

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.

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

29

30 ***Nomenclature***

31 L [m] – pipe length

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

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

34 $T_{in,i}$ [°C] – temperature at inlet of pipe

35 $T_{out,i}$ [°C] – temperature at pipe outlet

36 T_1 [°C] – supply temperature

37 T_2 [°C] – return temperature

38 T_g [°C] – ground temperature

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

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

41 $T_{r,s}$ [°C] – return temperature in the secondary side of heat exchanger

42 $T_{s,p}$ [°C] – supply temperature in the primary side

43 $T_{r,p}$ [°C] – return temperatures in the primary side

44 $T_{s,d}$ [°C] – design supply temperature

45 $T_{r,d}$ [°C] – design return temperature

46 T_i [°C] – indoor room temperature

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

48 ΔT_m [K] – mean arithmetic temperature difference

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

- 50 U_i [W/mK] – overall heat loss coefficient
- 51 U_{11} [W/mK] – heat loss coefficient in the supply pipe without thermal influence of return
52 pipeline
- 53 U_{22} [W/mK] – heat loss coefficient in the return pipe without thermal influence of supply
54 pipeline
- 55 U_{12} [W/mK] – heat loss coefficient due to thermal influence of return pipeline
- 56 U_{21} [W/mK] – heat loss coefficient due to thermal influence of supply pipeline
- 57 p_1 [Pa] – minimum permitted pressure in a DH network
- 58 Δp_{cs} [Pa] – pressure drop over the customer substation
- 59 Δp_{ah} [Pa] – pressure drop in the heat exchanger delivering the heat to the observed area
- 60 Δp_t [Pa] – increase in pumping pressure
- 61 Δp_{tot} [Pa] – total pressure drop
- 62 \dot{m} [kg/s] – mass flow rate
- 63 \dot{Q}_{hd} [W] – heat demand
- 64 \dot{Q}_{hd} [W] – actual heat demand
- 65 $\dot{Q}_{a,hd}$ [W] – design heat demand
- 66 \dot{V}_t [m^3/s] – total volume flow rate
- 67 c_p [kJ/kgK] – specific heat capacity of water
- 68 d_i [m] – internal pipe diameter
- 69 f [–] – friction coefficient
- 70 n_1 [–] – radiator exponent
- 71 ρ [kg/m^3] – water density
- 72 ε [–] – temperature efficiency of heat exchanger

73 η [–] – pump efficiency

74

75 *Abbreviations*

76 cs – Consumer substation

77 DH – District heating

78 DHC – District heating and cooling

79 DHW – Domestic hot water

80 LTDH – Low temperature district heating

81

82 **1. Introduction**

83 Use of renewable energies and waste energy is highly necessary and required by national
84 and international regulations [1]. Future district heating and cooling (DHC) systems may enable
85 transition to a complete renewable society [2], meaning that the future DHC systems will be based
86 on completely renewable energies such as solar, waste heat, and geothermal energy. To enable
87 higher share of renewables in the DHC system, the temperature of the district heating system
88 should be lowered [3]. Throughout the historical development of the district heating (DH) system,
89 the system development has passed through four generations [4]. At the beginning, the DH
90 temperature was decided based on energy supply technologies that were providing high
91 temperature water. Consequently, the DH temperature was decreased due to implementation of
92 heat pumps and a general idea to increase the system efficiency by decreasing the DH temperature
93 [5]. Lately, due to lower heat demand in new passive and low energy buildings, the supply
94 temperature could be decreased to 45 - 55°C [5]. The DH system operating with these temperature
95 levels is called low temperature DH (LTDH) [3, 6]. There is no a clear limit for LTDH, but the
96 temperature levels should be within the given range.

97 Current DH systems belonging to the second and third generation with the temperature of
98 80 to 100°C are facing many challenges [4]. In low energy buildings, there is no need for heating
99 at high temperature levels and heating demand is decreasing. High temperature levels are

100 unfavorable in terms of utilizing renewable energy sources and waste heat. Finally, due to higher
101 share of the distribution losses in the total DH heat demand, competitiveness of the DH system in
102 the low heat density area is decreasing [7]. In order to be competitive in the areas with low heat
103 densities and low energy buildings, it is important to achieve low heat losses for a high efficiency
104 of the DH system. In general, the heat losses in Norway are in the range of 8-15% of the
105 delivered heat [8, 9]. The merit with the low supply temperature is that the temperature difference
106 between the pipe and the ground is less than in the case with the high supply temperature. This
107 facilitates that the heat losses to the ground are less and the demand for insulation can be reduced
108 for certain DH areas. LTDH systems have better opportunities for utilization of waste heat and
109 renewable heat sources, as well as lower distribution losses. However, on the way to the
110 transition to LTDH, there is a problem with high return temperature and the low temperature
111 difference between the supply and the return temperature in the network [10].

112 Until now, several LTDH projects have been successfully realized. Some conclusions and
113 the most important characteristics are explained. Seven low energy apartment buildings in
114 Lystrup, Denmark, have been connected to LTDH in an attempt to reduce distribution losses [11].
115 This is done by reducing pipeline dimensions, setting the distribution temperature to 55°C, and
116 using twin pipes. In addition, booster pumps that raise the pressure in the area enables further
117 reduction of the pipeline dimensions. In the mentioned project, two consumer substation
118 connection types with LTDH are implemented: 1) with storage tanks and 2) with high heat output
119 heat exchangers [11]. The use of DH storage tanks makes it possible to reduce the pipeline
120 dimension as it reduces the peak demand. The total costs and benefits of these two alternatives for
121 the LTDH connection are roughly the same, and both are viable solutions. The result is a reduction
122 of energy use of 75 % compared to the traditional DH systems [11]. In Albertslund, Denmark,
123 LTDH has been introduced for 1544 refurbished houses from the 1960's. The distribution network
124 for DH was replaced by twin piping laid in shorter routes, and the houses were completely
125 renovated to a standard close to today's low energy regulations. The houses were fitted with
126 individual heat exchangers for instant heating of domestic hot water (DHW). This design requires
127 a higher peak heat rate from the DH network, but eliminates the need for storage tanks. It is
128 expected that this solution will result in a 62 % reduction in distribution heat losses. This is
129 achieved at an extra cost of 20 million DKK and will result in a profit of 31 million DKK over the
130 project lifetime of 50 years [12]. In Chalvey, England, a small scale LTDH network has been

131 constructed, supplying ten zero emission houses. The houses are equipped with photovoltaics to
132 cover part of their electricity demand, while DH covers the heat demand. Each house is equipped
133 with a heat exchanger for DHW. Heat is produced using biomass, air to water heat pump, two
134 ground-source heat pumps, and 20 m² of solar collectors. The DH central contains a large storage
135 tank. The heat in the storage tank is used to cover peak load. Due to possibility to charge the
136 storage tank by any of the mentioned energy supply technology, the flexibility of the system is
137 increased. Heat production is optimized for a low temperature system of 55°C and LTDH is
138 completely based on renewable energies [13]. In all these examples, the introduction of LTDH has
139 produced good results when it comes to distribution loss savings, low temperature heat production,
140 and customer satisfaction.

141 Since the issue on the distribution losses is relevant for the future development of LTDH
142 [4], it is highly important to model properly the distribution system with belonging issues. There
143 have been different studies dealing with steady-state analysis of new concepts for the DH
144 distribution systems [14], detail pressure drop models for the DH system [15], the dynamic heat
145 loss model [16], and pipe network models based on producer data [17]. The mentioned studies
146 have not treated heat dynamics due to annual variation in heat demand and flow and pressure
147 control at the same time. Regardless of the topic importance and future development of the smart
148 DH, the studies on the prosumers in the DH [18, 19] have not provided a general method to be
149 used for the performance analysis, design, and operation optimization of the DH system including
150 distributed heat sources. In our study, the model for the pressure and thermal losses with the
151 hourly heat load input was implemented to treat in a proper way relevant issues in the DH system.

152 In Norway, the new buildings are being built to high standards with reduced space heating
153 demands, and thereby the demand density in the DH network is decreasing. This will induce that
154 the percentage of distribution heat losses is increasing [7]. However, there are still a high
155 percentage of the existing buildings requiring higher temperature and heat demand. The DH
156 system is not a system where the technology and parameters may be changed at once [4], yet there
157 is transition to come to the defined aims [3]. This means that the DH system is under continuous
158 development. For example, once installed pipes will be used as long as they are not damaged
159 regardless of increased heat demand [8]. Therefore, this study by analysing a DH system in
160 Trondheim, Norway, had two aims. The first aim of the study was to estimate the challenges in the

161 transition process from the current DH to LTDH systems, while integrating low energy and
162 passive house buildings. The second aim was to estimate possibilities and increased
163 competitiveness of LTDH in the low heat density areas.

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

171

172 **2. Methodology**

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

182

183 *2.1. DH thermal network model*

184 To be able to keep a sufficient supply temperature to the last customer substation in the
185 DH system, it is important to have reliable calculations on the temperature drop in the
186 distribution network. The temperature drop in the distribution system is dependent on several
187 parameters: temperature levels in the DH system, ground temperatures, the pipe length, the mass
188 flow rate, and the heat loss coefficient [21]. To develop a general model for the thermal losses a

189 part of the pipe with one inlet and one outlet node was observed. Equation (1) gives the outlet
190 temperature of the pipe as:

191

$$192 \quad T_{out,i} = \begin{cases} T_g + (T_{in,i} - T_g) \exp\left(-\frac{U_i L_i}{\dot{m}_i c_p}\right) \\ T_g \end{cases} \quad (1)$$

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

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

208

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

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

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

213

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

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

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

$$220 \quad q_{tot} = q_1 + q_2 = 2 (U_{11} - U_{12}) \cdot (T_m - T_g) \quad (4)$$

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

224

225 2.2. Pressure drop and pumping power

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

$$227 \quad \Delta p_{tot} = 2 \cdot R \cdot L_{tot} + \Delta p_{cs} + \Delta p_{dh} \quad (5)$$

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

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

$$232 \quad R = \frac{\Delta p_f}{L} = \frac{8 \cdot f}{d_i^5 \cdot \pi^2 \cdot \rho} \dot{m}^2 \quad (6)$$

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

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

241
$$p_x = p_1 + H_p - R \cdot L_x - \Delta p_{cs} \quad (7)$$

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

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

251
$$P = \frac{\Delta p_{tot} \cdot \dot{V}_t}{\eta} \quad (8)$$

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

255

256 *2.3. Customer substation model*

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

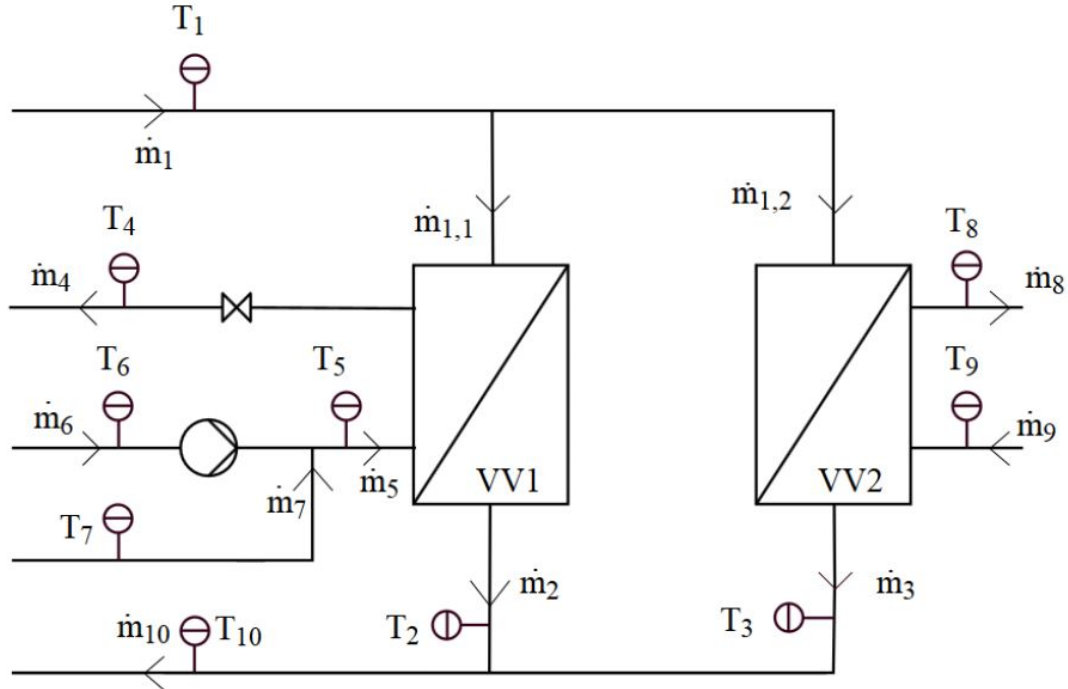


Fig. 1. Layout of customer substation

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} .

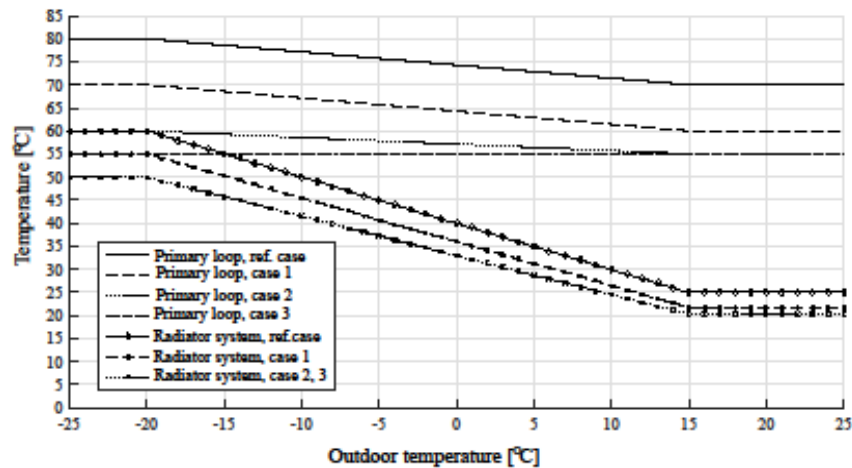
272 However, the challenging part is defining the return temperature and mass flow rate from
 273 customer substation including both the space heating system and the DHW system. The text
 274 below describes the calculation method for the return temperature in the DH system.

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

$$276 \quad T_{r,p} = T_{s,p} - \varepsilon \cdot (T_{s,p} - T_{r,s}) \quad (9)$$

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

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



288
 289 Fig. 2. Outdoor temperature compensation for supply temperature in the primary loop of DH
 290 system

291

292 Table 2. The design supply and return temperatures in the heating system depending on various
 293 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

294
 295 The temperatures shown in Table 2 were selected based on technical considerations of the
 296 DH heat provider, which stated that the design supply and the return temperatures should have
 297 level of 60/40°C or lower [27]. The temperature in the radiator heating system would decrease
 298 with decrease of the supply temperature in the DH system. However, the previous studies showed
 299 that the supply temperature could be as low as 55°C without causing problems with the indoor
 300 comfort level. The high temperature difference between the supply and the return provides low
 301 mass flow rate that could lead to the issues with the control valves. Therefore, this issue was
 302 considered while modeling.

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

306
$$\frac{\dot{Q}_{hd}}{\dot{Q}_{d,hd}} = \left(\frac{\Delta T_m}{\Delta T_{m,d}} \right)^{n1} \quad (10)$$

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

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

311
$$T_r = 2 \cdot \left(\left(\frac{\dot{Q}_{hd}}{\dot{Q}_{d,hd}} \right)^{\frac{1}{n1}} \cdot \left(\frac{T_{s,d} + T_{r,d}}{2} - T_i \right) + T_i \right) + T_s \quad (11)$$

312 where T_s and T_r are the supply and return temperatures of the radiator, $T_{s,d}$ and $T_{r,d}$ are the
 313 design supply and return temperatures, and T_i is the room temperature. The room temperature in
 314 this study was set to be the constant value of 21°C and the radiator exponent equal to 1.3. It can

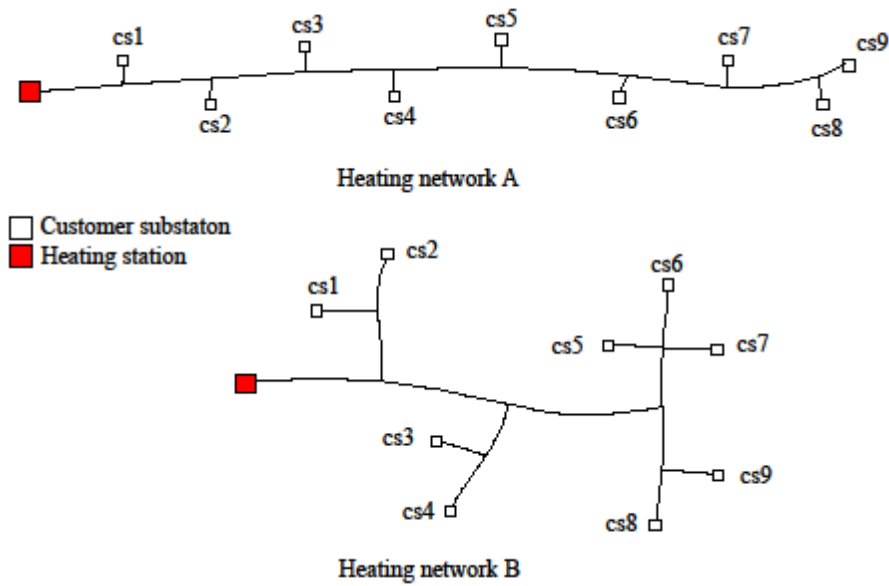
315 be argued what may be the indoor room temperature. Therefore, analysis how the value of the
316 indoor temperature could influence the return temperature was also performed.

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

321

322 **3. Description of the low and high heat density area**

323 The two heating networks, given in Fig. 3 were introduced to estimate the performance
324 and future competitiveness of LTDH. The accurate selection of the network structure is essential
325 for achieving an energy efficient and profitable low temperature heating grid resulting in low
326 specific heat losses. The linear density is a parameter that is used to define competitiveness of a
327 DH network compared to alternative energy supply methods. In addition, it shows how much
328 heat is delivered per meter of the pipe length [28]. At the same time, the competitiveness is
329 dependent on the local topology and the situation on the energy market creating various
330 profitability threshold, for example, 0.2 MWh/m in Denmark [29] and 1.5 MWh/m in Canada
331 [30]. The heating network A represents the case with the low heat density and is characterized by
332 the linear density of 1.3 MWh/m. The high heat density with the linear density of 2.3 MWh/m
333 characterized the heating network B. The linear heat density may have different values,
334 depending on DH network development and building heat demand. For example, for small house
335 areas it may be up to 1 MWh/m, while for small DH networks is it up to 5 MWh/m [28]. In this
336 study, the heat density values for the low and high heat density areas were defined based on the
337 values in [28-30].



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

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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 Varmer 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.

354 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			cs5
Sport center	2724	2008-2009			cs6
Kindergarten	2000	2011	Low energy building standard		cs7
Health and welfare building	2696	2011		64 rooms	cs8
Office building	8600	2010-2011			cs9

355

356 Each building was equipped with a substation providing hydronic space heating and the

357 DHW as given in Fig. 1. The designed heat demand for the substations and the pipes was chosen

358 to be 20 % higher than the heat load in 2013 in order to cover increase in heating demand if

359 necessary. It was assumed that the radiator was used in each room and could cover all the needs

360 for heating. Based on available data these buildings showed low heating demand, hence, it was

361 concluded that they could be connected to LTDH without significant changes in the network

362 structure. LT-DH could be implemented either in existing heating networks or via development

363 of a new DH system. Hence, it was decided to develop a network models for A and B systems

364 and compare the results with the existing DH system. Due to different design requirements for

365 those cases and the reduced temperature difference for LTDH, the analysis aimed to find a

366 solution for the transition of the existing DH systems to more efficient LTDH.

367

368 *3.1. Development of the DH system*

369 For the purpose of the study, it was decided to model both the existing and the new DH

370 systems. The reason for this was that the design conditions for the pipes are different due to

371 different temperature differences. Information flow how development of the DH system may be

372 made for the low energy buildings is shown on Fig. 4. The information flow chart in Fig. 4 was

373 developed in collaboration with the DH company and was implemented to test different DH grid

374 development scenarios.

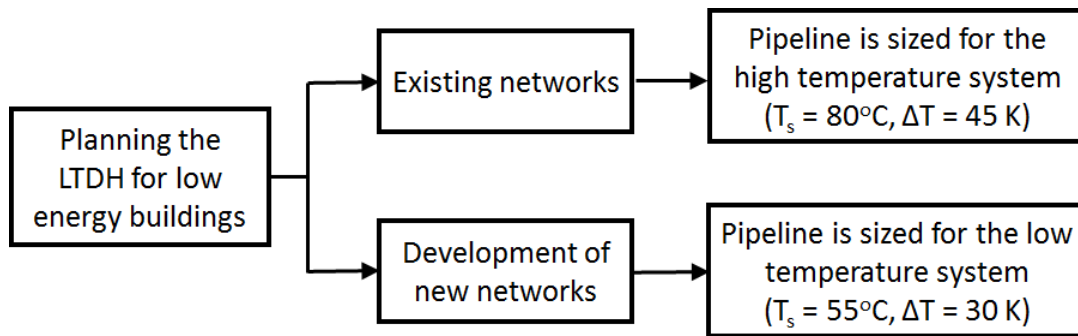


Fig. 4. Planning of LTDH for the low energy buildings

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.

400 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

401
 402 The development of the new DH network implies that the designed pipes have to be able
 403 to satisfy customer demand with the supply temperature of 55°C, with the designed temperature
 404 difference of 30 K. Further, for a better energy efficiency and profitability, it is important that
 405 heat losses remained low. Therefore, the plastic pipes with diameter of 32 mm and smaller with
 406 good insulation characteristics were implemented. The decision about technical parameters of
 407 LTDH was based on several demonstration projects developed in Denmark. Sizing of the new
 408 LTDH network was based on the following:

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

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

419 Table 5. Pressure drop constrains for the new development

Scenarios	$\Delta p1$	$\Delta p2$	$\Delta p3$
Main lines	$R \leq 150 \text{ Pa/m}$	$R \leq 300 \text{ Pa/m}$	$R \leq 600 \text{ Pa/m}$
Service lines	$R \leq 250 \text{ Pa/m}$	$R \leq 550 \text{ Pa/m}$	$R \leq 800 \text{ Pa/m}$

420
 421 To calculate the pressure drop, pressure level, and the pump power, it was necessary to
 422 define certain limits for the calculation. Necessary parameters for calculation of the model
 423 introduced in 2.2 are summed-up in Table 6 based on practical constraints and values given by
 424 the DH company [27].

425 Table 6. Parameters for the presser drop and level calculation

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

426
 427 The minimum static pressure p_1 should be kept above the saturation pressure in order to
 428 avoid boiling and cavitation in the pipe. The saturation pressure is lower than 1 bar for the
 429 temperatures below 100°C. For security margin, the pressure could be increased up to 5 bar [34],
 430 however in this study the value was chosen equal to 1.5 bar due to the small DH system.

431

432 *3.2. Issues with the high return temperature*

433 LTDH is a paramount of the DH technology that should be achieved. The high return
 434 temperature from the customer substation is considered as one of the main issues for decreasing
 435 the supply temperature. Different type of errors causing the faults in the return temperature are
 436 identified such as: system design, heat exchangers, control, and errors outside of the substation
 437 [4, 10]. Based on the literature review and the discussion with the DH company, the following
 438 errors were introduced in the models: short bypasses, aging and fouling of the heat exchangers,
 439 indoor temperature set point errors, and fail adjustment of the outdoor compensation curve.
 440 Introduction of each error in the model is explained in brief below.

441 *Bypass* - The inspection of the DH system indicates that the bypass valves may be
 442 installed intentionally or unintentionally in the primary loop at the customer substation.
 443 Sometimes they may stay open due to fails or neglecting. Further, bypasses or some additional
 444 pipes may be installed sometimes in the system with some purpose, but this has been forgotten
 445 over time. All these may influence a short circulation, i.e. that the supply water is mixed directly
 446 with the return water and thereby the return water temperature is increased. In this study, four
 447 various situations when the water bypasses from the supply to the return line were tested, 1%,
 448 2%, 5%, and 10% of the flow might bypass.

449 *Aging and fouling of the heat exchangers* - Over the time, fouling of the heat exchangers
 450 appears introducing the decrease of the heat exchanger efficiency. This is a minor throughput that

451 causes high return temperatures in customer centers. However, district heating companies are
452 experiencing an increasing amount of leaks in heat exchangers. This problem can lead to high
453 replacement costs of the heat exchanger. It was chosen to briefly investigate how temperature
454 efficiency can affect the return temperature, if there is a problem with the heat exchanger.
455 Therefore, the different efficiencies of heat exchangers were tested by changing the value in
456 Equation (9). The temperature efficiency was set to 0.85 as a reference for both heat exchangers
457 in Fig. 1. To introduce aging and fouling the temperature efficiency was decrease from 0.85 to
458 0.6 [26]. In the analysis, when adjusting the temperature efficiency of one heat exchanger, the
459 temperature efficiency of the other was kept at 0.85.

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

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

469

470 **4. Results**

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

477

478 *4.1. Heating demand*

479 Hourly heating demand data for different customers were obtained from Statkraft Varme
480 AS from the direct measurement at the customers. The heating load data for 2013 are

481 summarized in Table 7. A coincidence factor given in Table 7 is a dimensionless factor
 482 explaining that all the maximum heat loads from different users or buildings do not appear at the
 483 same time. The value of the coincidence factor is between 0 and 1, and usually lower than 1. It is
 484 important to consider coincidence factor for sizing and energy planning to avoid oversizing [38]
 485 In building blocks with different occupants and different life habits, heat use is different.
 486 Therefore, the values of the coincidence factor for the building block are lower than in the other
 487 buildings in Table 7.

488

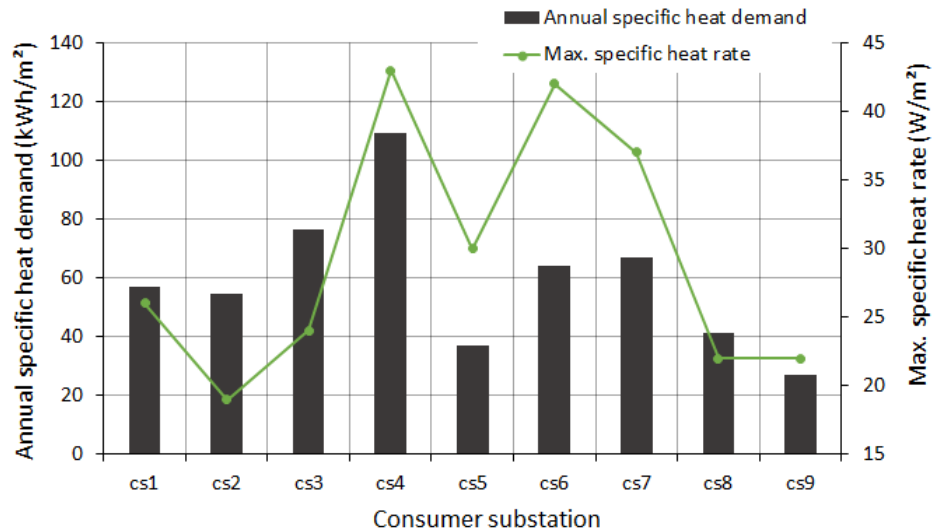
489 Table 7. Specification of energy use for different customers

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

490

491 From Table 7, it can be seen that Building block D showed the highest maximum specific
 492 heat demand and the highest specific annual heat demand in comparison to all the buildings in
 493 spite of the fact that the building was constructed under the passive house standard. The reason
 494 for this could be explained by diverse occupancy patterns or poor operation of the customer
 495 substation. Utilization time in Table 7 describes how long the system should operate with the
 496 maximum heat rate to cover the annual heat demand. The reason why building blocks and
 497 welfare buildings showed a higher utilization time than the office buildings was due to the fact
 498 that the heating system operated longer and the use of the DHW was higher. The other buildings
 499 (office, primary school and sport center) had a high maximum heat demand with lower total heat
 500 use and thereby the utilization time was low. Table 7 shows that the coincidence factor was
 501 higher for buildings where the heat demand was dependent on the outdoor temperature and the
 502 share of the DHW use was in general low. Due to diversities in the heat use, the coincidence

503 factor was lower at the building blocks. A summary of the specific energy demand and annual
 504 heating demand for the customers in Table 7 is shown in Fig. 5. In Fig. 5, it is possible to note
 505 that the substations cs1, cs2, cs6, cs7 and cs8 showed total annual heat demand around 60
 506 kWh/m², however, the maximum specific heat rate (W/m²) was different.

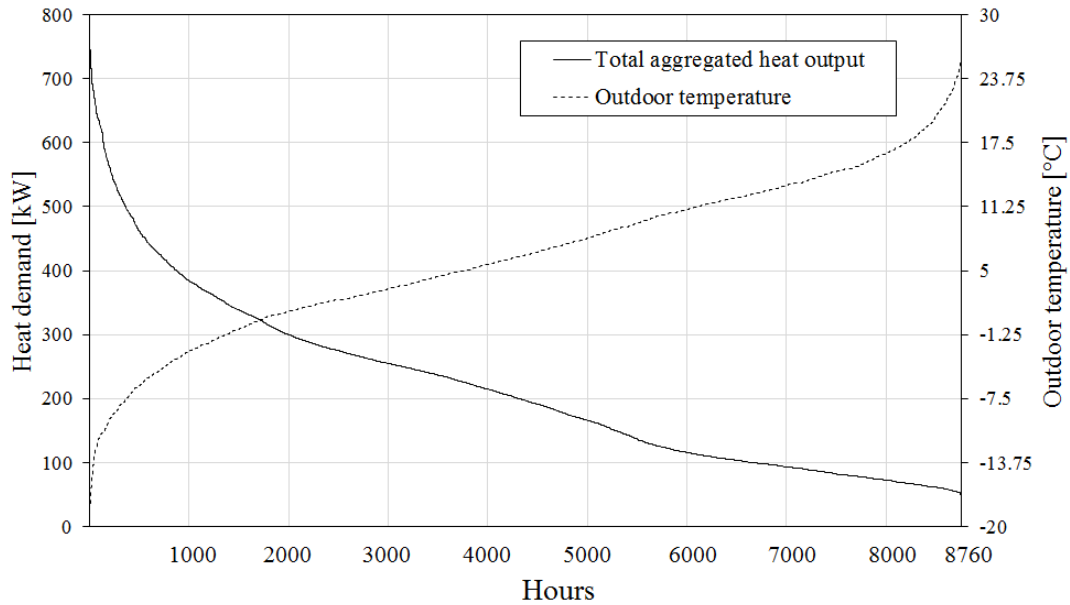


507

508

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

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



514

515

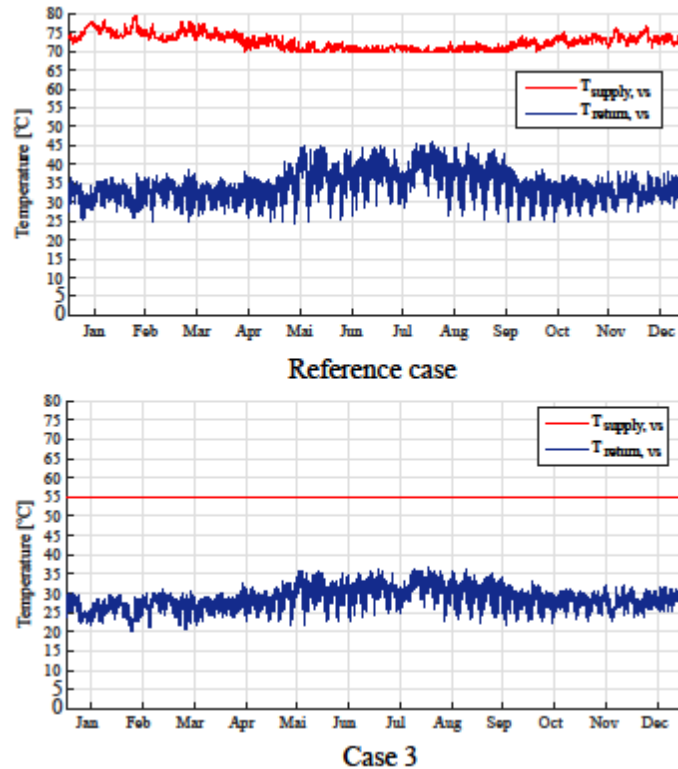
516

Fig. 6. Heat duration year for the reference year

517

518 *4.2. Temperature distribution in the DH grid*

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



526

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

528 From Fig. 7 it can be seen that the analyzed DH system (see Fig. 3 and Section 3) operated
 529 with the lower temperature difference during the summer time due to the lower heat demand.

530 Further, the analysis of the reference case revealed that the average temperature difference was

531 45 K during the winter and 30 – 35 K during the summer. For the LTDH scenario (Case 3) the

532 temperature difference was 30 – 35 K during the winter and 25 K during the summer. To recall,

533 the reference case was designed for the temperature difference of 45 K, while Case 3 (LTDH)

534 was designed for the temperature difference of 30 K, see Fig. 4. All these meant that Case 3 or

535 LTDH managed at some extend better to maintain the design temperature difference over the

536 year, regardless of the change in the heating load. This advantage of LTDH would enable reliable

537 values of the return temperature over the year. This was an important conclusion with focus on

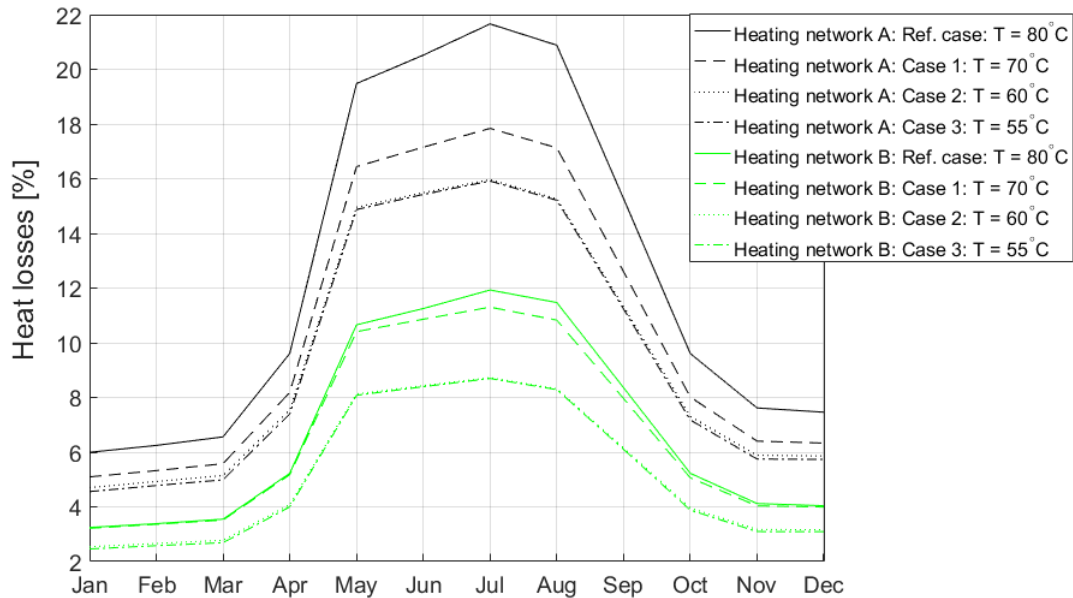
538 the lower return temperature.

539

540 4.3. Heat loss and pump energy

541 The annual distribution of heat losses for the various scenarios in Table 4 are shown in

542 Fig. 8.



543

544

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

545

546

547

548

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.

549

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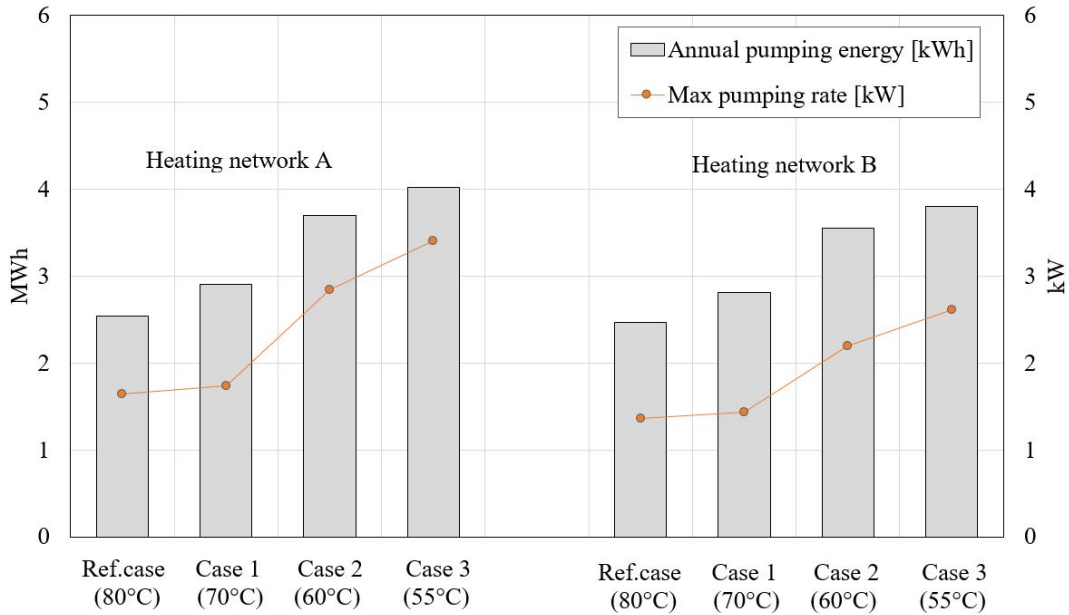
555

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.

556

557

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

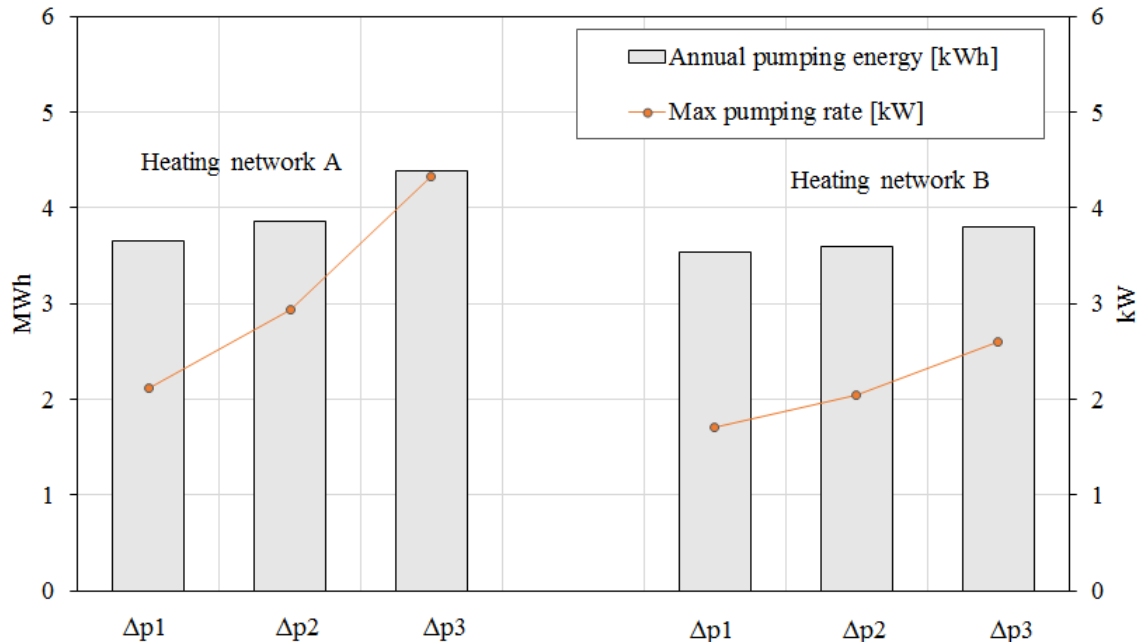


558
 559 Fig. 9 Annual pumping energy and maximum pumping power with various temperature levels in
 560 the existing DH system

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

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

568



569
 570 Fig. 10 Annual pump energy and maximum pump power under different conditions for pressure
 571 drop in the new heating network

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

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

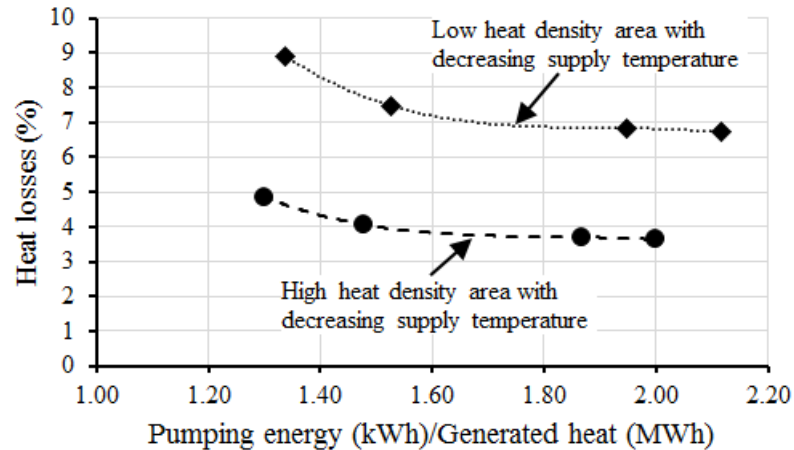
587 B were almost not affected due to increase in R-value. This is because pipe diameters were not
 588 changed significantly with the change of R-value. The pressure drop is approximately
 589 proportional to the pipe diameter in fifth extent and thereby was affected in a higher degree than
 590 the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

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

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

Heating network A - low heat density				
	Reference case (80°C)	Case1 (70°C)	Case2 (60°C)	Case3 (55°C)
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

603



604

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

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

617

618 4.4. Issues with the return temperatures

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

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

625

626 *Fault in the return temperature due to bypasses or short circuits*

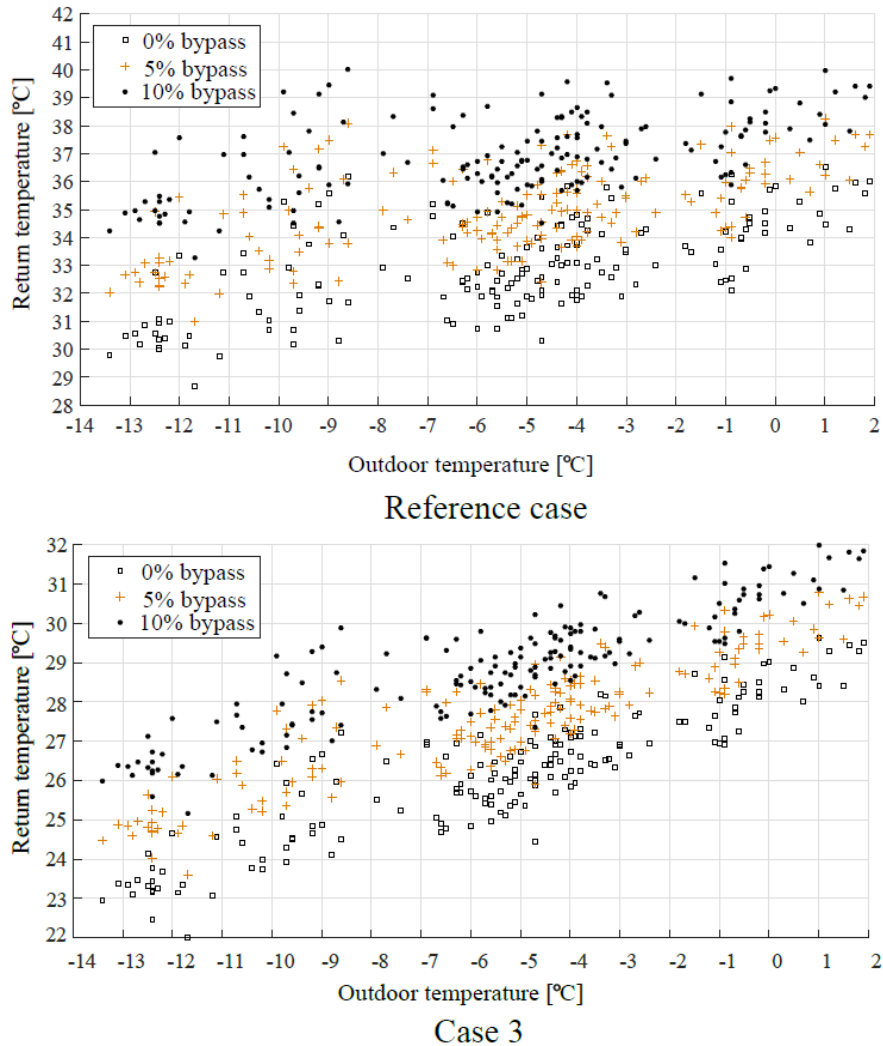
627 Table 9 shows increase in the return temperature during the summer and winter due to the
628 bypass. The simulation implied use of constant percentage of the mass flow rate. The winter
629 period was considered from October to March, and summer period was from April to September.
630 The results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

631 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

632
633 The results in Table 9 show that increase in the average return temperature is higher for
634 the reference case in comparison to Case 3 (LTDH). The findings for 10% could be considered as
635 the most representative, since literature review indicated that this percentage is typical for the
636 Swedish DH system leading to increase in the return by 4 K [4]. As it was explained previously,
637 the bypass is used in the DH systems with low heating densities and high heat losses. This is
638 done to avoid the risk of temperature drop below predefined minimum level. This is particularly
639 relevant during the summer season when the mass flow rate is low and the heat losses are high.
640 For the low energy buildings, the heating season is normally shorter. Thus, the bypass valves can
641 have a greater impact on energy efficiency of LTDH networks associated with the low-energy
642 buildings compared with the traditional DH systems [40]. The results showed that effect of
643 bypassing to the return temperature was less during the summer than during the winter. The
644 return temperature versus the outdoor temperature for different bypassing percentage is shown in
645 Fig. 12.

646



647

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

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

657

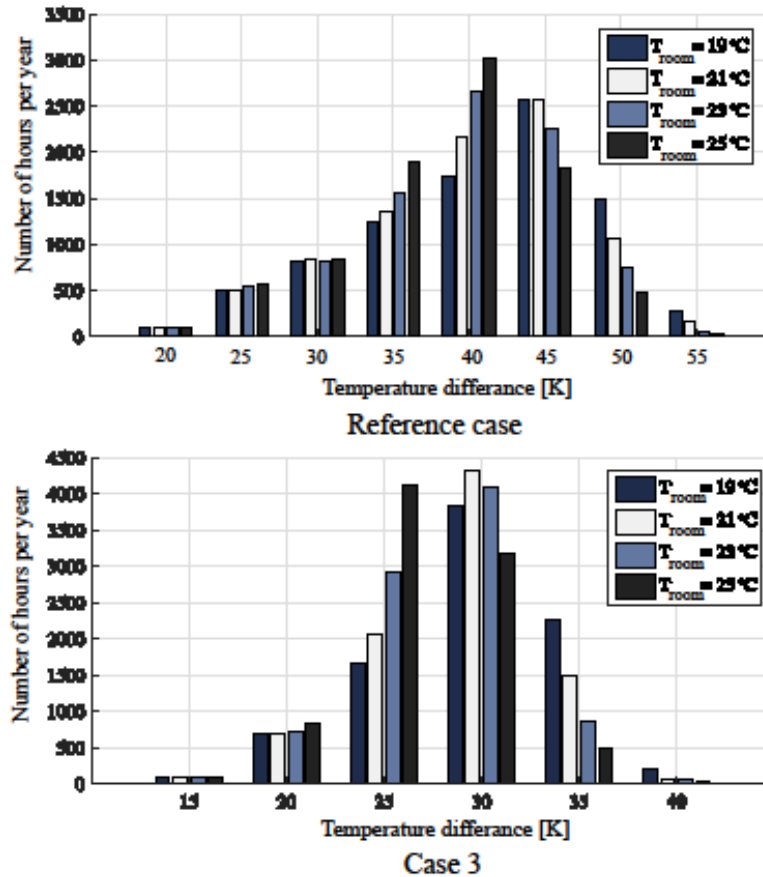
658 *Fault in the return temperature due to aging and fouling of the heat exchangers*

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

666

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

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



679

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

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

692

693

694 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 %

695
 696 The summary results for the entire analyzed network in Table 10 show that both heat loss,
 697 annual pumping energy, and the maximum flow rate increased due to increasing of the set-point
 698 indoor temperature. Further, it can be noted that the annual performance were more sensitive on
 699 the increase in the set-point indoor temperature for Case 3 than for the reference case.

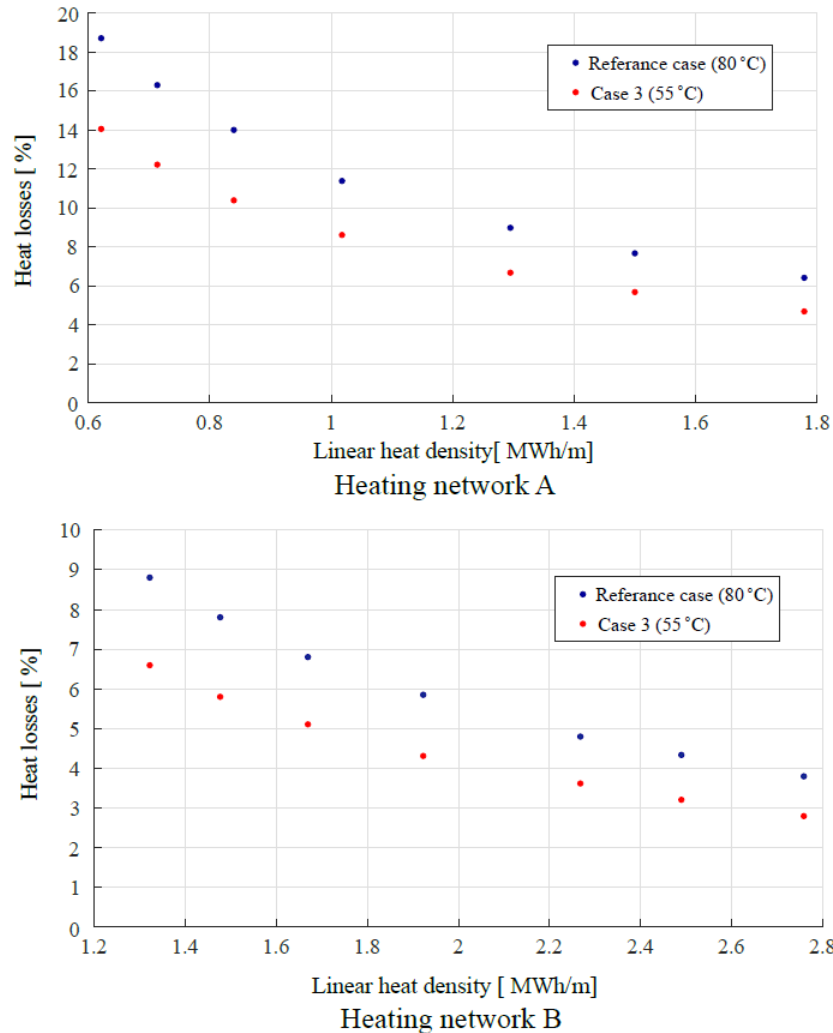
700
 701 *Faults in the return temperature due to fail adjustment of the outdoor compensation curve*

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

708
 709 *4.5. Competitiveness of LTDH in the low heat density areas*

710 The long-term program aiming improvement in the building energy efficiency will
 711 facilitate decreasing of the linear heat densities (MWh/m) and heating densities (MWh/m²). This
 712 could create new challenges for LTDH, since the profitability of the DH system is dependent on
 713 high demand and high linear density. Therefore, it was of interest to examine how linear densities
 714 affect heat losses and pumping energy in the heating networks A and B. Different linear heat
 715 densities were introduced by changing the length of the networks given in Fig. 3. Fig. 14 shows
 716 relative heat losses under various linear densities for heating networks A and B.

717

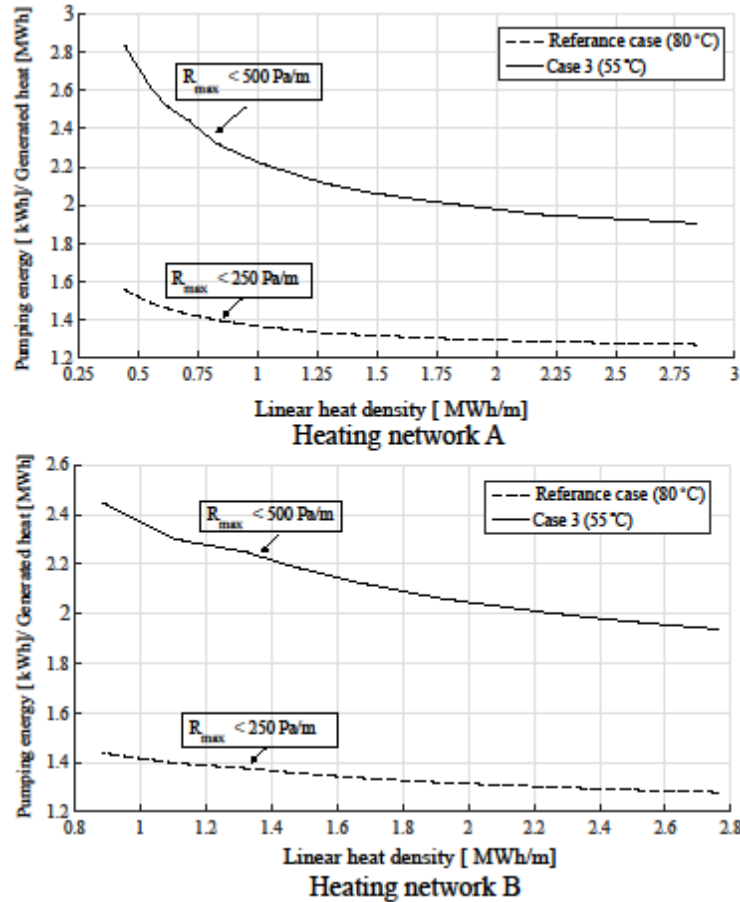


718

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

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

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



728

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

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

737

738 5. Discussion

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

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

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

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

770 return temperature due to various substation design, a further analysis of the heat transfer
771 coefficient and the heat transfer surface should be made.

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

779

780 **6. Conclusions**

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

787 The transition of the existing DH system was planned in two ways: 1) decreasing the
788 supply water temperature without change in the pipe size; 2) sizing the DH network based on the
789 LTDH requirements. For the existing DH system, the design pressure drop was in the range of 50
790 - 250 Pa/m and 80°C for the supply temperature. The results showed that the pressure drop
791 doubled with the reduction of the return temperature from 80°C to 55°C, and thereby the
792 pumping power and energy increased. This increase was lower for the grid with the high heat
793 density due to the shorter DH network. Further, it was found that the transition to LTDH
794 facilitates the reduction of the heat losses up to 25 %. A large scale DH system would show
795 higher heat losses due to versatile network structure with different linear densities. The trade-off
796 analysis between the heat losses and the pump energy use showed that considerable amount of
797 the heat would be saved by decreasing the supply temperature, while the increase in the annual
798 pump energy use would not be high.

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

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

826 Several assumptions were made under the study of LTDH. Therefore, it is important to
827 have a critical look on the findings. All the mentioned findings should be considered carefully,
828 because the benefits of lowering the temperature in the DH system could be mistreated due to
829 aforementioned issues. Nevertheless, the results provided a clear picture of the improvement in

830 energy efficiency of the DH network due to lowering the temperature level and huge potentials
831 and challenges related to LTDH.

832

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838

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