coating that could lead to pipe deterioration over time. Therefore, the analysis on friction coefficient is required to identify the effects on the pressure drop and pumping energy in the DH grid.

6. Conclusions

The study aimed to analyze energy efficiency potential and related issues of the DH system under the transition of the existing heating network to LTDH. Two DH networks were analyzed, with the low linear heat density of 1.3 MWh/m and with the higher linear heat densities of 2.3 MWh/m. The study considered a group of real customers located in Trondheim, Noway, with existing heating demands. The observed buildings were built as low energy and passive house buildings.

The transition of the existing DH system was planned in two ways: 1) decreasing the supply water temperature without change in the pipe size; 2) sizing the DH network based on the LTDH requirements. For the existing DH system, the design pressure drop was in the range of 50 - 250 Pa/m and 80°C for the supply temperature. The results showed that the pressure drop doubled with the reduction of the return temperature from 80°C to 55°C, and thereby the pumping power and energy increased. This increase was lower for the grid with the high heat density due to the shorter DH network. Further, it was found that the transition to LTDH facilitates the reduction of the heat losses up to 25 %. A large scale DH system would show higher heat losses due to versatile network structure with different linear densities. The trade-off analysis between the heat losses and the pump energy use showed that considerable amount of the heat would be saved by decreasing the supply temperature, while the increase in the annual pump energy use would not be high.

The study examined the causes that could lead to increase in the return temperature of the DH network, such as bypassing, exhaustion of heat exchangers, influence of the indoor set-point temperature, and adjustment of the outdoor compensation curve. The results showed that the bypassing induced less increase in the return temperature of LTDH in comparison to the existing DH system. The results on exhaustion of the heat exchangers showed that the average annual temperature difference was decreased by 3 - 5 K due to exhaustion of the heat exchangers. This effect was bigger for the existing system with the high temperature and for the DHW heat exchangers. In the future, the heating season for buildings will be shorter, therefore, it is expected that heat exchangers placed in the substations for the space heating and the DHW will play even a more important role for the return temperature decrease. Analysis of the increase in the indoor set-point temperature showed that the LTDH system was more sensitive to those changes. The result showed that with the increase of the indoor room temperature from 21°C to 25°C the mass flow rate increased by 11% for LTDH and by 7% for the existing system. This lead to increase in the pumping energy by 15% and 7.5%. Further, the results showed that proper choice of the outdoor temperature compensation curve could facilitate increase of the temperature difference by 3 K. In addition, the heat losses and pumping energy were reduced.

The heating demand is expected to decrease for future buildings. With low population densities this can challenge the competitiveness of DH systems due to lower linear heat densities (MWh/m). The results considering different linear densities showed that percentage heat losses increased for the low linear heat densities. With the reduction of the supply temperature to 55°C, the pumping energy increased in comparison to the temperature level of 80°C. This was due to the lower temperature difference that led to the changes in the mass flow rate and the high pumping power rate. One of the main findings regarding the competitiveness of LTDH was that pump energy increased more for the heat density lower than 1 MWh/m for the heating network with the long distance. The conclusions related to the competitiveness of LTDH may be used to evaluate the threshold value for the DH system connection. This is particularly relevant in the areas with low linear densities.

Several assumptions were made under the study of LTDH. Therefore, it is important to have a critical look on the findings. All the mentioned findings should be considered carefully, because the benefits of lowering the temperature in the DH system could be mistreated due to aforementioned issues. Nevertheless, the results provided a clear picture of the improvement in energy efficiency of the DH network due to lowering the temperature level and huge potentials and challenges related to LTDH.

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