Heating and Domestic Hot Water Systems in Buildings Supplied by Low-Temperature District Heating

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Heating and Domestic Hot Water Systems in Buildings Supplied by Low-Temperature District Heating

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PREFACE AND THESIS OUTLINE

This thesis is submitted as a partial fulfilment of the requirements for the Degree of Doctor of Philosophy at the Technical University of Denmark, Department of Civil Engineering.

The thesis is conceptually divided into two parts. Part I is dedicated to the delivery of DHW supplied by low-temperature district heating, including research on an energy efficient bypass solution. Part II is focused on the feasibility of supplying space heating systems in existing and low-energy buildings using low-temperature district heating. Each part has own specific background, methods and results and discussion, but they share a common introduction, hypothesis, conclusions and suggestions for further work.

The thesis is based on the following three ISI articles, corresponding to three individual sub-hypotheses. The thesis reports only the main findings and the full-length articles can be found in the Appendix.

The first article focuses on challenges and their solutions related to the heating of domestic hot water by low-temperature district heating with a supply temperature of 50°C.


The first paper argues that the service pipes of low-temperature district heating substations based on the instantaneous principle of DHW heating need to be kept warm by using a bypass solution to ensure fast provision of DHW. The second paper therefore investigates the feasibility of redirecting the bypass flow to the bathroom floor heating to reduce the heat loss from the network while making good use of the “waste” heat to take the chill off the floor in bathrooms.


The third paper investigates the feasibility of connecting a single-family house from the 1970s to low-temperature district heating and answers the question of how much and for how long a period does the supply temperature of low-temperature DH need to be increased above 50°C. The house represents typical example from the Danish building stock and the investigation shows the advantage of combining energy-saving measures with the implementation of renewable-energy based heat supply at the same time.

- Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment, Brand M, Svendsen S. accepted for Energy in September 2013.
ACKNOWLEDGEMENTS

This thesis is a result of three years of research performed at the Technical University of Denmark at the Department of Civil Engineering’s Section of Building Physics and Services, additionally extended by six months’ work on the relevant research project with industrial partners and consultant companies.

I want to thank to all partners from COWI, Danfoss A/S, Danish Technological Institute, Grontmij and the many others I met during this work for accepting me as full-value team member and helping me orient in the world of district heating. I would also like to thank my main supervisor, Professor Svend Svendsen, for his guidance through my entire PhD project, and my co-supervisor Bjarne W. Olesen for the opportunity to discuss various scientific questions.

I am very grateful to Danfoss A/S for their partial financial support of my studies. A special thanks is due to my other co-supervisor Jan Eric Thorsen from Danfoss A/S for giving me the benefit of his huge knowledge and always having a positive and kind approach. Thanks also to the Strategic Research Centre for Zero Energy Buildings for their partial financial support and for giving me the opportunity on a regular basis to discuss the work for my PhD project in the perspective of other colleagues from the centre. And thanks also to Janusz Wollestrand and Patrick Lauenburg from Lund Technical University for accepting me as a part of their research group during my half-year external stay at Lund Technical University and for sharing their knowledge about district heating with me.

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Many thanks also to my family and friends in the Czech Republic for being there for me during my frequent visits, but also to my “Danish family”, consisting of many extraordinary people who have created such a wonderful environment for me that I have never felt that I am 800 km from my home country.

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My last and greatest thanks are addressed to my aunt Lida, for her endless support in many difficult life situations. She has contributed a great deal to making my Ph.D. studies possible and therefore this thesis is dedicated mainly to her.

Lyngby, 31st August 2013

Marek Brand
ABSTRACT

District heating (DH) systems supplied by renewable energy sources are one of the main solutions for achieving a fossil-free heating sector in Denmark by 2035. To reach this goal, the medium temperature DH used until now needs to transform to a new concept reflecting the requirement for lower heat loss from DH networks required by the reduced heating demand of low-energy and refurbished buildings combined with the lower supply temperatures required by using renewable heat sources. Both these needs meet in the recently developed concept of low-temperature DH designed with supply/return temperatures as low as 50°C/25°C and highly insulated pipes with reduced inner diameter. With this design, the heat loss from the DH networks can be reduced to one quarter of the value for traditional DH designed and operated for temperatures of 80°C/40°C. However, such low temperatures bring challenges for domestic hot water (DHW) and space heating (SH) systems, from the perspective of both DH customers and the DH company. The aim of this work was therefore to identify, evaluate and suggest solutions.

The first part of the research focused on the feasibility of supplying DHW with no increased risk of Legionella and on the performance of low-temperature DH substations.

The Danish Standard DS 439 for DHW requires that DHW should be delivered in reasonable time, without unwanted changes in desired temperatures (comfort) and without increased risk of bacterial growth (hygiene). While the comfort requirements set the minimum DHW temperature to 45°C, the hygiene requirements set it to 60°C, which is simply not reachable for low-temperature DH. However, the German DHW standard DVGW 551 makes no requirement about minimum DHW temperature if the overall DHW volume is below 3L. This rule was adopted as a cornerstone for the research and for the whole low-temperature DH concept in general, so the minimum DHW temperature is defined by a requirement for 45°C at the kitchen tap.

The performance of a low-temperature DH substation with instantaneous DHW preparation was evaluated based on the results from laboratory measurements supplemented with results from the verified numerical model developed in MATLAB-Simulink. The laboratory measurements showed that the low-temperature substation can heat the required flow of DHW to 47°C with 50°C DH water while keeping the return temperature as low as 20°C. The results of numerical simulations considering the influence of the DH network, represented by a 10 m long service pipe connection for the substation equipped with an external bypass with a set-point temperature of 35°C, showed that the time needed to produce 40°C DHW was 11 s with and 15 s without the external bypass, respectively. DS 439 suggests 10 s as the reasonable waiting time for DHW, so a low-temperature DH substation based on the instantaneous principle of DHW preparation should be equipped with bypass solution keeping the service pipe warm and reducing the waiting time.

Traditional bypass solutions simply redirect the bypassed water back to the DH network without additional cooling, but bypassed water can instead be redirected to floor heating in the bathroom to be further cooled and thus reduce heat loss from the DH network while improving comfort for occupants and still ensure fast DHW preparation. Various solutions for the redirection and control of bypass flow were developed and their detailed performance tested on the example of a low-energy single-family house modelled in building energy performance simulation tool IDA-ICE 4.22. The effect on the DH network was simulated with the commercial program Termis on a case study of 40 single-family houses supplied by low-temperature DH. In comparison to the reference case with a traditional external bypass, the proposed solution resulted in
average cooling of bypassed water by 7.5°C, reducing the heat loss from DH network during non-heating period by 13% and increasing the average floor temperature by 0.6-2.2°C without causing overheating. The price for heating the bathroom floor during the non-heating period depends on the location of the house and was between 98 and 371 DKK/house, but it seems reasonable to bill all customers with an even and discounted price, reflecting the fact that 40% of the heat delivered to the bathroom floor is covered by reduced heat loss from the DH network.

It can be concluded that low-temperature DH with a supply temperature low as 50°C can be used for the delivery of DHW with the desired temperature and without increased risk of Legionella if the DH substation and DHW system are designed for the low-temperature supply conditions. To ensure the fast provision of DHW during non-heating periods, the supply service pipe should be kept warm, preferably with the bypass solution redirecting the bypass flow to bathroom floor heating and thus at least partly exploiting the additional heat loss caused by keeping the DH network ready to use.

The second part of the work focused on SH systems in low-energy and existing buildings supplied by low-temperature DH.

The feasibility of supplying existing buildings with low-temperature DH was investigated using the IDA-ICE program by modelling the example of single-family house from the 1970s, representing a typical example of Danish building stock. The results show that, to maintain the desired indoor temperature and not exceed the originally designed flow rate from the DH network, the DH supply temperature would need to be increased above 50°C in cold periods. In its original state, the house would need to be supplied with a DH temperature above 50°C for 21% of the year and above 60°C for 3% of time, with the highest temperature being 73°C. But if the windows are replaced, which can be expected because their lifetime is coming to an end, the maximum supply temperature is reduced to 62°C and the periods are reduced to 7% and 0.2% respectively. Further improvements, such as the addition of ceiling insulation or the installation of low-energy windows and low-temperature radiators, will allow DH water supply at 50°C the whole year around. The results show that supplying existing buildings with low-temperature DH is not a serious problem and that DH companies should be stricter in reducing the supply temperature, which is very often kept high just because of the malfunctioning of the in-house systems of customers. Moreover DH companies should require that all newly installed and refurbished DH substations should be designed for low-temperature DH to ensure the gradual transition to a temperature level of 50°C in the shortest possible period.

The IDA-ICE program was also used to model the performance of a space heating system with radiators in the low-energy single-family house. The space heating system was investigated from the perspective of the customer, represented by thermal comfort, and the DH company, represented by a smooth heat demand and low return temperature. To accord with the literature, the modelling of internal heat gains reflected the improved efficiency of equipment by reduction of value from 5W/m² to 4.2W/m², also modelled as intermittent heat gains based on a realistic week schedule. Furthermore, the indoor set-point temperature was increased from 20°C to 22°C to reflect a temperature level preferred by occupants. The results showed that an SH system with radiators can provide the desired indoor temperature while ensuring a smooth heat demand from the DH network and proper cooling. However, using input values suggested by the literature leads to up to 56% greater heat demand than values suggested in the Danish national calculation tool Be10, and in 40% lower connection power than for an SH system dimensioned in accordance with DS 418. Use of Be10 input data in cost-effectiveness analyses for DH networks therefore means worse results, because less
heat is sold to customers and there is higher heat loss in the network. Similarly, higher connection power than needed means bigger pipe diameters are needed, resulting in higher heat losses as well. Using realistic values is therefore very important for feasibility calculations of DH.
RESUMÉ


Første del af arbejdet fokuserer på mulighederne for forsyning af varmt brugsand uden øget risiko for legionella bakteier, samt funktionalitet og performance af lav temperatur fjv. stationen.

Af Dansk Standard DS 439 fremgår at varmt brugsand skal leveres indenfor acceptabel tid (komfort) og uden uøndvendig temperatur variationer (komfort) samt uden øget risiko for vækst af legionella bakteier (hygiejne). Mht. komfort kræves en brugsands temperatur på 45°C. Mht. hygiejne kræves en temperatur på 60°C, hvilket ikke umiddelbart er foreneligt med lav temperatur konceptet. For dette arbejde er der taget udgangspunkt i det tyske regelsæt for vandforsyning, DVGW 551, som under forudsætning af mindre end 3 liter volumen i varmt vands systemet (efter varmeveksleren) ikke stiler krav til temperaturen ift. hygiejne. Temperaturkravet er således bestemt af komfortkravene i DS439, dvs. 45°C til rådighed for kækkenvasken.

Funktionalitet og performance for lav temperatur fjv. stationen er blevet analyseret vha. udviklede dynamiske Matlab-Simulink modeller og verifieret op imod laboratorier målinger. Disse viste, at der ved specificeret flow kunne opnås en brugsandstemperatur på 47°C ved 50°C forsyningstemperatur, med en tilhørende retur temperatur på 20°C. Resultaterne fra den dynamiske model viste, at det er muligt at komme fra system tommag (ingen tapning over længere tid, 10 m delvis nedkøl stikledning, by-pass temperatur på 35°C og ingen varme behov) til 40°C brugsandstemperatur på 11 sekunder. Ifølge DS 439 betragtes en passende ventetid på varmt brugsand som 10 sekunder, hvilket derved i praksis er overholdt ved anvendelse af by-pass (varmholdning af stikledninger).

Traditionel by-pass fungerer ved at en termostat styrer ventil holder fjv. vandet på et passende temperatur niveau ved fjv. stationen. Derved bliver fjv. vandet dog ikke afkølet, hvorved det ledes tilbage til fjernvarme nettet ved unødvendig høj temperatur, med højere termisk net tab til følge. I stedet foreslås at fjv. vandet ledes igennem gulvarmen for badeværelset, hvorved afkølingen opnås, inden vandet ledes tilbage til fjernvarme værket. Derved reduceres det termiske net tab. Yderligere reduceres ventetiden for det varme brugsand, idet fjv. stationen bliver forsynet umiddelbart med varmt fjernvarmevand også uden for perioder med tapning af varmt brugsand. Forkellige løsnings forslag for by-pass er blevet analyseret og valideret vha. IDA-ICE simuleringer værktøjet for bygningens vedkommende. For fjv. nettet er TERMIS blevet anvendt for analyserne. Sammenlignet med traditionel by-pass er der eftervist gennemsnitligt 7,5°C reduceret afkøling af by-pass vandet, hvilket reducerer fjernvarmennettets varmetab med 13% i perioder hvor der ikke kræves varme i bygningen. Den gennemsnitlige gulv temperatur blev forøget med 0,6 – 2,2°C, dog uden at dette leder til overhedning af badeværelserne. Opvarmningen af badeværelses gulvene over
sommersen er beregnet til at koste 89 – 371 Kr./hus, hvilket skal ses i lyset i at 40% af denne energi ellers ville være tabt i fjv. forsynings nettet, og denne del ville forbrugerne skulle betale under alle omstændigheder, idet nettabet fordeles kollektivt forbrugerne imellem.

Det kan konkluderes, at lav temperatur fjv. med en forsyningsstemperatur på 50°C kan anvendes til at producere varmt brugsvand ved passende temperaturer, med acceptable ventetider og uden øget risiko for legionella under forudsætning af at fjv. stationen tilgodeser de specifikke krav herfor. Ventetiden reduceres yderligere ved at anvende by-pass gennem gulvarmen for badeværelset i sommer perioden for derved at holde stikledningen varm og udnytte en del af den energi som ellers ville være gået tabt fjv. nettet.

Anden del af arbejdet har fokuseret på varmesystemet for lav energi bygninger og eksisterende bygninger med henblik på at analysere muligheden for at forsyne disse med lav temperatur fjv.

Muligheden for at forsyne eksisterende bygninger med lav temperatur fjv. er blevet analyseret vha. IDA-ICE programmet. Typiske én-familie huse fra 1970 er taget i betragtning, idet de repræsenterer en væsentlig del af bygningsmassen i Danmark. Simuleringsresultater viser, at for at kunne holde den ønskede rumtemperatur, kræves en højere forsyningsstemperatur end 50°C for de koldeste perioder. For det originale hus fra 1970 kræves en forsynings temperatur på over 50°C i 21% af tiden og over 60°C i 3% af tiden. Den højeste nødvendige fjv. temperatur er 73°C. Energienoveres med nye vinduer, hvilket er oplagt idet de eksisterende har udstået deres levetid, bliver den maksimale forsyningsstemperatur reduceret til 62°C og varigheden for temperaturer over 50°C hlv. 60°C er 7% og 0,2%. Yderligere tiltag, som isolering af loftet eller udskiftning af vinduer sammen med radiatorer dimensioneret til lav temperatur drift medfører at 50°C forsyningsstemperatur kan anvendes hele året rundt. Derudover burde fjv. værkerne stille krav om at der for om- og nybygning kræves at bygningen forberedes til lav temperatur drift.

Detaljerede analyser omkring varmesystemet blev udført for lav energi én familie huse, fokuserende på brugeren, mht. termisk komfort, og fokuserende på fjv. værket, mht. jævn energi belastning og lav retur temperatur. Resultater viser at ønsket rum temperatur sammen med en jævn varme belastning kan opnås.

Et mere realistisk varme grundlag fremkommer ved at anvende varmetilskid 4,2W/m² i stedet for 5,0 W/m² som typisk fremgår af litteraturen. Derudover afspejler en indendørs temperatur på 22°C, i stedet for den typiske anvendte værdi på 20°C, et mere retilvisende billede mht. energiforbrug. Anvendes antagelserne fra litteraturen, fås 56% større beregnet varmebehov sammenlignet med Be10 beregningerne og 40% lavere forbrug sammenlignet med dimensionering iht. DS 418. Ved anvendelse af Be10 beregninger vil varmegrundlaget blive for lavt, hvad er en udfordring for fjv. konceptet, idet forholdet mellem leveret (nyttiggjort) og tabt varme bliver forringet. Omvendt hvis varmegrundlaget bliver estimeret for højt, j.f DS 418, har det den konsekvens for fjv. nettet at rørdimensionerne bliver for store, med deraf følgende øget varmetab.
LIST of PUBLICATIONS
Publications included in the thesis

Article I

Article II

Article III
Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment, Brand M, Svendsen S. accepted to Energy in September 2013.

Publications and work not-included in the thesis
The list below reports additional research work not included in the thesis either because I was not the main author or because the topic is already covered by the articles included in the thesis.

PEER-REVIEWED INTERNATIONAL ARTICLES

RESEARCH PROJECTS

- EUDP 2008 “CO₂-reductions in low energy buildings and communities by implementation of low temperature district heating systems. Demonstration cases in Boligforeningen Ringgården and EnergyFlexHouse”
  - chapter in report part 2 (detailed study about waiting time for DHW)
  - task manager for part 3 (comfort bathroom concept, floor heating)
  - application for International Energy Agency award
  - contribution with the above mentioned Article II and Article III
- EUDP 2011 “Heat pumps for heating of DHW supplied by low-temperature district heating”
  - contribution to the report with calculation of heat demand for the example of typical low-energy buildings
- Svensk Fjärrvärme “Nästa Generations Fjärrvärme” (Next Generation District Heating)“
  - contribution to the report with calculations and text about use of different space heating systems in low-energy buildings supplied by low-temperature district heating
- Brochure of the Strategic Research Centre for Zero Energy Buildings – parts 1 and 3
  - contribution to the information booklet 1 “Definition and Role in Society” and booklet 3 “Design Principles, Design Guidelines and Built Examples
NOMENCLATURE

List of abbreviations

CB   comfort bathroom
ECL  electronic controller from Danfoss
DHW  DHW circulation
DHWCU DHW storage unit
DH   district heating
DHGSU district heating storage unit
SUB  district heating substation
DHW  domestic hot water
FH   floor heating
FF   fossil free
HEX  heat exchanger
IHEU substation based on instantaneous principle of DHW heating
      (instantaneous heat exchanger unit)
LTDH  low-temperature district heating
PTC2+P proportional-thermostatic controller for DHW from Danfoss
SP   service pipe - pipe connecting street pipe with DH substation
SH   space heating
TRV  thermostatic regulation valve
FJVR  TRV controlled by the fluid temperature

List of Symbols

\( \dot{m} \) [kg/s] mass flow
\( T_{\text{bypass}} \) [°C] bypass set point temperature
\( T_{\text{floor}} \) [°C] floor temperature
\( T_{\text{op}} \) [°C] operative temperature
\( T_{\text{ret}} \) [°C] return temperature
\( T_{\text{soil}} \) [°C] temperature of soil
\( \Delta T_{DB} \) [°C] Deadband
\( P_{\text{max}} \) [kW] maximum heating power
\( TRW \) [°C] weighted average return temperature
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1 INTRODUCTION
To reduce CO₂ emissions and increase the security of supply, in 2011 the Danish Government decided [1] to achieve a fossil-free heating and electricity sector for buildings by 2035 and complete independence of fossil fuels by 2050. The Energy Performance of Buildings Directive (EPBD) [2] requires that all new public and other buildings should be constructed as nearly-zero energy buildings [3] from 2018 and 2020 respectively. The Danish national heating plan [4] judges that this will be achieved mainly by a further spread of district heating (DH) based on renewable heat sources (RES). The most cost-effective use of these sources is related to their efficiency [5], so the DH supply and return temperatures should be as low as possible. This is also required by the need to reduce the heat loss from DH networks, which will make it economically possible to supply buildings with reduced heating demand, such as low-energy and refurbished existing buildings, which it would be uneconomical to supply with traditional medium temperature DH. To reflect these needs, the concept of low-temperature DH with supply/return temperatures of 50/25°C respectively (see Figure 1.1), matching the exergy levels of supply and demand sides [6], has recently been developed and successfully tested in a settlement of low-energy houses [7]. The deployment of low supply/return temperatures and DH pipes designed with smaller diameters and greater insulation thicknesses reduces the heat loss from the network to one quarter of the heat loss expected from a traditionally designed and operated DH network with 80/40°C [8]. However, the reduced supply temperature and focus on energy efficiency brings some new aspects which still need to be addressed in relation to both main tasks for DH, i.e. domestic hot water (DHW) and space heating (SH).

![Figure 1.1 – Concept of low-temperature district heating](image)

The fact that the DH supply temperature is as low as 50°C puts focus on the risk of Legionella in the DHW system and poses the question of whether the DH substation can heat DHW to the desired temperature of 45°C within reasonable time and provide the desired cooling of DH water. With regard to SH systems, we need to distinguish between low-energy and existing buildings. For new low-energy buildings, reduced DH supply temperatures do not represent any serious problem because their low heat demand allows the design of SH systems with low supply/return temperatures of 50/25°C. However, low-energy buildings comprise, and for some more years will continue to comprise, only a small share of the building stock, while 85-90% are older buildings with considerably greater energy demands [9], heated by radiators designed for a supply temperature of 70°C or higher. Reducing the DH supply temperature to 50°C would therefore cause thermal discomfort for their occupants and an undesirably high return temperature and flow to the DH network. However, these two types of buildings are geographically mixed, so the feasibility of supplying both types by the same DH network should be investigated.
1.1 Objective of Research
The goal of the project was to identify the challenges related to using low-temperature district heating with supply temperatures as low as 50°C in space heating and DHW systems and to suggest solutions. The research took into consideration both low-energy buildings with reduced heating demand and existing buildings with high heating demand.

1.2 Scope
DH systems are very complex; they consist of the heat production side, represented by heat plants, the heat demand side, represented by the buildings, and the heat transmission systems transferring the heat between them by means of DH water flowing in DH pipes. This thesis focuses on the heat demand side represented by in-house SH and DHW systems, on the DH house substation as their common interface to the DH network, and on their interaction with the rest of the DH system.

The broad nature and complexity of the topic requires some assumptions and limitations, which can be summarised as:

- The work focused mainly on the in-house DHW and SH systems, including the DH substation
- A maximum volume of 3L for the DHW system is assumed, so there is no requirement for a minimum temperature of DHW to deal with Legionella
- Existing buildings are defined as buildings built after 1970 in accordance with Danish building regulations
- The research did not examine the heat production side or the heat transmission system, but only uses results from recently published studies
- The research did not develop refurbishment solutions to reduce heat demand, but only uses results from recently published studies.
1.3 Hypothesis
The main hypothesis of this research was:

It is possible to decrease the district heating supply temperature to 50°C and operate a district heating network with a reasonable flow and cooling of district heating water, and maintain desirable indoor temperatures and fast delivery of domestic hot water, without increased Legionella risk, for both existing and low-energy buildings.

The main hypothesis can be divided into four sub-hypothesis (see Figure 1.2):

1) With district heating substations specially designed for low-temperature operation, it is possible to reduce the district heating supply temperature to 50°C and still provide domestic hot water at the required temperature level, without increasing the waiting time for domestic hot water and without increasing the risk of Legionella, and at the same time ensure a return temperature of district heating water as low as 20°C.

2) Bypass flow redirected during the non-heating period into bathroom floor heating is additionally cooled in comparison with traditional bypass solutions and thus reduces the heat loss from the district heating network while giving occupants the sensation of a warm floor at a discounted price.

3) Existing buildings can be supplied with low-temperature district heating systems designed for a supply temperature of around 50°C if the district heating supply temperature is increased during very cold periods and DHW substations are changed.

4) The space heating system in a low-energy building supplied with low-temperature district heating at 50°C can provide the desired indoor temperature and maintain a smooth load on the district heating network and a low return temperature.
Figure 1.2 – Relation between main hypothesis and sub-hypothesis
2 BACKGROUND

2.1 Energy Supply - Situation and Political Decisions
With regard to the energy sector, the EU is currently facing two main challenges. The first one is the climate change caused by the emission of considerable amounts of CO₂ from burning fossil fuels, and the second one is security of supply connected with the import of fuels mainly from non-EU countries and their increasing price caused by their diminishing reserves. Buildings in the EU account for approx. 40% of the total energy use [2], so reducing energy used mainly for SH and heating of DHW will contribute significantly to improving the situation. In 2010, therefore, the EU commission issued a recast version of the Energy Performance of Building Directive (EPBD) requiring all member states to implement in their national building codes the requirement that all new buildings built after 2020 should meet high energy-saving standards [2]. However, this is just the first step. The second step is to become completely independent of fossil fuels and base energy supply purely on renewable energy sources (RES).

So every EU country had to prepare a national plan for including more RES in the national energy system. This was also the main objective of the research study, Heat Plan Denmark 2010 [4]. The study suggested Denmark should become completely fossil-free by 2050, but the deadline for the heating and electricity sector has been brought forward to the year 2035. This is to be achieved by energy savings in buildings, improved efficiency on the energy production side, and further expansion of DH to neighbouring areas increasing the DH share of heat delivery from 50% (in 2010) up to 70%. The heat sources for the DH are expected to be centralised heat pumps, solar thermal heat plants, and geothermal heat plants with/without heat pumps. The remaining 30%, mainly in areas with low heat demand, is to be covered by individual heat pumps.

To make the energy sector fossil-free on time, the Danish Government is supporting a lot of research activity via its Energy Technology Development and Demonstration Programme (EUDP) [10], where complete list of the projects can be found. There is on-going work on energy savings in both low-energy buildings and the refurbishment of existing buildings, e.g. at the Strategic Research Centre for Zero Energy Buildings (ZEB) [11] [12] [13]. With regard to the further development of DH, the low-temperature DH project in Lystrup [14] [15] [7] and the implementation of heat pumps to low-temperature DH [16] should be mentioned.

2.2 Heat Supply in Denmark
In 2010, DH covered 50% of total heat demand (63% of households) in Denmark, and it has a long tradition from the beginning of the 20th century [17]. But it was the oil crisis in the 1970s that really boosted its expansion. At that time, the political problems in the Middle East resulted in disruption of oil supplies and steep increases in oil prices. The reason for the extensive expansion of DH was the higher efficiency of centralised heat sources compared to individual boilers, saving fuel and thus also money. Furthermore, DH made it possible to use waste heat, e.g. from the production of electricity – known as combined heat and power (CHP) – or from industrial processes, and raised the possibility of burning communal waste to get energy instead of using landfill. For the same reasons, DH is considered an environmentally friendly heating solution because the heat is produced with lower or without CO₂ emissions.

DH can be also defined as a heating system with high flexibility with regard to heat sources. And this is exactly what is needed to achieve a 100% fossil-free heating sector by 2035. The integration of fossil-free
and renewable energy sources on a large scale is easier, cheaper and faster than changing individual heat sources. Figure 2.1 shows that in 2011 fossil-free heat sources contributed 52% of the total fuel mix in Danish DH: 32% from biomass, 19% from waste, and less than 1% from solar and thermal heat sources.

Figure 2.1 – left: Share of DH in heat delivery in Denmark 2012; right: Share of heat sources in DH for 2011 [18]

In 2010, the average DH supply ($T_{\text{sup}}$) and return ($T_{\text{ret}}$) temperatures were:

- heating season $T_{\text{sup}}$ 78.7°C and $T_{\text{ret}}$ 41.4°C
- non-heating season $T_{\text{sup}}$ 73.3°C and $T_{\text{ret}}$ 44.1°C

Figure 2.2 (left) shows the development of fuel source shares in DH between 1994 and 2012. The share of fossil fuels has continuously decreased while the share of non-fossil fuels has increased. Figure 2.2 (right) shows the price development of oil, natural gas, wood pellets, wood chips and straw for DH companies, supporting the need to drop fossil fuels due to increasing price, as well as their environmental impact.

Figure 2.2 – left: Development of heat source shares in DH in the period 1994-2012 [19]; right: Development of fuel prices for DH in the period 1997-2011 [18]

However, bio fuels are not seen as a long-term solution, because Denmark’s own production cannot cover the fuel demand. Moreover, biofuels will be needed for fuel production for the transportation sector after 2035. So, energy for DH will have to come from “non-burnable” RES, such as geothermal and solar heat, possibly in combination with large-scale heat pumps.
2.3 Low-Temperature DH - Definition and Justification

To further expand and operate DH systems in a cost-efficient way, the following should be considered:

- The increasing number of low-energy and refurbished buildings with reduced heat demand
- RES of heat as the only heat sources after 2035
- The heat supply of buildings that were originally designed for medium supply temperatures.

The heat demand for SH in low-energy or refurbished existing buildings is low in comparison with typical existing buildings, but the absolute heat loss from the DH networks remains the same. This means that the ratio between the heat loss from the DH network and the heat used by the customers increases and heat loss represents a bigger portion in the heating bill, reducing cost-effectiveness. So reducing heat losses in the DH network is one of the key issues for future DH heating.

The heat loss from the network could be reduced by the physical improvement of DH pipes, starting with increasing their thickness and improving the properties of the insulation material, continuing with the integration of supply and return media pipes into one casing (twin pipes [20]), and ending with the reduction of pipe diameter based on the optimisation method to exploit all the differential pressure available on the way to the individual customer [21]. Heat loss from the DH network is also proportional to the difference between the supply temperature and the temperature of the surrounding ground, so lower supply temperature means lower heat loss.

The DH supply temperature can be reduced only to the level that still guarantees:

- The delivery of DHW with the required temperature of 45°C DHW at the tap [22]
- The design indoor temperature in the buildings, usually 20°C of operative temperature [23]

With regard to state-of-the-art technology, specially developed low-temperature heat exchangers (HEX) with a logarithmic mean temperature difference (LMTD) of 6.5°C can produce 45°C DHW from 50°C hot water on the primary side while ensuring the desired cooling of primary water to 20°C. Such HEXs are applied in low-temperature DH substations [24] [25] (discussed later in section 3.1.2). Furthermore, reducing the DHW temperature to 45°C reduces heat losses from DHW pipes and storage tanks and thus increases the efficiency of heat sources. It is estimated that reducing DHW temperature from 60°C to 45°C increases the efficiency of solar collectors by 10% and the coefficient of performance factor of HP by 30% [26].

Low-temperature DH is a concept mainly for low-energy and refurbished buildings. Space heating systems in both types of building can be designed with a supply temperature of 50°C (defined by DHW requirements) and a return temperature of 25°C, i.e. a cooling of 25°C.

The low-temperature DH concept can therefore be characterised as a DH concept with:

- Heat loss reduced by using twin pipes with at least class 2 insulation and reduced media pipe diameter
- A design supply temperature of 50°C and a design return temperature of 25°C, with the option of higher temperatures during peak periods
The cost-effectiveness of the low-temperature DH concept was proved first in a theoretical study of a settlement with 92 single-family houses (low-energy class 1 [27]), which showed it was fully competitive with a solution with individual heat pumps, mainly thanks to the low heat loss from the DH network, calculated to be just 12% of the heat delivered [14].

The concept was built and successfully tested at Lystrup in Denmark in 2010 [15], on a settlement of 40 low-energy houses, low-energy class 1 [27]. With low supply/return temperatures, the annual heat loss from the DH network measured in 2012 [7] was as low as 17% of delivered heat, i.e. one quarter of the value for a network designed with traditional pipes and operated with temperature levels of 80/40°C.

Dalla Rosa et al. [28] investigated the possibility of using a low-flow DH system, characterised with design supply/return temperatures of 80/25°C and compared its cost-efficiency with the traditional concept of 80/40°C and low-temperature DH 50/25°C in the example of the settlement with 40 low-energy houses. Dalla Rosa reported that although the low-flow system resulted in smaller pipe diameters that were expected to reduce the overall heat loss from the network, the higher supply temperature meant higher heat loss than the low-temperature DH with a supply temperature of 50°C. Moreover, the authors concluded that it was better to design the DH network with smaller pipe diameters and increase the supply temperature to 60°C during very cold periods instead of designing the DH network with an all-year-round supply temperature of 50°C and bigger pipe diameters. But the study did not consider the investment cost and energy-efficiency for RES of heat related to the increase of supply temperatures, which can have considerable influence on the results. The heat loss from the DH network can be further reduced using the optimisation method to exploit all the available differential pressure for each individual customer, resulting in additional reduction of pipe diameter [21].

With decreasing heat demand in buildings and the need to deploy more renewable sources of energy, the low-temperature DH seems to be an appropriate solution. However, it should be pointed out that these considerations do not reflect existing buildings designed originally with SH and DHW systems for 80/40°C, as discussed in Section 4.1.
3 PART I - DHW HEATED BY LOW-TEMPERATURE DH

This part investigates the feasibility of supplying DHW systems using low-temperature DH and is divided into two halves. The first half focuses on the requirements for the DHW system and the performance of a low-temperature DH substation based on the instantaneous principle of DHW heating. The second half focuses on the development of an energy and cost-efficient bypass solution. The research work is described in more detail in ISI papers [24] and [29].

3.1 Specific Background

3.1.1 Requirements for DHW heating

Delivery of heat for DHW preparation is one of the main tasks for DH. DHW systems can be basically divided into two main parts: the DHW heater (in the case of DH, this is the DH house substation) where the DHW is heated from cold potable water, and the in-house DHW distribution system, i.e. pipes connecting the heat source with individual taps.

Danish Standard DS 439 [22] stipulates the following requirements for all DHW systems:

- Hygiene – DHW should be delivered without increased risk for bacterial growth (DS 439, chapter 2.5.1)
- Comfort – DHW should be delivered in reasonable time, with the desired temperature and without unwanted fluctuations in temperature

When DH is the source of heat, the DH substation should also fulfil requirements on:

- Performance – the DH substation should be able to heat DHW up to the desired temperature with the defined DH supply temperature while providing the desired cooling of DH water.

DHW temperature

DS 439 [22] stipulates that the DHW should be delivered to every DHW tap with a minimal temperature of 50°C, but the temperature can drop to 45°C during peak situations. However, later in the text, the minimal DHW temperature required from the tap is lower and varies depending on the DHW use (tapping types) and the DHW is already expected to be mixed with cold water. Specifically, in the kitchen DHW is required to be 45°C and, at other tapping points, 40°C.

Waiting time for DHW

The waiting time for DHW expresses how long the occupant should have to wait for the DHW with the desired temperature after opening the tap. The waiting time consists of the time needed for the DHW heater (i.e. the DH house substation) to produce the DHW with the desired temperature (the recovery time) and the transportation time needed to deliver DHW from the substation to the tap. Excluding DHW systems with DHW circulation, the transportation time depends on the length of the DHW pipes and their diameter. The recovery time of the substation is discussed later in this section.

DS 439 defines the “reasonable time” to deliver DHW with the desired temperature for all DHW tapping types as 10 s with a flow of 0.2L/s (DS 439, chapter 4.2.2.). However, in the case of hand washing, the waiting time is counted only to the moment when DHW with 30°C is delivered to the tap [22] (part 4.6.4), because 30°C is considered as sufficient to start hand washing. Moreover it should be mentioned that in
real use the waiting time is longer because the real flow for individual DHW tapping types is less than 0.2 L/s.

For the DHW systems with a waiting time longer than 10 s, DS 439 suggests using DHW circulation to increase the comfort for occupants and avoid wasting water flushed directly to the drain during the period of waiting for DHW with the right temperature. However, it should be mentioned that the use of DHW circulation increases the heat losses from keeping the DHW system ready to use.

**Hygiene**

For DHW, the risk of bacterial growth mainly concerns Legionella bacteria. Legionella can be present in DHW and, when the DHW is aerosolised by tapping (most often during showering), the bacteria can be inhaled to the lungs. Depending on their concentration and the person’s state of health, it can cause milder Pontiac Fever or the more severe Legionnaires disease, which is very dangerous for old people or people with a weak immune system [30]. Since the both diseases have a very similar development to regular influenza, many cases are left unrevealed.

Favourable conditions for Legionella growth are large volumes of stagnating DHW, enough nutrients and a favourable temperature range. Figure 3.1 shows that the highest risk of Legionella proliferation is in temperature range 35-45°C, i.e. exactly the temperatures of DHW used for tapping.

![Figure 3.1 – Risk of Legionella proliferation related to the DHW temperature][1]

This is why most national DHW standards require a minimal DHW temperature of 60°C to be out of the favourable growth conditions. However, according to [26], Legionella bacteria can survive temperatures of up to 80°C by hiding in amoebas attached to the sediments on the inner surfaces of DHW pipes or storage tanks. It should be mentioned that our knowledge of the risk of Legionella is in many cases ambiguous and on-site measurements are full of uncertainties. But the high temperature at the DHW heater itself does not guarantee there is no risk of Legionella, because mainly in big DHW systems hydronic misbalance can create parts of the DHW system where the DHW temperature drops to the temperature range favourable for Legionella growth [24]. So it is arguable that a minimal temperature of 60°C is not really enough.
Apart from high temperature, the alternative solutions to the risk of Legionella include micro-filtering at the DHW tap, ultraviolet light disinfection, electrolytic or chemical treatment, and cavitation. But all of these solutions need either additional energy or maintenance, have considerable running costs or use chemical substances, so keeping the DHW temperature above a certain level is the simplest and most reliable solution.

However, the risk of Legionella can be also kept low without introducing Legionella elimination solutions or keeping the DHW over 60°C simply by reducing the water volume in DHW system. The German standard [32] makes no requirement about minimal DHW temperature if the overall volume of DHW (excluding HEX) is below 3L. An attempt to do something similar was made in Danish DS information DS/CEN/TR 16355 [33], but the document comes to ambiguous conclusions, not really providing firm guidelines on minimal temperature level.

The “rule of 3L” is a cornerstone of the whole low-temperature DH concept for DHW, defined by:

- Minimum DHW temperature of 45°C, based on the comfort requirements
- Maximum length of DHW pipes, based on the maximum allowed volume of 3L
- No storage of DHW, based on the maximum allowed volume of 3L

In addition to low-temperature DH, the same concept can be used for other low-temperature heat sources, such as solar-thermal collectors or heat pumps.

The state-of-the-art DHW HEX with a temperature drop between primary and secondary sides of 3°C and an additional 2°C temperature drop as an effect of cooled DHW pipes at the beginning of tapping means that the first requirement defines the minimal supply temperature of low-temperature DH as 50°C.

The second requirement gives the maximum length of DHW pipes. It is suggested that the DHW fixtures should be individually connected with PEX pipes with an inner diameter of 10 mm, which allows 38 m of pipe in total. In the case of steel pipes with DN15 or DN10, the maximum length is reduced to 15 or 25 m respectively, which is still seen as enough for a single-family house if the location of all DHW tapping points is planned during the design phase of the house. Figure 3.2 shows an example of the design in the pilot low-temperature DH project in Lystrup, where the total length of the DHW pipes is 12.6 m. Since the DHW pipes have an inner diameter of 10mm, this means only 1 L of DHW. Proper location of the tapping points also means there is no need for DHW circulation, which is another source of energy losses.

<table>
<thead>
<tr>
<th>DHW fixture</th>
<th>nominal flow [L/min]</th>
<th>length to fixture [m]</th>
<th>volume in pipes [L]</th>
<th>velocity [m/s]</th>
<th>transportation delay [s] for: nominal flow</th>
<th>flow 0.2L/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>shower</td>
<td>8.4</td>
<td>2.2</td>
<td>0.17</td>
<td>1.8</td>
<td>1.2</td>
<td>0.9</td>
</tr>
<tr>
<td>basin</td>
<td>3.4</td>
<td>4.1</td>
<td>0.32</td>
<td>0.7</td>
<td>5.8</td>
<td>1.6</td>
</tr>
<tr>
<td>kitchen</td>
<td>6</td>
<td>6.3</td>
<td>0.49</td>
<td>1.3</td>
<td>4.9</td>
<td>2.5</td>
</tr>
</tbody>
</table>

Figure 3.2 – Example of location and connection of DHW tapping points designed in proximity of DH house substation based on the instantaneous principle of DHW in Lystrup. The table shows the transportation delay for nominal and expected flows and lengths of individual feeding pipes (inner diameter 10 mm)
The last requirement for no storage of DHW water leads to the development of a low-temperature house substation with a buffer for DH water (discussed in the next chapter).

### 3.1.2 State-of-the-art Low-temperature DH substations

A DH house substation is a device needed in buildings supplied by DH to heat DHW and/or determine the amount of heat transferred to the SH system. Moreover, the substation provides the border between the primary side (DH side) and the secondary side (house installations) very often needed to reduce temperature and/or pressure and create hydronic separation of the primary and secondary sides.

Usually, the DHW part of the house substation consists of the HEX and controllers controlling the heating of the DHW to the desired temperature connected together with pipes and fittings. Traditional high and medium-temperature DH house substations can be divided to two groups:

- Substations based on the instantaneous principle of DHW preparation (IHEU = Instantaneous Heat Exchanger Unit)
- Substations with a DHW storage tank

A substation based on the instantaneous principle of DHW heating produces DHW only when needed, whereas in a substation with a storage tank the DHW is heated slowly and stored to be ready for use. DHW storage tanks are generally used to reduce the design heating power needed for DHW preparation. In the case of DH, they mean that the diameter of pipes in DH network can be reduced, leading to reduced heat loss and also reduced peak heat power for DH heat sources.

The DH house substation can also provide a building with SH, either through an additional HEX (indirect SH) or the DH water can be used directly in the SH system (direct SH). House substations with a direct SH connection can also be equipped with a mixing loop to reduce DH water temperature, an adjustment often based on the outdoor temperature known as weather compensation. Design temperatures for the SH part of the substation are more a question of the SH system than the substation, so here the focus is on the DHW design temperatures.

![Diagram showing low-temperature DH substations](image)

**Figure 3.3 – Low-temperature DH substations; left: instantaneous DHW principle [34], right: storage tank for DH water [25], [35]**

The DHW HEX in traditional DH substations are designed for minimum DH supply/return temperatures of 60/30°C (summer conditions of medium temperature DH), whereas a low-temperature DH substation should work at the temperature levels of 50/25°C and produce DHW of at least 45°C.
The use of low-temperature DH therefore requires some modification to:

- The DHW HEX
- The DHW controller
- The DHW storage tank

**DHW HEX**
The key component of a low-temperature DH substation is a highly efficient HEX with Micro Plate™ design of plates [36], specially developed for low supply temperatures by Danfoss (see Figure 3.4). Compared to traditional HEX for high and medium temperatures, the HEX for low-temperature DH should be more efficient because the temperature difference between the DH water supplied and the DHW produced is for design conditions only about 3°C (50°C/47°C) while in traditional HEX it is as much as 10°C (60°C/50°C). Such a low temperature difference in the case of low-temperature HEX is possible thanks to the special “dimpled” pattern of the HEX’s plates, which in comparison with traditional fishbone corrugated plates increases the heat transfer area and the overall heat transfer coefficient while maintaining high cooling of primary water (i.e. low return temperature).

![Figure 3.4 – The new Micro Plate™ design compared with the traditionally used fish bone design (courtesy of Danfoss A/S)](image)

Moreover changing the corrugation pattern reduces the pressure drop down to 65% of traditional HEXs, making possible closer installation of individual plates and thus a more compact size. An example of such a HEX is the XB37H or XB06H+ implemented in a low-temperature DH substation such as Akva Less II TD [34] or Akva Les II S [37].

**DHW controllers**
The state-of-the-art DHW controller is a combined proportional-thermostatic DHW controller with an integrated differential pressure controller and $e_{save}^{TM}$ function, which ensures that the heat exchanger is cold during standby, e.g. PTC2+P [38]. At the first sight, it may be surprising that the controller is a simple self-acting mechanical controller without any electronics, but the reason is to make the product as simple as possible to reduce the cost, extend operation time and eliminate possible malfunctions.

A DHW controller with a combined proportional-thermostatic function ensures that when the customer asks for DHW, the DH flow is set to the maximum value until the DHW reaches the desired temperature, when the DH flow drops to the value needed to maintain the desired DHW temperature. This feature is very important at the beginning of DHW tapping, when the DH water in the service pipes, the HEX and
other parts of the substation can be cold and a low flow could increase the waiting time for DHW considerably. The differential pressure controller maintains constant differential pressure and thus constant performance characteristic of the valves.

**DHW storage tank**

To follow the German standard DVGW 551 [32], the water volume in the DHW system cannot be more than 3L. This requirement will be not met by traditional substations with a DHW storage tank, usually accounting for 100-150L. The solution is to “move” the storage of DHW water to the primary side and store DH water instead [25]. DHW is then prepared on the instantaneous principle in the HEX (see Figure 3.3) as in the case of a house substation based on the instantaneous DHW preparation principle. This solution is called District Heating Storage Unit (DHSU).

The unit with the buffer tank for DH water was originally designed to reduce the pipe dimensions in the DH network to further reduce the heat loss, but [7] documented that heat loss saved due to the reduced size of pipes in the DH network is lost by additional heat loss from the DHW storage tank, so this solution, with higher investment cost and higher space requirements, is suggested for use mainly on the outskirts of DH networks experiencing capacity problems.

### 3.1.3 Waiting time for DHW and DH bypass

As mentioned in chapter 3.1.1, the waiting time for DHW delivery consists of time needed for the DHW substation to produce DHW water (known as recovery time) and the transportation time needed to transport the DHW produced by DHW pipes to the tap.

The recovery time depends not only on the physical properties of the DHW HEX (heat transfer properties, volume of water, the HEX’s mass), but also on the DHW controller steering the flow of DH water, on the temperature of the DH water entering the substation, and on the history of DHW tapping. The bigger the HEX, the longer the waiting time, because DH water needs to heat up more thermal capacity (water and the HEX’s mass) before DHW with the desired temperature is produced. Similarly, the longer the time since the previous DHW tapping, the more time is needed because the HEX has cooled down. To speed up the heat flow from the DH at the beginning of tapping, the state-of-the-art proportional-thermostatic DHW controller opens the DH flow to the maximum, until the desired DHW temperature is reached, and then throttles down just to maintain the desired DHW temperature.

However, this description is fully valid only for a house substation with a buffer tank for DH water (DHSU), where the temperature of DH water supplied to the DHW HEX is expected from the very first moment to be 50°C, because the DH water is stored in the buffer tank. The situation is different for the substation with instantaneous DHW preparation (IHEU), because the HEX is supplied by DH water taken directly from the DH network. During the non-heating season, the DH water standing in the service pipes can cool down as a result of there being no heat demand in the building. This will extend the recovery time of the substation. To prevent the cooling down of the service pipes, the traditional solution is to maintain a small flow of DH water and “bypass” (see Figure 3.5) it back to the DH network just on the border of the DH substation (external bypass) or to let the DH water flow through the DHW HEX (internal bypass) by installing the bypass valve in the house substation. Having a bypass valve installed in each house substation is a better solution than having a bypass valve installed only at the end of each street pipe, because it keeps the supply service pipe warm for each customer.
Figure 3.5 – Various bypass strategies for IHEU; left: no bypass; middle: external bypass (cold HEX); right: internal bypass (warm HEX).

Both types of bypass reduce the waiting time for DHW, but bypassing DH water back to the DH network without proper cooling increases the heat loss from the DH network. The typical set-point temperature used for the external bypass in a low-temperature DH network is 35°C except for the buildings at the end of the streets, where the set-point temperature is increased to 40°C. The internal by-pass offers shorter waiting time for DHW after idling of substation, but this is paid for by higher heat consumption for its operation and greater heat loss from the HEX which is kept always warm. Furthermore, in some countries keeping the DHW HEX warm is seen as a solution that increases the risk of Legionella growth, so it is not very much used. Another disadvantage in using an internal bypass is the reduced efficiency of the HEX caused in medium DH by sedimentation on the DHW side from maintaining the HEX at higher temperature.

However, the temperature of DH water supplied to the house substation in the very first moments after a period without heat demand or during bypass operation is also influenced by the thermal capacity and transportation time in the service pipe. Let’s consider a substation based on the instantaneous DHW principle without an external bypass just after DHW tapping performed during a non-heating period. The service pipe is full of 50°C DH water, which means that the DH water in the service pipe will cool homogenously over the whole length in accordance with the cooling curves reported by Dalla Rosa [39], presented in Figure 3.6 (left). It can be seen that, for the AluFlex 20/20/110 pipe surrounded by soil with a temperature of 8°C, the DH water standing in the service pipe will homogenously cool down to 20°C in 180 minutes.
After 180 minutes, the customer opens the DHW tap again and “fresh” DH water at 50°C starts to flow to the service pipe from the DH distribution pipe in the street while the cooled DH water standing in the supply service pipe will enter the substation. The delivery of fresh DH water in the substation will be postponed by a transportation delay. Furthermore, thanks to the thermal capacity of the service pipe wall, being at the initial moment at 20°C, the DH water supplied will be cooled down for some period at the beginning of tapping. Therefore, depending on the flow rate of the DH water and the initial temperature of the service pipe, it will take some time before the DH water with a temperature of 50°C reaches the inlet to the substation, as can be seen in Figure 3.6-(right). During this period the substation will be supplied with DH water cooled by standing in the service pipe, increasing the recovery time of the substation. For the AluFlex 20/20/110 service pipe 10 m long and IHEU controlled with combined proportional-temperature DHW controller it will take almost 7.5 s to deliver DH water at 45°C and roughly another 20 s to deliver 50°C warm DH water to the DH substation. The influence of service pipe thermal capacity on the bypass operation is similar.

### 3.1.4 Low-temperature DH substation with integrated heat pump

As already discussed, the minimum DH temperature of 50°C is defined by the requirement to produce 45°C DHW without an additional energy source. However, if we go even further in accord with the recent trend in DH development, which is characterised by reducing heat loss in the DH network and the integration of more RES, the DH supply temperature can be lowered even more if there is an auxiliary energy source in each substation.

This idea is behind the concept of low-temperature DH heating with the supply temperature reduced to 35-40°C, but deploying a small heat pump installed in each substation to lift the DHW temperature to the desired temperature of 45°C [16], [40]. The heat source for the heat pump is the DH water, meaning that the temperature increase required is very small, resulting in a high coefficient of performance (COP). The heat pump lifts only the DHW part, because the supply temperature of 40°C is expected to be enough for space heating of low-energy buildings designed with floor heating or low-temperature radiators. For two
identical DH networks designed with the same pipe diameters reduction of DH supply temperature from 80°C to 40°C results in a 35% reduction in heat loss.

![Diagrams](image)

*Figure 3.7 – Principle of low-temperature DH substation combined with micro heat pump [40].*

During the development phase of the substation, few variants were analysed [40]. The variants differed in the location and size of the DHW storage tank (primary/secondary side), location of the heat pump evaporator/condenser (DH supply/return), and in exploring the possibility of using heating water returning from the SH system. The variants were evaluated from the perspective of the COP of the heat pump, exergy and maximal flow and cooling of the DH water. To get a high COP for the heat pump, the temperature of the DHW should be kept as low as possible and therefore, applying the same philosophy as for low-temperature DH, the DHW is produced with the temperature of 45°C, which means the DHW system must have a volume below 3L. The solution with the best performance was the variant with the condenser and evaporator on the primary side, boosting the DH water up to 53°C, which is then stored in the buffer tank for DH water. DHW is then heated on the basis of the instantaneous principle in a Micro Plate™ HEX as in case of DHSU. The prototype was built and tested in laboratory conditions with an average COP of 5.3. Full-scale testing of five units is currently going on in Birkerød, Denmark [16]. This concept is expected to be beneficial for DH network outskirts that experience problems with a supply temperature drop.

### 3.1.5 DHW systems supplied by low-temperature DH

**Single-family houses with low volume DHW system**

In small DHW systems, such as for single-family houses, the risk of Legionella can be kept low by reducing the overall water volume below 3L [32]. This mean maximum DHW pipe lengths of 38 m (with inner diameter 10 mm) or 15 m with DN15 pipe, which is enough for a single-family house if the location of all DHW tapping points is planned during the design phase of the house. Proper location of the tapping points also means there is no need for DHW circulation, which is another source of energy losses. In buildings with DHW storage tanks, the solution is to store DH water instead of DHW and to have the DHW heated on the basis of the instantaneous principle [25].

**Supply of existing single-family houses**

For DHW systems in buildings currently supplied by traditional DH, low-temperature DH will require replacing existing DH substations originally designed for minimal DH supply temperatures of 60/30°C with low-temperature substations designed for 50/25°C. The cheapest solution is just to replace the current HEX with a new low-temperature HEX, but an optimal solution also requires the DHW controller to be state-of-the-art, so it is easier to replace the whole substation. In buildings equipped with traditional DHW storage tanks, it is not possible just to replace the HEX and keep the current storage tank with its inner heating coil,
because DHW can no longer be heated up to the 55°C required by the standard [22], so the solution is to replace the whole DH substation with the low-temperature version. An alternative and also cheaper solution could be to turn the DHW storage tank into the buffer tank for storage of DH water and add a low-temperature HEX on the outlet, but this solution is not always possible because the DHW storage tank needs to have been designed to withstand the pressure on the DH side, usually 10 bars. In DHW systems with DHW volume in the pipes above 3L, the existing DHW pipes need to be replaced with smaller diameter pipes (preferably always connecting just one tapping place and the DHW source) to comply with the 3L rule.

DHW in multi-storey buildings
The state-of-the-art for multi-storey buildings is the concept of flat or apartment stations [41], where each apartment has own substation, and the DHW and SH pipes are laid out only in the horizontal direction. In this way, each flat owner has complete control over the settings of the DHW and SH systems, and all energy consumed is measured with just one meter. The flat station concept fits perfectly with the concept of low-temperature DH, because a properly designed DHW system in each flat will have a DHW volume below 3L. Moreover, not having DHW and SH risers between the flats reduces noise propagation.

However DHW and SH in multi-storey buildings is traditionally distributed by vertical risers, resulting in large water volumes. Delivering DHW with the desired temperature and in reasonable time in such systems therefore requires DHW circulation, in which the circulated DHW returns to the source of heat with a temperature of at least 55°C [22]. With low-temperature DH, this is simply not possible. The best solution would be the installation of individual flat stations, but this would require extensive investments. Therefore if such traditional DHW multi-storey systems are to be supplied by low-temperature DH, some kind of Legionella elimination system will be needed.

One promising solution might be a new system for the elimination of Legionella in DHW systems with temperature below 50°C that has recently been tested on 10 multi-family houses in Sweden with good results [26]. The system works on the principle of Advanced Oxidation Technology (AOT), in which UV lamps irradiate a catalytic surface to form free radicals. The radicals then break down contaminants in the water. The process occurs only inside the purifier and leaves no harmful residuals in the water. One UV lamp is installed on the cold water supply to the DHW heat exchanger and another UV lamp on the DHW circulation just before the heat exchanger. The disadvantages of such a solution are the considerable investment and running costs, because the lamps need to be changed once a year. Over the lifetime of 20 years, this amounts to about DKK 3800 per annum [42]. This solution might work for DHW systems supplying several flats where the tenants share the costs, but is currently too expensive as an alternative for use in single-family houses. Moreover, the study concluded that the bacterial contaminants are found in almost all piping systems, so the realistic vision should be to provide a technical solution to limit Legionella growth to a low level rather than attempt complete elimination.
3.2 Delivery of DHW

3.2.1 Methods
This chapter describes methods used in the research only in general. A detailed description is in ISI article [24].

Based on a study of the literature, a minimum DHW temperature of 45°C and a waiting time of 10 s were adopted.

Full-scale measurements of IHEU
The performance evaluation of a low-temperature house substation based on the instantaneous principle and supplied with 50°C DH water focused on the production of DHW with the required temperature in the required time with proper cooling of DH water. The performance of an IHEU substation equipped with a PTC2+P controller for DHW was measured in well-defined conditions in the laboratories of the Technical University of Denmark. Previously calibrated type T thermocouples were used for the test, measuring the inlet/outlet temperatures on both primary and secondary sides. Water flow rates were measured on both sides with traditional analogue flow meters. The substation was connected to the campus DHW system to provide the inlet of the substation with 50°C water constantly.

First we adjusted the DHW controller to produce 47°C DHW to confirm it was possible to achieve 45°C at the tap. The temperature of 47°C was chosen because it is the required temperature of 45°C with the addition of 2°C to cover the effect of cooled DHW pipes at the beginning of a tapping period. Then we left the substation to cool down to the temperature of the ambient environment to simulate a long period without tapping. After that, we started tapping DHW with a flow of 8.4L/min to simulate DHW demand for showering with 40°C warm DHW and observed the temperature development of the DHW produced, defining the recovery time of the substation. The waiting time was defined as the time needed for the DHW to reach 40°C – a choice based on the consideration that at the beginning of DHW tapping the DHW temperature drops by 2°C due to cooled DHW pipes and that 38°C is considered as a comfortable temperature for DHW use.

Development of numerical model for IHEU
The realistic evaluation of the performance for the IHEU accounted also for the impact of the DH network represented by the influence of the bypass solution and the supply service pipe. Since it would be very difficult to measure the detailed performance in a full-scale experiment, we developed a numerical model for this purpose. The philosophy of the model can be seen in Figure 3.8.

![Figure 3.8 – Structure of model investigating waiting time for DHW in the DH substation without a storage/buffer tank](image)

overall performance
substation | service pipe

`HEX` DHW controller service pipe

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The model we developed not only allowed investigation of the bypass performance, but can also be used as a fast decision tool to customise a substation for various requirements specified by DH companies.

The numerical model of the IHEU substation was developed in MATLAB and Simulink [43] based on earlier models from Danfoss A/S. The model of the substation consists of a module representing the HEX, a module representing the PTC2+P DHW controller, and additional blocks representing mainly the time delays in the pipes and their thermal capacity. The model of the HEX is based on the model in Persson [44], where it is well described. The model consists of three sections, and each section consists of a cold and a hot side and the wall between them (see Figure 3.9).

![Figure 3.9 – Description of the numerical model of the HEX with three sections](image)

The three sections model is considered to be accurate enough to model the overall performance of a HEX [44]. The philosophy of HEX modelling is fundamentally based on an energy balance between the primary (hot) and secondary (cold) side including heat transfer through the wall separating the two sides, described by equations (1)-(4).

Energy balance equation for cold medium:

$$\frac{d}{d\tau}(m_c \cdot c_{p,c} \cdot T_{c,\text{out}}) = \dot{m}_c \cdot c_{p,c} \cdot (T_{c,\text{in}} - T_{c,\text{out}}) - \alpha_{c,w} \cdot A_c \left(\frac{T_{c,\text{in}} - T_{c,\text{out}}}{2} - T_w\right)$$  \(1\)

Equation (1) rearranged and written for HEX-section 1:

$$\frac{d}{d\tau}(T_{c,\text{out}}) = \frac{1}{m_{c1} \cdot c_{p,c}} \left[\dot{m}_c \cdot c_{p,c} \cdot (T_{c,\text{in}} - T_{c,\text{out}}) - \alpha_{c,w} \cdot A_c \left(\frac{T_{c,\text{in}} - T_{c,\text{out}}}{2} - T_w\right)\right]$$  \(2\)

Energy balance equation for hot medium:

$$\frac{d}{d\tau}(m_h \cdot c_{p,h} \cdot T_{h,\text{out}}) = \dot{m}_h \cdot c_{p,h} \cdot (T_{h,\text{in}} - T_{h,\text{out}}) - \alpha_{h,p} \cdot A_h \left(\frac{T_{h,\text{in}} - T_{h,\text{out}}}{2} - T_w\right)$$  \(3\)

Energy balance equation for HEX plates:

$$\frac{d}{d\tau}(m_w \cdot c_{p,w} \cdot T_w) = \alpha_{h/w} \cdot A_h \left(\frac{T_{h,\text{in}} - T_{h,\text{out}}}{2} - T_w\right) - \alpha_{c,w} \cdot A_c \left(\frac{T_{c,\text{in}} - T_{c,\text{out}}}{2} - T_w\right)$$  \(4\)
where:

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_c$</td>
<td>total plate area on the cold side in one section</td>
<td>$[m^2]$</td>
</tr>
<tr>
<td>$c_p,c$</td>
<td>specific heat capacity of water</td>
<td>$[J/(kg.K)]$</td>
</tr>
<tr>
<td>$m_c$</td>
<td>mass of water in the section</td>
<td>$[kg]$</td>
</tr>
<tr>
<td>$m_i$</td>
<td>mass flow of water through the section</td>
<td>$[kg/s]$</td>
</tr>
<tr>
<td>$m_w$</td>
<td>total mass of plates in the section</td>
<td>$[kg]$</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
<td>$[-]$</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
<td>$[-]$</td>
</tr>
<tr>
<td>$T_{cin}$</td>
<td>temperature of water coming into the section</td>
<td>$[^{°}C]$</td>
</tr>
<tr>
<td>$T_{cout}$</td>
<td>temperature of water leaving the section</td>
<td>$[^{°}C]$</td>
</tr>
<tr>
<td>$T_w$</td>
<td>average wall temperature in the section</td>
<td>$[^{°}C]$</td>
</tr>
<tr>
<td>$\alpha_{c,w}$</td>
<td>convective heat transfer coefficient</td>
<td>$[W/(m^2.K)]$</td>
</tr>
<tr>
<td>$\delta_p$</td>
<td>distance between individual HEX plates</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$\lambda_w$</td>
<td>thermal conductivity of water</td>
<td>$[W/(m.K)]$</td>
</tr>
</tbody>
</table>

This approach adopts some simplifications: no heat conduction between the sections in the direction of water flow, negligible heat resistance in the HEX walls, and no heat losses to the surroundings. However, their influence on the accuracy in this application is negligible. The DHW controller was modelled as a set of individual mechanical parts with the parameters given by the manufacturer. Part of the implementation to the MATLAB Simulink can be seen in Figure 3.10.

![Figure 3.10 – Part of the Simulink model of the IHEU substation – detail on HEX model](image)

The model was successfully verified with the data from laboratory measurements, under both steady-state and dynamic conditions (see Figure 3.11 and Table 4-2). Figure 3.11 shows the results obtained from the measurements of recovery time for the IHEU and compares them with the results from the numerical model developed for the same initial conditions. It can be seen that the model is in good agreement with the measured data and can therefore be used. Numerical values are reported in Table 4-2: case M for measurements and case 0 for the simulation.
Influence of service pipes and bypass solutions

The influence of the supply service pipe without the bypass on the recovery time of the house substation was discussed in section 3.1.3, but if a bypass solution is used, its influence needs to be added. It must be stressed that the bypass solution is active only in periods without a need for heating, i.e. in non-heating periods. The bypass flow is usually controlled by a thermostatic valve FJVR [45], controlled on the basis of the temperature of the passing fluid, and installed just after the inlet to the IHEU. To ensure the stability of the control process, the valve has a deadband of ±2.5°C. For a set-point temperature of 35°C, this means that the valve opens when the temperature of the fluid drops below 32.5°C and closes when the temperature reaches 37.5°C. This intermittent operation is also called “pulse” mode.

By recurrent application of the numerical code for cooling of DH water in service pipes [46] combined with the numerical code for the dynamic heat transfer in the service pipes [39], we obtained a temperature profile along the 10 m long AluFlex 20/20/110 service pipe (Figure 3.12) for the case of an IHEU with an external bypass. We assumed constant 50°C DH water at the beginning of the service pipe (maintained by heat demand of previous/following DH customers) and a bypass flow rate of 3L/min in periods when the bypass was opened.
The Simulated substation times flow deadband from continuous operation, showing the influence of the deadband on the temperature profile. Figure 3.12 shows the temperature profile along a 10 m long SP AluFlex 20/20/110 for an IHEU equipped with an external bypass in traditional and continuous operation modes. Traditional bypass: set-point 35°C, deadband ±2.5°C, bypass flow rate 3 L/min, $T_{ground}$=14°C. Continual bypass: temperature drop 50-35°C, flow rate 0.024 L/min, $T_{ground}$=14°C.

The figure shows that the temperature profile along the supply service pipe varies depending on how many times the bypass valve has opened since the previous tapping. The temperature profile of DH at the inlet to the substation can be obtained by the additional application of code for the dynamic calculation of heat transfer in the service pipe and knowing the primary flow rate to the substation.

Figure 3.12 also shows the temperature profile for a theoretical bypass controller providing continuous flow without the deadband. The results of Dalla Rosa et al. [47] show that such a bypass control results in a 30% reduction of heat loss from the service pipe. Moreover, it maintains the temperature at the inlet to the substation continuously at 35°C, so it would be expected to reduce the maximum length of heat recovery.

Simulated cases
The bypass influence on the recovery time of the IHEU was simulated for the following cases:

1) No bypass – service pipe cools down during non-heating season
2) Traditional external bypass with a set-point temperature of 35°C, operating in pulse mode
3) Continuous bypass – constant continuous bypass flow ensuring 35°C at the inlet of substation

The case without the bypass investigated whether it is possible to operate an IHEU without a bypass solution and how long it will take to produce DHW with the desired temperature if the service pipe is full of 20°C or 35°C DH water at the beginning of DHW tapping. The second case considered the use of an external bypass solution with a set-point temperature of 35°C, modelled as an on/off controller with a deadband of ±2.5°C simulating the performance of a real external bypass. The last case was an external bypass modelled with a controller without the deadband, which was expected to result in lower heat loss from the service pipe.
3.2.2 Results and Discussion

Study of the literature gave a good indication on the feasibility of delivering DHW at the tap with a temperature of 45°C if the overall volume of the DHW system is below 3L.

3.2.2.1 Performance of IHEU

Table 3-1 shows the recovery times of the IHEU for different boundary conditions as results from the numerical model. The upper part of Table 3-1 reports the recovery time for the IHEU without the influence of the service pipe, while the lower part includes this influence.

Table 3-1 – Recovery time for IHEU

<table>
<thead>
<tr>
<th>Case</th>
<th>Description</th>
<th>Recovery time of IHEU [s] to produce DHW</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>30°C</td>
</tr>
<tr>
<td>M</td>
<td>experimental measurement</td>
<td>4.1</td>
</tr>
<tr>
<td>0</td>
<td>verification of num. model</td>
<td>4.6</td>
</tr>
<tr>
<td>1</td>
<td>pure recovery time, T=50°C, HEX20</td>
<td>3.7</td>
</tr>
<tr>
<td>2</td>
<td>pure recovery time, T=50°C, HEX40</td>
<td>4.1</td>
</tr>
<tr>
<td>3</td>
<td>continual bypass 50°C - 35°C</td>
<td>5.7</td>
</tr>
<tr>
<td>4</td>
<td>ext. bypass before 2nd bypass flow</td>
<td>5.9</td>
</tr>
<tr>
<td>5</td>
<td>ext. bypass before 3rd bypass flow</td>
<td>7.2</td>
</tr>
<tr>
<td>6</td>
<td>without bypass – 35°C</td>
<td>7.2</td>
</tr>
<tr>
<td>7</td>
<td>without bypass – 20°C</td>
<td>11.0</td>
</tr>
</tbody>
</table>

Excluding the influence of the service pipe, the recovery time to produce 40°C DHW is 7.4 s if the substation is left idle for a period long enough for all components to cool down to 20°C. The recovery time to produce 40°C is reduced when the DHW HEX is changed to 20 plates instead of 40 plates. Use of a DHW HEX with 20 plates slightly increases the return temperature of DH water from 19.1°C to 21.8°C, but this value is still in the acceptable range and is justified by the fact that the amount of material is halved and so is the price of the HEX.

For a 10 m long service pipe (Aluflex 20/20/110), with water cooled to 20°C by a long period of idling, the time needed for the IHEU to produce 40°C warm DHW without an external bypass increases to 15 s (see Table 3-1). Moreover, the transportation time needed to deliver DHW to a shower at the end of a 2.2 m pipe is an additional 1.2 s. Taken together, this gives a waiting time of more than 17 s, meaning discomfort for the customer as well as wasting water while waiting for DHW with the desired temperature. With the deployment of an external bypass with a set-point temperature of 35°C, the recovery time decreases by 3-4 s, depending on the bypass phase. Using continuous flow with a set-point temperature of 35°C will decrease the recovery time by another 1 s, but can result in increased heat loss from the service pipes depending on how the solution is realised.
These results might give the impression that the use of external bypass saves only 3.5 s on average, i.e. 22% of the waiting time, but it should be pointed out that a bypass installed in the substation also has crucial importance for keeping the whole network warm and thus is relevant not only for the service pipe of individual customers but for whole DH network. Comparison with traditional DH substations is not available.

The recovery time could be further shortened by reducing the thermal mass (metal components) and water content, e.g. using DHW HEX with 20 plates instead of 40 and reducing the size of fittings (pipes and armatures) in the substation or alternatively making them from plastic. Another alternative could be an internal bypass to keep the DHW HEX warm, but for low-temperature applications such a solution is not currently available because the self-acting controller used for this solution in traditional DH needs higher difference between the DH supply and the DHW set-point temperature, as mentioned in [48]. This has also been confirmed by laboratory test [24]. The waiting time for DHW in older systems was very short if the pipes were kept continuously warm and ready to use. This resulted in high comfort for customers, but at the same time in high heat losses from the system. Therefore the energy efficiency question arises in connection with the validity of the 10 s waiting time limit for customers, calling for full-scale investigation.
3.3 Cost-efficient Use of Bypassed DH Water in Bathroom Floor Heating

The results from the numerical simulations of a low-temperature IHEU showed that to provide DHW in reasonable time during a non-heating period, a bypass solution needs to be applied. Since the traditional bypass solutions redirect the bypass flow back to the DH network almost without cooling and increase heat loss from DH network, we investigated the feasibility of making the bypass more energy- and cost-efficient. We chose the option of redirecting the bypass flow to bathroom floor heating in order to force the additional cooling of bypassed water resulting in a reduction of heat loss from the DH network while giving the sensation of a warm floor to the users. This is referred to as the “comfort bathroom” (CB) concept.

3.3.1 Methods

Various technical proposals for redirecting and controlling the bypass flow for bathroom floor heating have been developed (see Figure 3.13). Some of the solutions were implemented with a traditional FJVR bypass valve [45] supplying a floor heating loop with the intermittent “pulse” flow caused by the deadband of the controller (as described in section 3.2.1). Others were implemented with a needle valve providing continuous flow, which was expected to reduce the heat loss from the service pipe.

The needle valve maintains a constant pre-set flow rate, irrespective of the temperature of the bypassed water, while the FJVR valve always stops the flow when the temperature of the bypassed water exceeds the set-point temperature. This feature is very important if DHW is tapped several times a day, filling the service pipe with 50°C warm DH water. In relation to the use of bypass water in floor heating, this means...
that with the needle valve the floor heating loop is supplied with 50°C warm DH water for some time after DHW tapping, whereas with the FJVR valve the bypass flow is stopped.

Apart from the difference in control valves, the technical proposals also differed for directly and indirectly connected space heating systems and took into account the presence of a mixing loop for adjusting the supply temperature.

**Overall performance of bypass flow redirected to the floor heating**

The advanced level of IDA-ICE, version 4.22 [50], was used to evaluate the performance of the different technical solutions for the implementation of CB. The advanced level is an object-based interface (similar to MATLAB-Simulink) which allows the use of detailed component models and the development of your own models, e.g. a mixing loop. To reflect various distances of the customers from the heat plant, we assumed three bypass flow rates needed to maintain a temperature of 35°C at the end of service pipe. Bypassed water was redirected to floor heating in both bathrooms of the reference 157m² single-family house built in accordance with Danish building requirements class 2015. The outputs of the simulation were heating power delivered by floor heating, an increase in indoor temperature, and additional cooling of bypassed water. Table 3-2 shows a matrix of the simulated cases for three locations in the DH network, covering various solutions for CB implementation as well as the solution with bathroom floor heating controlled by a traditional thermostat (a FJVR valve) and with no floor heating at all. All the simulations included the influence of DHW tapping.

**Table 3-2 - Matrix of simulated cases**

<table>
<thead>
<tr>
<th>case #</th>
<th>abbreviation</th>
<th>( T_{up} ) to CB &gt;35°C</th>
<th>flow [kg/h]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>1.77 4.68 9.36</td>
</tr>
<tr>
<td>1</td>
<td>noCB - FJVR</td>
<td>yes</td>
<td>✓ ✓ x</td>
</tr>
<tr>
<td>2</td>
<td>noCB - external BYP</td>
<td>no</td>
<td>✓ ✓ ✓</td>
</tr>
<tr>
<td>3</td>
<td>CB - needle valve</td>
<td>yes</td>
<td>✓ ✓ ✓</td>
</tr>
<tr>
<td>4</td>
<td>CB - FJVR</td>
<td>no</td>
<td>2.6 7.1 14.0</td>
</tr>
<tr>
<td>5</td>
<td>CB - electr. step valve</td>
<td>no</td>
<td>✓ ✓ ✓</td>
</tr>
<tr>
<td>6</td>
<td>CB - FJVR + mixing loop</td>
<td>no</td>
<td>x 7.1 x</td>
</tr>
</tbody>
</table>

The average heating power transferred to the bathroom floor from the bypass flow was used as input data for simulating the influence on the heat loss from the DH network during non-heating periods. The simulations were performed using the commercial program Termis [51] on the example of an existing low-temperature DH network connecting 40 low-energy houses. The performance evaluation included improvement in the thermal comfort for occupants expressed by an increase in floor temperature and the cost-efficiency of the DH network expressed as a reduction in heat loss.

**Detailed modelling of bypass flow to the floor heating**

To investigate the realistic performance of a bypass flow redirected to bathroom floor heating, it was necessary to model in detail the performance of both FJVR and needle valves and also the influence of DHW tapping.
First we simulated in detail the operation of both bypass solutions, implemented on a 10 m long service pipe Aluflex 20/20/110, adjusted to the same set-point temperature of 35°C. The simulation was performed using the numerical code for the dynamic calculation of heat transfer in the service pipe developed by Dalla Rosa in [39] for three different locations in the DH network. The locations differed with regard to the temperature expected at the beginning of the service pipe. It was 50°C for a building located close to the heat source, 40°C for a building located in the middle of DH network, and 37.3°C for a building located in the outskirts of the DH network. The results showed that the pulse operation of the bypass flow, caused by the deadband of the FJVR valve, results in a 50% increase in the bypassed water volume and therefore also in higher heat loss from the service pipes. For example, for the location close to the DH heat plant, the needle valve results in a continuous flow of 1.77 L/h, while the FJVR valve results in 2.6 L/h, divided into four pulses (1 pulse every 15 minutes). The flows for other locations are reported in Table 3-2.

To verify the simulation results, we performed full-scale measurements of both simulated valves in a real DH network. However, even using the DH meters with the highest available flow resolution of 1 litter was not enough to measure precisely the volume of bypassed water and confirm the intermittent or pulse operation of the traditional bypass control valve. So we made a few scenarios to cover the uncertainty about the performance of the FJVR valve and investigated its influence on the performance of bathroom floor heating supplied by the bypassed water.

First we modelled the FJVR valve in accordance with the results of the detailed numerical calculation as a pulse flow with an overall bypassed volume 50% higher than in the case of the needle valve (see Table 3-2, case 4). Second, we modelled it as a pulse flow, but with the same volume of bypassed water as with the needle valve. And third, we modelled it as a continuous bypass flow with the same volume flow rate as with the needle valve, but with the electronic valve able to stop the bypass water when the temperature is over set-point, i.e. 35°C (Table 3-2, case 5). The simulation of the performance of the needle valve was also included (case 3). The detailed simulation of performance was also made using the advanced level of IDAICE, version 4.22, and included the influence of DHW tapping. DHW tapping was assumed every three hours, lasted five minutes, and occurred daily between 6:00 and 24:00.

![Figure 3.14 - Comparison of: a) $T_{\text{flow}}$ and b) $T_{\text{ret}}$ in CB for two locations in the DH network modelled with four different control strategies for the bypass flow, with the output for results in steps of 3.6 s.](image-url)
Figure 3.14 compares the floor temperature $T_{\text{floor}}$ in the bathroom and the return temperature of bypassed water for two locations in the DH network (close and middle proximity to the heat plant) for three modelled variants of the FJVR valve and also compares them with the continuous flow provided by a needle valve. First, it can be seen that the needle valve (triangle marker) results in the highest floor and return temperatures after DHW tapping, because it does not stop the flow of bypassed water. Second, although the FJVR (diamond marker) nominal flow was 50% higher than with the needle valve, the automatic stop function applied during and after DHW tapping in fact results in lower floor surface temperature and greater cooling of bypass water than with the needle valve. And finally, there is a negligible difference in the floor surface temperature and the return temperature of the bypass water between the FJVR valve modelled as continuous (round marker) or pulse flow (square marker) as a result of floor thermal mass.

This finding therefore allows us to model the pulse FJVR as a continuous flow without significant influence on the overall performance. This saves computational time, because the simulation of pulse bypass needs time steps small enough to catch the nature of the intermittent bypass (around 30 seconds) while other parts of the model can be modelled with longer time steps.

### 3.3.2 Results and Discussion

Figure 3.15 (note: comparable control strategies have the same markers) compares $T_{\text{floor}}$ in the bathroom of low-energy single-family houses situated at short, medium and long distances from the heating plant for a two-day period during the non-heating season. It shows CB solutions realised with a needle valve and an FJVR valve with a 3°C deadband as well as both reference cases without CB. While the floor surface temperature is about 23°C for the reference case without floor heating, the surface temperature of the floor supplied by bypass flows increases by approx. 0.5°C, 1.25°C and 2°C for the houses located at short, medium and long distances from the heating plant respectively. This shows that by using the CB concept the customer improves comfort by the sensation of a warm floor, but that the performance of the CB solution differs depending on the location of the customer in the DH network. The figure also shows that the difference in floor surface temperature between the solution realised with the needle valve solid lines) and the traditional FJVR valve with deadband (dashed lines) is almost negligible. It is true that the nominal flow for the FJVR valve is 50% higher than for the needle valve, but with the DHW tapping, the automatic stop function of the FJVR valve activated when the temperature of the DH water increases over 35°C results in just a slightly higher volume of bypassed water. The results for the traditional control of floor heating using an FJVR valve with a set-point temperature of 25°C installed at the return of the floor heating loop (diamond marker on dashed line) show that the traditional solution corresponds to the CB solution realised with a traditional FJVR valve (round markers) with a set-point temperature of 35°C installed in buildings at medium distance from the heat source.
Figure 3.15 – $T_{\text{floor}}$ during a non-heating period in a bathroom with CB realised with a needle valve or an FJVR valve

Table 3-3 reports the average values during a non-heating period for all simulated cases. The results show that the performance of the solution with the needle valve (case 3) is very similar to the one with traditional FJVR valve with 3°C deadband. However, the FJVR solution should be preferred because of the advantages of the FJVR valve as a simple adjustment, no need for re-adjustment when changing the DH supply temperature, and its lower risk of clogging.

Table 3-3: Comparison of simulated cases for non-heating period 15/4 – 15/11, i.e. 5160 hours, at a price of 650 DKK/MWh.

<table>
<thead>
<tr>
<th>case #</th>
<th>nominal bypass flow</th>
<th>$T_{\text{in}}$ avg.</th>
<th>$T_{\text{in}}$ avg.</th>
<th>$T_{\text{out}}$ avg.</th>
<th>$T_{\text{out}}$ avg.</th>
<th>bypass ed volume</th>
<th>average heat output from FH</th>
<th>energy delivered by SP</th>
<th>energy used in FH</th>
<th>heat demand incl. FH</th>
<th>increase of heat demand</th>
<th>CB cost for customer</th>
<th>CB cost for DH company</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.77</td>
<td>23.5</td>
<td>24.4</td>
<td>25.2</td>
<td>24.3</td>
<td>97</td>
<td>1212</td>
<td>500</td>
<td>2945</td>
<td>17%</td>
<td>325</td>
<td>-49</td>
<td></td>
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<td>22.4</td>
<td>35.0</td>
<td>9.7</td>
<td>0</td>
<td>435</td>
<td>0</td>
<td>2352</td>
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<td>0</td>
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<tr>
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<td>23.0</td>
<td>23.3</td>
<td>9.1</td>
<td>32</td>
<td>430</td>
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<td>2518</td>
<td>7%</td>
<td>107</td>
<td>-3</td>
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</tr>
<tr>
<td>4</td>
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<td>22.7</td>
<td>23.0</td>
<td>23.2</td>
<td>9.7</td>
<td>29</td>
<td>435</td>
<td>150</td>
<td>2502</td>
<td>6%</td>
<td>98</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>1.77</td>
<td>22.9</td>
<td>22.9</td>
<td>22.9</td>
<td>6.9</td>
<td>22</td>
<td>322</td>
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<td>2467</td>
<td>5%</td>
<td>75</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>4.68</td>
<td>23.5</td>
<td>24.4</td>
<td>25.2</td>
<td>32.6</td>
<td>96</td>
<td>1454</td>
<td>496</td>
<td>2943</td>
<td>17%</td>
<td>322</td>
<td>-199</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>7.1</td>
<td>22.4</td>
<td>22.4</td>
<td>35.0</td>
<td>25.7</td>
<td>0</td>
<td>1089</td>
<td>0</td>
<td>2348</td>
<td>0%</td>
<td>0</td>
<td>97</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>4.68</td>
<td>23.2</td>
<td>23.8</td>
<td>24.4</td>
<td>24.1</td>
<td>72</td>
<td>1069</td>
<td>373</td>
<td>2721</td>
<td>14%</td>
<td>242</td>
<td>-151</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>7.1</td>
<td>23.2</td>
<td>23.7</td>
<td>24.3</td>
<td>25.7</td>
<td>64</td>
<td>1089</td>
<td>333</td>
<td>2681</td>
<td>12%</td>
<td>216</td>
<td>-119</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>4.68</td>
<td>23.1</td>
<td>23.3</td>
<td>23.8</td>
<td>16.8</td>
<td>46</td>
<td>727</td>
<td>236</td>
<td>2586</td>
<td>9%</td>
<td>153</td>
<td>-90</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>7.1</td>
<td>23.2</td>
<td>23.7</td>
<td>24.4</td>
<td>25.7</td>
<td>63</td>
<td>1089</td>
<td>323</td>
<td>2697</td>
<td>12%</td>
<td>210</td>
<td>-119</td>
<td></td>
</tr>
</tbody>
</table>

- **Nominal bypass flow**: The flow rate of the bypass valve in kg/h.
- **$T_{\text{in}}$ avg.**: The average temperature at the inlet in °C.
- **$T_{\text{out}}$ avg.**: The average temperature at the outlet in °C.
- **bypass ed volume**: The volume of bypassed water in m³.
- **average heat output from FH**: The average heat output from the floor heating in W.
- **energy delivered by SP**: The energy delivered by the supply in kWh.
- **energy used in FH**: The energy used in the floor heating in kWh.
- **heat demand incl. FH**: The heat demand including the floor heating in kWh.
- **increase of heat demand**: The increase in heat demand as a percentage of the base heat demand.
- **CB cost for customer**: The cost of the bypass valve to the customer in DKK.
- **CB cost for DH company**: The cost of the bypass valve to the DH company in DKK.
The CB concept controlled by the FJVR valve results for the three locations in the DH in an increase of average floor surface temperature from 22.4°C to 23.0°C, 23.7°C and 24.6°C, respectively. At the same time, the weighted average return temperature drops from 35°C for the traditional bypass operation to 23.2°C, 24.3, 25.6°C, respectively. From the economic perspective, the customer gained a warm floor in the bathroom for the additional cost of between DKK 98 and 371 per year, i.e. a 6% to 20% increase in the annual heat demand. However, we suggest setting the same, discounted price for all customers, reflecting the fact that the heat used in the comfort bathroom would be lost in the DH network anyway if it was not extracted for bathroom floor heating. This is documented in the last column of the table, where the cost for the operation of a traditional external bypass is estimated at about DKK 100 for all locations.

Table 3-3 reported results only from the perspective of individual buildings. The overall evaluation of the performance of CB tested in the low-temperature DH network supplying 40 low-energy houses resulted in cooling of bypassed water by 7.5°C and reduced heat loss during the non-heating season by 13% in comparison with the traditional external bypass solution. This corresponds to 40% of heat demand in the bathrooms during summer, meaning that the part needed to be covered by the customer decreases to 60% of the real additional demand in the bathroom.

3.4 General Conclusion from Part I

DHW systems equipped with low-temperature DH substations can be supplied by low-temperature DH to provide DHW with the required temperature and without increased risk of Legionella. Fast provision of DHW is ensured by bypass flow, which if redirected to bathroom floor heating, partly relocates the heat loss from keeping the DH network warm and improves thermal comfort of the customers.
4 PART II – SPACE HEATING SYSTEMS SUPPLIED BY LOW-TEMPERATURE DH

Part II investigates the feasibility of supplying space heating systems (SH) using low-temperature DH, and is divided into two halves, the first focusing on existing buildings and the second on new low-energy buildings. More detail on research related to existing buildings can be found in ISI paper [52]. The research related to low-energy buildings is reported in conference paper [53].

4.1 Existing Buildings

4.1.1 Specific background

Most of the Danish building stock consists of buildings built around the 1970s, as a result of a peak in population growth [54]. In what follows, these buildings are called “existing buildings”, meant in the sense of a counter-pole to low-energy buildings. Compared to low-energy building, e.g. class 2010 [55] with an energy framework of 63kWh/(m².a), existing building from the 1970s have considerably greater energy demand, resulting in a typical energy use of about 200 kWh/(m².a). The energy demand of buildings built after 1977 drops significantly as a consequence of the building regulations (BR1977) demanding a lower U-value for construction elements to reflect the energy crisis in the 1970s [56]. However, existing buildings will continue to make up a large share of the building stock for many years to come and it is estimated that their share in Denmark in 2030 will be about 85-90% [3]. So the question arises as to whether such buildings can cope with low-temperature DH with supply temperatures of 55-50°C and, if not, what renovation measures need to be carried out on the building envelope and the SH and DHW systems, and how should the DH network be operated. These buildings are usually equipped with SH and DHW systems designed for supply temperatures of around 70°C or higher, so a reduction of DH supply temperature would be expected to cause discomfort for the occupants. So one possible solution is to operate the DH network with a supply temperature of 50°C for most of the year and increase the DH supply temperature only during cold periods. However, once the DH supply temperature drops below 60°C, the DHW substation needs to be replaced with a low-temperature version (as discussed in chapter 3.1.2).

Reduction of DH supply temperature

From the perspective of occupants, the DH supply temperature can be reduced as long as it does not violate their thermal comfort. This needs to take account of the fact that occupants tend to maintain an indoor temperature of 22°C rather than 20°C [7] and should focus on the operative temperature rather than the air temperatures sometimes experienced. This is important, especially in older buildings where the low temperatures of inner-surfaces of the construction have a big influence on the thermal comfort of the occupants. From the perspective of DH, the maximum hydraulic capacity of the DH network and the availability of the heat sources that can provide peaked DH supply temperature during cold periods need to be considered.

However, the maximum supply temperature needed in the SH systems can be further reduced by improving the building envelope or by replacing the original SH system with a low-temperature system extracting more heat by better cooling of DH heating water. From the long-term perspective, the preferred solution is to reduce the heat demand by improving the building envelope, but due to the investment cost not every house owner is willing to do this. Replacing the SH system is a cheaper and faster solution, but it does not bring any energy savings; it just allows existing buildings to be supplied by DH with reduced supply
temperatures. Refurbishment measures carried out on existing houses vary from no measures (original state) to extensive renovation, including replacing the windows and wall and roof insulation. Replacing the windows is the most typical refurbishment, because the window lifetime of 30 years has passed and a relatively small investment brings considerable heat savings.

Considering the renewable sources of heat which needs to be built before 2035, it is cheaper to refurbish the existing buildings as fast as possible to reduce their peak capacity and thus the investment costs for low-temperature DH. Subsidies for the building refurbishment could be therefore from the long-time perspective advantageous [57].

4.1.1 Methods

The feasibility of integrating existing buildings to low-temperature DH with a design supply temperature of 50°C was modelled in the advanced level of the IDA-ICE program, version 4.22. The approach applied was to lower the original supply temperature curve of the space heating system until the operative temperature indoors drops below the desired value of either 22°C or 20°C. We chose a 157m² single-family house built in 1973 as a typical representative of the Danish building stock. The house was part of a Realea renovation project to investigate the reduction in energy demand for refurbished houses from the 1970s and the project reported enough information to develop and verify a model of the house [12], [58]. The house was modelled as a multi-zone model with 12 zones, each representing one room. The difference between the measured and modelled heat demand for the reference case of the non-renovated house was only 2.5%.

First we dimensioned the SH system with radiators for temperature levels of 70/40/20 (supply temperature, return temperature, air temperature) based on DS 418 [59] to cope with a constant outdoor temperature of -12°C without internal or external heat gains. We chose the air temperature as design temperature with a view to evaluating the influence of the system when later operated with operative temperature. The nominal heat output of real radiators [60] was chosen as close as possible to the dimensioned values. The more over-dimensioned the radiators are, the more the supply temperature can be reduced, which means the model would not reflect the design conditions.

Then we included the heat gains expected from occupants and equipment (4.2W/m² – constant) [53] and ran the model with the Danish reference year weather file. By step-by-step lowering of the supply temperature, we defined a non-linear supply temperature curve for the SH system. To maintain the same hydraulic conditions in the DH network, however, we limited the maximum flow rate from the DH network to the value defined originally for the design conditions of 70/40/20. Using the same approach, the supply temperature curve was also defined for the house with the original windows replaced around the year 2000 with standard ones and for the house with low-energy windows and additionally insulated ceiling. Table 4-1 reports the complete list of simulated cases. Moreover, for all three building states, we investigated replacing the original radiators with low-temperature radiators. Finally, we supplied the model with measured weather data to compare the results for the design reference year with real weather.

4.1.1 Results and Discussion

Figure 4.1 shows the supply temperature (T_{sup}) curves needed for the SH system to maintain an operative temperature (T_{op}) of 20°C and 22°C for the typical single-family house from the 1970s according to the numerical simulations. The curves represent results for the building in three different stages of envelope
refurbishment and include the option of the installation of low-temperature radiators. The maximum flow in the DH network and the SH system is exceeded only in the case of “high flow”.

Figure 4.1 – Supply temperature curves for all the cases investigated

It can be seen that reducing the desired operative temperature from 22°C to 20°C reduces the maximum supply temperature needed by about 5°C for all three stages of building envelope refurbishment. Installation of low-temperature radiators with the same projection area, just with more heat transfer plates, further reduces the maximum supply temperature needed by 6°C for the non-renovated house, by 3°C for the house with new-windows, and down to 50°C for the extensively renovated house.

Figure 4.2 reports the duration of the period when the supply temperature needed to be increased over 50°C for all the cases investigated. The house with the original windows already replaced can be supplied with low-temperature DH at 50°C and maintain an operative temperature of 22°C for approx. 83% of the year and needs a maximum supply temperature of 67°C.
The increase in the DH supply temperature above 60°C is needed only for 1.6% of the time (140 hours). This result is based on an operative temperature of 22°C as a realistic temperature desired by customers. The operative temperature of 20°C, which is usually used for energy calculations, means the period with DH supply temperature increased above 60°C drops to 18 hours and the maximum supply temperature drops to 62°C. However, considering 22°C as a realistic operative temperature desired by customers is crucial for a proper evaluation of the feasibility of supplying existing buildings with low-temperature DH. Underestimation of the operative temperature will result in underestimation of the maximum supply temperature needed, and therefore in complaints from customers. With the additional improvements on the building envelope, such as low-energy windows and ceiling insulation and the installation of low-temperature radiators, DH with a supply temperature of 50°C will ensure 22°C operative temperature during the whole heating season.

Table 4-1 compares the annual heating demand, the maximum heat power ($P_{max}$), the maximum supply temperature needed, and the average return temperature (TRW) from the SH system for all the cases simulated. It can be seen that the light (replacement of windows with standard ones) and extensive (low-energy windows and ceiling insulation) renovations reduce the peak heat power needed for the SH system.
by about 21% and 45% respectively, and the annual heat demand by 25% and 50% respectively. So the reduction in annual heat demand can be used only as a rough estimation for the reduction in peak heat power; both values are usually needed in relation to the connection of refurbished buildings to the DH network.

Table 4-1 - Comparison of heating demand, peak heat power ($P_{\text{max}}$) and weighted average return temperature (TRW) for all the cases investigated.

<table>
<thead>
<tr>
<th>T_{\text{operative}}</th>
<th>internal heat gains</th>
<th>T_{\text{supmax}}</th>
<th>RAD$^a$</th>
<th>$P_{\text{max}}$</th>
<th>TRW</th>
<th>heating demand reduction</th>
<th>heating demand reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>[°C]</td>
<td>[W/m²]</td>
<td>[°C]</td>
<td>kW</td>
<td>[°C]</td>
<td>MWh/a</td>
<td>[%]</td>
<td>[%]</td>
</tr>
<tr>
<td>non-renovated house - basic</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>20$^b$</td>
<td>0</td>
<td>70.0</td>
<td>O</td>
<td>9.2</td>
<td>37.6</td>
<td>x</td>
<td>-</td>
</tr>
<tr>
<td>20</td>
<td>0</td>
<td>70.0</td>
<td>O</td>
<td>9.4</td>
<td>40.2</td>
<td>x</td>
<td>-</td>
</tr>
<tr>
<td>20</td>
<td>4.18</td>
<td>73.0</td>
<td>O</td>
<td>9.9</td>
<td>30.1</td>
<td>20.0</td>
<td>-</td>
</tr>
<tr>
<td>22</td>
<td>4.18</td>
<td>78.0</td>
<td>O</td>
<td>10.5</td>
<td>32.9</td>
<td>24.6</td>
<td>-6%</td>
</tr>
<tr>
<td>22</td>
<td>4.18</td>
<td>67.3</td>
<td>LT</td>
<td>10.5</td>
<td>27.6</td>
<td>24.6</td>
<td>-6%</td>
</tr>
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<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>22$^c$</td>
<td>4.18</td>
<td>70.0</td>
<td>O</td>
<td>10.5</td>
<td>35</td>
<td>24.5</td>
<td>0%</td>
</tr>
<tr>
<td>22$^d$</td>
<td>4.18</td>
<td>78.0</td>
<td>O</td>
<td>10.5</td>
<td>33.1</td>
<td>24.6</td>
<td>0%</td>
</tr>
<tr>
<td>22$^e$</td>
<td>4.18</td>
<td>78.0</td>
<td>O</td>
<td>7.8</td>
<td>32.5</td>
<td>21.7</td>
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<td>0</td>
<td>70.0</td>
<td>O</td>
<td>7.7</td>
<td>33.0</td>
<td>x</td>
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<td>4.18</td>
<td>62.0</td>
<td>O</td>
<td>7.8</td>
<td>27.1</td>
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<td>21%</td>
</tr>
<tr>
<td>22</td>
<td>4.18</td>
<td>67.0</td>
<td>O</td>
<td>8.3</td>
<td>30.4</td>
<td>18.4</td>
<td>21%</td>
</tr>
<tr>
<td>22</td>
<td>4.18</td>
<td>59.0</td>
<td>LT</td>
<td>8.3</td>
<td>25.7</td>
<td>18.4</td>
<td>21%</td>
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<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>20</td>
<td>0</td>
<td>70.0</td>
<td>O</td>
<td>5.8</td>
<td>28.2</td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>4.18</td>
<td>53.1</td>
<td>O</td>
<td>5.47</td>
<td>24.7</td>
<td>9.9</td>
<td>45%</td>
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<td>57.0</td>
<td>O</td>
<td>5.80</td>
<td>28.1</td>
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<td>45%</td>
</tr>
<tr>
<td>22</td>
<td>4.18</td>
<td>50.0</td>
<td>LT</td>
<td>5.82</td>
<td>24.1</td>
<td>12.4</td>
<td>44%</td>
</tr>
</tbody>
</table>

$^a$: O = original radiators, LT = low-temperature radiators
$^b$: dimensioned on the basis of air temperature
$^c$: maximum flow limit increased to 400 L/h
$^d$: time delay in DH supply temperature control
$^e$: simulated with “measured weather data input”

Applying the weather file measured in the real location of the house in 2010 for the non-renovated house reduced the maximum heat power needed for SH by 25% and the annual heat demand by 12% in comparison applying the DRY weather file. The consequences of using air temperature instead of operative temperature when designing a SH system are shown to be 0.2kW at peak heat output, which is seen as
marginal, both for the SH and the DH system. However, using an operative temperature of 20°C instead of 22°C during the design phase leads to underestimation of the DH network requirements, would lead to people feeling thermal discomfort and asking the DH company to increase the DH supply temp.

With regard to the DHW system, once the DH supply temperature drops below 60°C, it will always be necessary for the original DH substation for DHW heating to be replaced with a specially designed low-temperature DH substation—depending on the original design, either one using the instantaneous principle of DHW heating or one with a storage tank for DH water (discussed in section 3.1.5). The existing DHW pipes will also need to be replaced with new pipes preferably with dimension DN10, to fit the requirement that the overall volume of DHW pipes is below 3L.
4.2 Low-Energy Buildings

The content of this chapter is based on the peer-reviewed conference paper “Optimal Space Heating System for Low-Energy Single-Family House Supplied by Low-Temperature District Heating” presented at the PassivHus Norden 2013 conference. However, it was found that the model for opening of the windows was set incorrectly, resulting in no window opening and thus overheating in the building. The set-point temperature, activating the bypassing of the heat recovery unit during over-heating was therefore reduced, resulting in up to 25% higher demand than with the right input parameters. The article itself is not part of the thesis, but the methodology used was correct and the thesis reports the corrected results.

4.2.1 Specific background

Space heating systems supplied by DH can be seen from two perspectives: the customer and the DH company. The perspective of customer can be expressed as the requirement for the desired indoor temperature at a good price, and the perspective of DH company is a requirement for a smooth and sufficient heat demand from the connected customers and high cooling of DH water.

As shown in chapter 2.3, low-temperature DH is also cost-efficient for low-energy houses thanks to its low heat losses. The main parameters in any cost-efficiency study are the connection heat power for DHW and SH (defining the dimensions of the DH network) and the expected heat demand of the buildings.

The connection heat power for a SH system in Denmark is based on the dimensioned heat loss of the building, calculated in accordance with standard DS 418 [59], for an outdoor temperature of -12°C, an indoor operative temperature of 20°C, and excluding internal and external heat gains. The contribution to the connection heat power for DHW depends on the type of DHW heater. For a substation based on the instantaneous principle of DHW preparation, it is 32.3 kW. For substations with a storage tank for DHW or a buffer tank for DH water, the connection power depends on the size of the tank. The bigger the tank, the lower the connection power that is needed. For example, for a 120L buffer tank for DH water, the connection power for DHW is only 3kW. Dimensioning of the DH network is therefore based on the aggregated connection heat power of the individual customers toward the heat source taking into account the simultaneity factors. The factors are considered separately for SH and DHW and for DHW they also depend on the type of DHW substation.

Heat demand is in general influenced by the construction of the building, the outdoor and desired indoor temperature, and internal and external heat gains. The internal heat gains and indoor temperature are expected to have considerable influence on the heat demand of low-energy buildings and any mistake in their estimation will therefore influence the feasibility study on the implementation of DH to the low-energy areas.

The annual heat demand of the specific building is calculated using the Danish national calculation tool, Be10, which is also used for checking the energy efficiency of the buildings. The annual heat demand for residential buildings covers the energy needed for heating, cooling, running of HVAC systems (fans, pumps, etc.) and DHW. The values suggested are internal heat gains 5W/m², an operative temperature of 20°C, and energy use for DHW of 13.1 kWh/m².a (based on 250L/a.m2 and a temperature lift from 10°C to 55°C). These values, however, are meant to be values for comparing the energy performance of different buildings and not treated as values reflecting the realistic heat demand of the building. Nevertheless, many engineers take these values as input for the calculation of the expected energy use of a building.
So it is not surprising that the real energy demand is higher than the one originally calculated. The reason is simply that people use the house in a different way than was assumed for the energy calculation, or to be more precise, the energy calculation used input values different from realistic ones. The occupants are the most important element. [7] found that occupants tend to maintain an operative temperature of 22°C instead of 20°C, and according to [61] the suggested value for DHW use is 800 kWh/(a. person). For a 159 m² house with 4 people, this results in an energy demand for DHW of about 20 kWh/(m².a) instead of 13.1 kWh/m², and the improved energy efficiency of currently used equipment reduces the internal heat gains from 5 W/m² to 4 W/m² [62]. All these differences mean higher heat demand.

However, the problem is not just the disillusion of building owners on receiving higher energy bills than promised; underestimated heat demand can lead to wrong conclusions from the cost-efficiency study on the suitability of using DH in the first place. Therefore we investigated the influence of using more realistic values for internal heat gains and indoor temperatures on the heat demand in low-energy houses.

Furthermore we investigated the performance of an SH system with radiators supplied by low-temperature DH from the perspective of the customer, represented by requirements for thermal comfort, and the perspective of the DH company, represented by a smooth heat demand and a low return temperature of DH water.

4.2.2 Methods
We built a multi-zone model (12 rooms = 12 zones) of a 159 m² low-energy house class 2015 [55] in IDA-ICE (see Figure 4.3). Then (defining case “01”) we dimensioned the heating system in accordance with the standard DS 418 [59], i.e. with a constant outdoor temperature of -12°C, an indoor operative temperature of 20°C, and excluding internal and external heat gains, to get the connection power for SH. We chose an SH system with radiators dimensioned for the designed conditions at 55/25/20°C (supply/return/operative temperature).

![Figure 4.3 – Ground plan of 159m² single-family house built in IDA-ICE](image)

The air flows for individual rooms were designed on the basis of BR10 requirements, resulting in a total flow rate 60 L/s, both for supply and exhaust. The performance of the heat recovery unit in the ventilation system was set to 76% based on the long experience of a building company [63], meeting the requirement of minimal heat recovery efficiency of 75% defined in BR10 [55]. The heat recovery unit starts to partially bypass the air (frost protection) when the temperature of the exhaust air after the heat recovery drops below 1°C. The temperature of the air supplied by the ventilation system was 4°C below the desired
temperature set-point in the room, i.e. 16°C and 18°C for an indoor temperature of 20°C and 22°C, respectively. For periods when the temperature of the air supplied after heat recovery dropped below the minimum value, a heating coil was activated. The heating coil was dimensioned to lift the air temperature from -15°C to 16°C, i.e. without taking the heat recovery unit into account.

The influence of the internal heat gains and operative temperature on the maximum heat output needed from the SH system and the annual heat demand was investigated for the Danish design reference year (DRY) for four cases, as reported in Table 4-2. The internal heat gains were first modelled as a constant value of 5W/m², i.e. in accordance with the values suggested in Be10; this is represented by case “A”. Then we defined our own schedule of internal heat gains, resulting in an average value of 4.18W/m² and repeated the simulation with constant heat gains of 4.18W/m²; this is denoted case “B”. The next modification was to implement heat gains with the same average value, but modelled as real activities, i.e. changing the values over time; see case “C”. The schedules for the activities are defined in [53]. The last modification investigated (case “D”), changed the desired indoor temperature to 22°C instead of 20°C. Then we compared the maximum heat output, heat demand and length of space heating period for all the cases investigated.

### 4.2.3 Results and Discussion

Table 4-2 compares the results for all the cases investigated. For easier orientation, colours express the magnitude of difference compared to the reference case, lowest (green) and highest (red), in each column.

It can be seen that the SH system based on radiators and the heating coil (HC), dimensioned classically on the basis of DS 418, needs 5.1 kW connection power (case 01). The radiators were chosen very close to required dimensioned heat output, which can be seen from the return temperature of 25°C. Increasing the indoor set-point temperature to 22°C resulted in a 4°C increase in the return temperature.

**Table 4-2 – Heat demand and heat power for all cases simulated**

<table>
<thead>
<tr>
<th>Case</th>
<th>Int. Heat Gains</th>
<th>Delivered Heat</th>
<th>Over-dimensioning</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[W/ m²]</td>
<td></td>
<td>P&lt;sub&gt;max&lt;/sub&gt; RAD</td>
</tr>
<tr>
<td>01*</td>
<td>0 - -</td>
<td>2948</td>
<td>0%</td>
</tr>
<tr>
<td>A</td>
<td>5 C 2810 138</td>
<td>2948</td>
<td>0%</td>
</tr>
<tr>
<td>B</td>
<td>4.18 C 3386 139</td>
<td>3525</td>
<td>20%</td>
</tr>
<tr>
<td>C</td>
<td>4.18 S 3519 138</td>
<td>3657</td>
<td>24%</td>
</tr>
<tr>
<td>D</td>
<td>4.18 S 4461 186</td>
<td>4647</td>
<td>58%</td>
</tr>
</tbody>
</table>

C = constant; S = scheduled; a outdoor temperature -12°C, no HG; b T<sub>ave</sub>-point 20°C/22°C; c T<sub>ave</sub>-point = 22°C; HC = heating coil; RAD = radiators; HC = heating coil; P<sub>max</sub> = maximal heating output; HREC = heat recovery; T<sub>ave</sub> weight AVG = averaged return temperature

For case “A”, the connection power drops by 33% to 3.5 kW, caused partly by the over-dimensioning of the radiators (25%) and partly by the over-dimensioning of the heating coil (44%). The over-dimensioning is the result of the fact that the radiator design was based on a constant outdoor temperature of -12°C and-15°C for the HC dimensioning, without consideration of heat recovery and excluding internal heat gains.
Reducing the constant internal heat gains from 5 W/m² to 4.18 W/m² represented by case “B” results in a considerable increase of annual heating demand by 20% and a heating period that is 8 days longer. Modelling of internal heat gains intermittently based on the detailed schedule, results in an additional 4% increase in heat demand and a heating period 5 days longer. Increasing the desired indoor temperature from 20°C to 22°C results in an additional heat demand of 27%, which is in accordance with results of Tommerup [62] and Olsen [64]. Regarding the reference case based on Be10 values, the heat demand increased by 58% and heating season by 42 days (1.4 months).

Including a more realistic demand for DHW heating (see Table 4-3) than the values predicted by Be10 calculations, the total heat demand for case “D” increased by 56%, which is a considerable difference from the perspective of both customers and the DH company.

### Table 4-3 – influence of DHW on total heat demand

<table>
<thead>
<tr>
<th>Case</th>
<th>Delivered Heat</th>
<th>SH &amp; DHW</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[kWh/(m².a)]</td>
<td>[kWh/a]</td>
</tr>
<tr>
<td>01</td>
<td>54%</td>
<td>-</td>
</tr>
<tr>
<td>A</td>
<td>13.1</td>
<td>2083</td>
</tr>
<tr>
<td>B</td>
<td>20.1</td>
<td>3200</td>
</tr>
<tr>
<td>C</td>
<td>20.1</td>
<td>3200</td>
</tr>
<tr>
<td>D</td>
<td>20.1</td>
<td>3200</td>
</tr>
</tbody>
</table>

Comparing the connection power calculated for SH system based on the standard dimensioning approach with the results of cases “C” and “D”, the original value 5.1 kW decreased by 1.1 and 1.0 kW, which is a decrease of approx. 20%. The fact that in traditional DH networks the connection power for SH is usually increased by another 20-30%, results in a total 30-40% over-dimensioning of SH connection power. On the one hand, this will ensure free capacity to connect more customers, but on the other hand, it will result in unnecessarily large DH pipes with higher heat loss, which is contrary to the philosophy of the low-temperature DH concept. Interestingly, the heat output from the radiator system designed on the basis of dimensioning heat loss for DS 418 with an indoor temperature of 20°C has roughly the same value as the heat output needed in case “D” for 22°C and reduced internal heat gains, and therefore gives favourable conditions for the occupants.

The heat demand on the DH network is smooth for all the cases simulated, without large oscillations (see Figure 4.4). The reason for this is the control of each individual radiator by a P-controller (mathematical model of traditional thermostatic valve with thermostatic head), which reacts immediately to changes of the heating demand. However, this is not the case for floor heating, which is usually controlled by on/off valves resulting in rapid oscillations in heat demand from the DH network [53].

42
Figure 4.4 – Heat load on DH network from SH system with radiators

Figure 4.4 shows the difference between modelling internal heat gains as continuous (cases A and B) and scheduled (cases C and D). But from the perspective of the DH network these oscillations are negligible.

With regard to the ventilation system, a traditional ventilation system with heat recovery should have a heating coil. For exhaust air with a temperature of 20°C, a heat recovery unit with 76% thermal efficiency can heat up outdoor air to 15°C (air supplied in the rooms is expected gain 1°C from the fan and heat gains in the distribution pipes) theoretically with a minimum outdoor temperature of -0.8°C, or, if it has a thermal efficiency of 85%, with an outdoor temperature of -13.3°C. However, in reality the heat recovery unit should activate frost protection, partially bypassing the inlet air when the temperature of the outlet air drops below 1°C, which means that the heating coil is already needed for an outdoor temperature of -2°C and -5°C for recovery efficiencies of 76% and 85%, respectively. It might be possible to use an additional pre-heater inserted before heat recovery, e.g. supplied by return water from the radiator system, but this system was not tested.

If the heating coil is designed for -15°C and no heat recovery (i.e. a temperature lift from -15°C to 15°C), this leads to approx. 45% over-dimensioning in comparison with a design with heat recovery. The heating coil is available for extremely cold weather, but on the other hand is unnecessarily big and expensive. The heating coil covers a small part of the heat demand, to prevent the supply of air colder than 4°C from the indoor set-point temperature and thus represents around only 4%. So it would be better to use electric heating coil rather than hydronic heating coil, because this would reduce the connection power, investment cost and complexity of the SH system. It is true that from the perspective of exergy this is not an ideal solution, but on the other hand recent increases in the share of RES for electricity production are slowly changing the mind set in favour of electric heating, making this solution more acceptable. However, it should be stressed that for electricity use, a primary factor of 2.5 should be applied, and not 0.8 for DH.
4.3 General Conclusions from Part II

Low-temperature DH can supply space heating systems in low-energy and most existing buildings in Denmark. Since existing buildings have greater heat demand and are equipped with space heating systems designed originally for traditional DH, in very cold periods the DH supply temperature should be increased to the level reflecting the refurbishment carried out on the individual buildings.

The use of input values for internal heat gains and indoor temperature suggested by recent literature results in an increase of approx. 60% in the expected heat demand compared to the values suggested by Be10. This is important because Be10’s underestimation of heat demand makes other individual heating solutions look more competitive than DH.
5 CONCLUSIONS

Low-temperature DH for DHW supply

The laboratory test confirmed that a low-temperature DH substation supplied with 50°C warm water can produce the required 47°C DHW, while cooling the DH water to 20°C. However, the numerical model showed that the delivery time of DHW is slightly above the 10 s suggested by the standard, due to the influence of the DH network even when the service pipes are kept warm using a bypass solution.

Numerical simulations showed that the proposed solution of redirecting the bypassed water during the non-heating period to floor heating in the bathroom reduces heat loss from the DH network by 13% while the floor temperature increases on average by 2.2°C. The heat loss saved in the DH network corresponds to 40% of the heat delivered to the bathroom and therefore it seems reasonable to bill customers for heat used in bathroom floor heating during the non-heating period at a discounted price.

Space heating systems supplied by low-temperature DH in existing buildings

Most existing single-family houses built after the 1970s can be supplied with low-temperature DH with a temperature of 50°C for almost 90% of the design reference year without exceeding the design water flow rate from the DH network while heated up to an indoor operative temperature of 22°C. During the rest of the year, the supply temperature should be increased in accordance with the non-linear supply temperature curve, for our case up to 67°C. Lowering the indoor set-point temperature to 20°C, installation of low-temperature radiators or refurbishment of the building will reduce the maximum temperature needed in the space heating system and extend the period when the DH supply temperature can be 50°C.

Reducing the DH supply temperature to 50°C requires the replacement of currently used substations with low-temperature DH substations to ensure the production of DHW with the required temperature and without increased risk of Legionella.

To ensure a gradual transition to a temperature level of 50°C in the shortest possible period, DH companies should require that all new installed and refurbished DH substations are already designed for low-temperature DH.

Space heating systems supplied by low-temperature DH in low-energy buildings

A low-energy single-family house equipped with a low-temperature space heating system with radiators can be supplied from a low-temperature DH system and ensure the required thermal comfort while maintaining a smooth heat demand on the DH network and cooling the heating water to a low return temperature.

Compared to the input values traditionally used, more realistic data for internal heat gains and indoor temperature reported in the literature result in as much as 58% higher heat demand and in up to 40% lower connection power for space heating. Use of the low values traditionally used negatively influences cost-effectiveness analyses for planned DH systems, so careful consideration of the correct input values is important for the future successful expansion of DH systems.
General conclusion

In general, it can be concluded that both low-energy and existing buildings can be supplied using low-temperature DH with a supply temperature of 50°C and still meet the requirements for DHW and thermal comfort if the DH substation, the DHW system and the space heating system are designed for low-temperature operation. However, to ensure the required thermal comfort for occupants in existing buildings while not exceeding the designed flow and return temperatures of DH water, the DH supply temperature should be increased above 50°C during cold periods, whose length will be defined by the space heating system and the refurbishment measures carried out on the building.
6 FURTHER WORK and RECOMMENDATIONS

Further work and recommendations can be summarised under the following headings:

DHW systems and Legionella

- The combination of uncertainty about Legionella and the respect of DH operators for Legionella bacteria could be a showstopper for low-temperature DH with a supply temperature (at the inlet to the substation) of 50°C. If this happens, the supply temperature should be increased to 55°C to ensure that the minimal DHW temperature produced by the LTDH substation is 50°C to comply with current DHW standards.
- Most research on the risk of Legionella has been carried out in old fashioned DHW systems where the problems are expected to be found. There is a lack of knowledge about the risk in DHW systems with a low volume of DHW and therefore more research is needed.
- To improve energy efficiency and avoid the risk of Legionella, it is advisable to stop using DHW circulation. Keeping DHW circulation will require an increase in DH supply temperature to at least 65°C to ensure that DHW returns from the circulation with 55°C. However, many DH companies already today guarantee only a DH supply temperature of 60°C at the substation, which means that the temperature from DHW circulation must be below 55°C in many houses already today.
- An alternative solution to the advanced oxidation technology needed to eliminate Legionella in a DHW system with DHW temperature below 50°C and water volume above 3L could be to use an electric heater, increasing the DHW temperature locally to the level of thermal disinfection (60-70°C), followed immediately by a heat recovery system to cool the disinfected DHW to 45°C by transferring the excess heat to the new DHW just coming from the circulation loop. We are not aware of the existence of any such solution on the market.

DH substations

- Reducing the number of plates in the DHW HEX from 40 to 20 will reduce the recovery time of the substation by about 1 s, while the increase in the return temperature of DH will be marginal. A similar reduction in mass could be applied to the pipes and fittings in the substation, e.g. by integration of individual fittings and/or by using plastic components. Any of these solutions will reduce the price of the substation and thus improve DH’s position on the market.
- An internal bypass solution is currently applicable only as part of the IHPT DHW controller. However, IHPT controllers as designed today cannot operate with a DH supply temperature below 50°C. In view of the trend of insulating DHW HEX, the recommendation is to develop and test solutions for an internal bypass also for low-temperature house substations. However, national DHW regulations should be studied first to see if this solution can be applied.
- Reducing the recovery time for the substation could be also realised by reducing the DHW flow at the beginning of tapping. Using a self-acting controller to implement this solution would be very difficult, but the solution could be realised electronically, going hand-in-hand with the requirement that all new heating systems installed after 2014 have to be equipped with an electronic controller for weather compensation.

Comfort Bathroom

- The recommendation is to develop and implement an electronic valve for controlling the bypass flow, reducing the volume of bypassed water.
• The recommendation is to make a more detailed model of floor heating in the bathroom, reflecting various pipe layouts. This can be done in IDA-ICE by dividing the bathroom into small segments and connecting them hydraulically based on the pipe layout. The individual segments should be connected for horizontal thermal transfer.

Low-temperature DH in buildings
• The recommendation is to investigate the performance of direct space heating system with radiators without a mixing loop (as in the Lystrup show case). A constant supply temperature of 50°C and low heat demand during moderate outdoor temperatures can lead to very small flows of heating water, which can cause problems for radiator thermostatic valves.
• The recommendation is to compare different space heating systems supplied by low-temperature DH, from the perspective of both the DH company and the customer.
• The recommendation is to investigate the feasibility of preheating the inlet air before it enters the heat recovery unit using heating water returning from radiators.
• DH companies should be stricter in reducing the DH supply temperatures because high temperatures are often required due to malfunctions in individual in-house systems.
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ISI paper I

Numerical modelling and experimental measurements for a low-temperature district heating substation for instantaneous preparation of DHW with respect to service pipes

Brand M, Thorsen J E, Svendsen S

Numerical modelling and experimental measurements for a low-temperature district heating substation for instantaneous preparation of DHW with respect to service pipes

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ABSTRACT

Traditional district heating (DH) systems are becoming uneconomic as the number of new and renovated buildings with reduced heating requirements increases. To keep DH competitive in the future, heat losses in DH networks need to be reduced. One option is to reduce the supply temperature of DH as much as possible. This requires a review and improvement of a DH network, in-house substations, and the whole domestic hot water (DHW) supply system, with the focus on user comfort, hygiene, overall cost and energy efficiency. This paper describes some practical approaches to the implementation of low-temperature district heating (LTDH) with an entry-to-substation temperature around 50 °C. To this end we developed a numerical model for an instantaneous LTDH substation that takes into consideration the effect of service pipes. The model has been verified and can be used for the further optimization of the whole concept as well for individual components. The results show that the way that the service pipe is operated has a significant effect on waiting time for DHW, heat loss, and overall cost. Furthermore, the service pipe should be kept warm by using a bypass in order to fulfill the comfort requirements for DHW instantaneously prepared.

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1. Low-temperature district heating

District heating (DH) is a way of providing buildings with heat for space heating (SH) and domestic hot water (DHW) preparation in an economical and environmentally friendly way. Nowadays, building regulations have been introduced worldwide and are helping to reduce energy consumption in buildings, because 40% of all energy consumption takes place in buildings. The energy policy of the European Union is currently focused on security of supply, energy savings, reducing production of CO2, and increasing the proportion of renewable energy [1]. DH is one of the best ways to achieve these goals in the building sector and its further development has high priority. According to Heat Plan Denmark 2008, DH systems supplied by renewable energy sources could cover up to 70% of heating demand in Denmark by 2050. The remaining 30% is assumed to be covered by individual heat pumps, installed mainly in areas with low heat demand density [2,3]. Nevertheless, traditional high and medium temperature DH systems are not optimal solutions for the future [4]. To accord with EPBD [5], all buildings constructed after 2020 must be near Zero-Energy Buildings [6] and this will create areas with lower heat demand than today. Table 1 shows the implementation of EPBD for newly built residential buildings in Denmark valid since 2008 [7] and the update in 2010 [8]. The values of the maximum specific primary energy demand, account for space heating, DHW heating and electricity for operation of HVAC systems (pumps, fans).

The DH networks in current use will not be able supply heat to these areas in an economical way, because the ratio between network heat losses and the heat consumption in the buildings would be unacceptably high, the cost of heat for the end-users would increase, and DH systems would cease to be competitive with other solutions, such as heat pumps [9]. Research in the field of DH has recently focused on the supply of areas with low heat demand and low-energy buildings [10] and on increasing the proportion of heat produced by renewable sources of energy, such as solar heat plants, geothermal energy, or heat pumps driven by electricity from renewable sources. Attention has also been paid to the smart grid concept, where buildings connected to DH network
are not only consumers of heat, but can also supply any surplus heat back to the network [11]. The solution for the future development of DH is to reduce the heat losses of DH networks by means of pipes with better insulation properties, to improve network design with reduced pipe diameters and the use of twin pipes [10,12], and to reduce the supply temperature of district heating water to the lowest possible level, and thus match the energy levels of supply and demand [13]. The District Heating Systems designed on the basis of this philosophy are called Low-Temperature District Heating (LTDH) Systems.

1.1. Full-scale demonstration of LTDH

The LTDH concept was first reported in [10], where a theoretical case study documented that LTDH is a good solution for the future and is fully competitive with heat pumps even in sparse housing areas. In Denmark, the first residential area with low-energy houses supplied by LTDH was built in 2009. After 2 years’ experience, the results were documented in [14] and agree with the theoretical calculations. The system supplies 40 Class 1 low-energy houses (for definition see Table 1) with district heating water on the design conditions of 50 °C/25 °C. The DH network was built with highly insulated twin pipes with series 2 insulation, which in combination with the reduced diameter of the media pipes has resulted in an average heat loss of only 17%. The pipes are commercially available steel pipes or AluFlex pipes; the AluFlex media pipes are made of PEX/PE layers with an aluminum foil between preventing water diffusion, PUR insulation foam with λ-value ranging from 0.022 to 0.023 W/(K m) and high density PE casing. A network design, specification and complete list of pipes used in Lystrup can be found in [14,15]. The project demonstrates that LTDH is a good solution even for low-energy houses. Two types of newly developed low-temperature district heating in-house substations have been tested by customers in real conditions: 29 Instantaneous Heat Exchanger Units (IHEUs) [16], and 11 District Heating Storage Units (DHSUs). The IHEU concept is a classical substation with an instantaneous heat exchanger (HEX) that has been modified for operation with low supply temperature, with an improved plates pattern and enlarged surface area in the HEX. To supply DHW with a short waiting time, the IHEU is equipped with an external bypass with a set-point temperature of 35 °C. The DHSU concept differs from IHEUs in having a buffer tank for the storage of district heating water, but the DHW is produced using an instantaneous principle as in IHEUs [17]. The advantage of the substation having a buffer tank lies in the heat load on the DH network being averaged out, which allows reduced pipe dimensions in DH network resulting in lower heat loss. On-site measurements evaluating the long-term performance of both types of DH substation are available. What we still lack are the detailed measurements of the short time-steps needed to evaluate the dynamic performance of the substations. These data are needed to verify the numerical model to be used for the further optimization of units and whole LTDH concept. This paper describes the LTDH concept with focus on the DHW supply and reports on the dynamics of IHEUs, including service pipe (SP) operation mode, as investigated using the numerical model we developed.

2. LTDH for DHW supply

2.1. Elimination of Legionella risk by using a system with a minimal volume of DHW

Since LTDH was mainly developed for low-energy buildings already designed with low-temperature space heating systems, the lowest acceptable forward temperature is defined by the requirement to prepare DHW with the desired temperature without needing additional heating. The hygienic requirements in the latest guidelines [18] are to produce DHW with temperature of 50 °C for single-family houses and 55 °C for multi-storey buildings to ensure a minimum temperature of 50 °C at the tap. In DHW systems with circulation, the temperature of re-circulated water should never fall below 50 °C. These requirements are based on the need to avoid Legionella growth in DHW pipes and storage tanks. Legionella grow in a temperature range between 20 °C and 46 °C in systems with a high volume of stagnating water, so the above-mentioned temperature levels are necessary both to ensure comfort and to meet hygienic requirements in the tap furthest away from the heat source. Nevertheless we see a high level of discrepancy between national standards and the results of research focused on Legionella. In the literature, the danger of Legionella growth in DHW systems is affected by the temperature of DHW, the nutrients in DHW, the flow type (laminar or turbulent) in the DHW pipes, and water stagnation [19]. On-site measurements have been performed in buildings using DHW for heating, and the results [20,21] show that Instantaneous Heat Exchanger Units (IHEUs) have fewer problems with Legionella than traditional units with a DHW storage tank. Both studies agreed that these findings are due to the fact that IHEUs usually produce DHW with a temperature of 60 °C, whereas in storage units the temperature is 50 °C or lower. Another reason for the higher occurrence of Legionella in DHW plants with storage tanks is a high volume of stagnating water, which results in sedimentation on the inner surface of the tank creating conditions for Legionella growth. Nevertheless, based on the studied literature, it can be concluded that the requirements to produce DHW with a temperature higher than 50 °C are only defined for DHW plants with a storage tank for DHW and old fashioned DHW building installations, which can be characterized as systems with vertical risers, branched pipes with large diameters (increasing the water volume of the system), and using DHW circulation. Such systems tend to hydraulic misbalance, which results in places with a reduced temperature and thus a risk of Legionella growth.

For new and renovated buildings, DHW installations can be designed in a much better way, with individual connection of DHW feeding pipes between the source of DHW and each tap, and with optimally reduced pipe diameters, defined by requirements for noise propagation and pressure drop. Such a solution reduces the volume of DHW in the system and, according to German Standard W51 [22], the temperature of DHW can be below 50 °C with no risk of Legionella promotion, if the total volume of the DHW system excluding HEX is less than 3 L. This allows the use of the LTDH concept. For newly built multi-storey buildings, the state-of-the-art solution is a district heating substation for each flat [23]. In this case, each flat has its own individual system for DHW preparation (with a volume of water below 3 L) and space heating. Such a solution is very different from the traditional huge DHW systems.
in which DHW circulation was used to reduce both waiting time for DHW and the growth of Legionella, and in which Legionella sometimes arose anyway because of hydronic misbalance [24]. As far as we know, no DHW systems using IHEUs with a volume of less than 3L and producing DHW with a temperature below 50 °C have been investigated in relation to Legionella risk.

2.2. User comfort in DHW supplied by LTDH

In addition to the hygienic requirements, a DHW system should also fulfill requirements for comfort defined by the desired level of temperature and waiting time for DHW. According to Danish Standard DS439 [25], DHW provided with a nominal flow rate of 0.2 L/s and a desired temperature level of 45 °C in the kitchen and 40 °C in other taps should be delivered within a “reasonable” length of time and without significant temperature fluctuations.

To comply with DS439, our suggested value for waiting time for DHW is 10 s (with a flow rate of 0.2 L/s) to avoid wasting of water and to provide users with DHW within a reasonable time. Waiting time for DHW can be studied from various perspectives. From the dynamic point of view, waiting time depends on transportation time and the thermal capacity of the components, i.e. pipes and substation. From the point of view of location, delays arise in three areas: the SP (the pipe connecting the street pipe with the in-house substation), the DH substation, and the DHW supply system in the building. Time delay in the SP and substation is related to the DH network, the substation type, and its control strategy, while time delay in the DHW supply system in buildings without DHW circulation is determined by the length and thermal capacity of pipes, the volume of water in individual pipes, the nominal flow, and to some extent also by the pipe insulation.

2.2.1. Waiting time for DHW in building supply system

For DHW systems with individual feeding pipes and an overall volume of pipes less than 3 L, DHW circulation is not usually needed, because transportation time is not critical with the short distance from the substation. Table 2 lists the transportation times for the individual fixtures in a typical house built in the pilot LTDH project, Larch Garden at Lystrup in Denmark [14]. It should be emphasized that the data are for transportation delays only, ignoring the thermal mass of the pipes. The table shows that a reasonably designed system, with fixtures close to each other and not far from the substation, gives a maximum transportation delay of about 6 s for the wash basin with nominal flow rates for individual tapping points. If we use a flow rate of 0.2 L/s (as suggested in [25]), the transportation delay decreases to 1.6 s. However, the suggestion of using 0.2 L/s for all tapping types is questionable. The overall volume of pipes in the reference DHW supply system is 0.98 L, which means it is possible to install longer pipes or more fixtures and still fulfill the requirement of a DHW system with a volume of less than 3 L. According to [22] the volume of HEX is not counted in the 3L volume. The velocity of water is below 2 m/s and thus problems with noise propagation during tapping are avoided.

### Table 2

<table>
<thead>
<tr>
<th>DHW fixture</th>
<th>Nominal flow [L/min]</th>
<th>Length to fixture [m]</th>
<th>Volume in pipes [L]</th>
<th>Velocity [m/s]</th>
<th>Transportation delay [s] for Nominal Flow [L/min]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shower</td>
<td>8.4</td>
<td>2.2</td>
<td>0.17</td>
<td>1.8</td>
<td>1.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.9</td>
</tr>
<tr>
<td>Basin</td>
<td>3.4</td>
<td>4.1</td>
<td>0.32</td>
<td>0.7</td>
<td>5.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.6</td>
</tr>
<tr>
<td>Kitchen</td>
<td>6.3</td>
<td>6.3</td>
<td>0.49</td>
<td>1.3</td>
<td>4.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2.5</td>
</tr>
</tbody>
</table>

2.2.2. Delay for DHW on the primary side

Delays on primary side consist of delay in the DH substation, the so-called recovery time and, for substations with an instantaneous principle of DHW heating, i.e. IHEUs, delay in the SP. For substations equipped with storage tanks, i.e. DHSU, the HEX is supplied direct from the buffer tank so that the SP has no effect on the recovery time.

2.2.2.1. Recovery time of the substation

The recovery time of a DH substation is the time needed for the substation to produce DHW with the desired temperature. It is defined by the type of substation, controller, the thermal capacity of the components (HEX, valves, pipes, fittings), the water volume of the HEX, the maximum allowed primary flow rate, and to some extent by the insulation of the substation and the time since the previous tapping.

Since substation recovery time is too complex for simple theoretical evaluation, we measured the performance of the IHEU prototype specially developed for LTDH project at Lystrup [14] in laboratory conditions, and then developed a numerical model of a substation equipped with a PTC2+P controller.

2.2.2.2. Service pipes

The simplest SP control strategy is a solution without a bypass (see Fig. 3). In this case, DH water in the SP flows only when customer has a demand for DHW or SH, and this means that during periods without heating demand (so-called idling), the SP cools down to ambient ground temperatures and the substation to the temperature of the technical room.

Fig. 1 shows the temperature drop in an SP AluFlex 20/20/110 (used for connection of the IHEUs) after tapping of DHW has finished and the substation is idling. The simulation was made using the commercial software COMSOL Multiphysics and reported by Dalla Rosa [26].

The cooling of the SP was simulated for an idling period from 0 to 720 min and was simulated for three different ground temperatures (3 °C, 8 °C and 14 °C) with and an initial temperature of pipe insulation of 15 °C (average value for typical conditions in Denmark). It can be seen that for periods without SH (i.e. for a low-energy house approximately 6 months of the year from April to October) with a typical average ground temperature between 8 °C and 14 °C, the water standing in the SP is cooled from 50 °C down to 20 °C if there is no tapping of DHW or bypass flow after about 3 and 4 h for the ground temperatures of 8 °C and 14 °C respectively.

![Fig. 1. Cooling down of DH water standing in an AluFlex 20/20/110 service pipe during idling. The initial temperature of water in pipe is 50 °C, and the initial temperature of insulation is 15 °C.](image-url)
At the beginning of each tapping, water standing in the SP has to be transported to the substation first before the “fresh” DH water with a temperature of 50 °C can enter the substation.

2.2.2.3. The effect of thermal capacity on delay in the service pipe. Flow-compensated temperature controllers with integrated differential pressure control are state-of-the-art for DHW production in substations. The philosophy of the controller is to allow the maximum possible flow rate on the primary side until the set-point temperature on DHW is reached [27]. Maximum flow on the primary side is defined by the \( k_p \)-value of the control valve and by the differential pressure available at the substation. Moreover, a flow restrictor limiting the maximum primary flow can be installed by the DH company. At Lystrup, the maximum flow rate on the primary side is 17.3 L/min. The IHEU is supplied by an SP with an inner diameter of 15 mm, i.e. for an SP with a length 10 m it takes 6.1 s before new water from the street pipe reaches the substation.

This simple calculation of transportation time in the SP does not take into account the effect of the thermal capacity of the SP. The impact of the thermal capacity of the pipe material on the delay has been numerically simulated with a Matlab code developed in-house and successfully validated with experimental data [28]. The impact of the thermal capacity of AluFlex 20/20/110 on a temperature drop in water supplied into the IHEU is shown in Fig. 2. The curves represent the temperatures measured at the inlet of the substation for a 10 m long AluFlex 20/20/110 pipe surrounded by ground with a temperature of 8 °C after 180 min of idling (i.e. the initial temperature of the water is around 20 °C, see Fig. 1) when DHW tapping starts. In this paper, we assume that the temperature of “new” DH water entering the SP during tapping or bypassing is 50 °C in all the cases simulated. This allows us to ignore any temperature drop in the DH network before water is supplied to the SP.

Fig. 2 shows that if the water in the SP has cooled to 20 °C, the transportation time for a flow rate of 17.3 L/min accounts for 6.1 s (independent of temperature), but the water delivered to the HEX at that time has already been cooled by the thermal capacity of the pipe to 38.2 °C. It takes another 2 s, i.e. 8 s in all, to deliver 45 °C water and around 32 s to deliver 49 °C to the HEX. Fig. 2 also shows the temperature development for an initial water temperature of 35 °C and for 20 °C with flows of 14.1 and 8.4 L/min.

The results presented show that during a summer morning (when the SP is cooled almost to the temperature of the ground), it takes 8 s to supply warm DH water to the substation and another 1–5 s to supply the DHW produced to individual taps without taking the recovery time of the substation into consideration.

2.2.3. Solutions reducing waiting time on the primary side

To decrease the time needed to supply “fresh” DH water to the substation, a solution with an external bypass (see Fig. 3) is widely used. In this case, the substation is equipped with a thermostatic valve that keeps water in the SP at a certain temperature level by “bypassing” a small amount of water directly back to the DH network and thus reducing waiting time substantially. The set-point temperature for the external bypass in LTDH is normally between 35 °C and 40 °C as a compromise between insufficient cooling of by-passed DH water and additional heat consumed by the customer and reduced waiting time for DHW preparation. The set-point temperature for traditional DH is higher.

Another way to reduce waiting time for DHW is an internal bypass (see Fig. 3). In contrast to the external bypass, the bypassed water flows through the HEX and keeps it warm. This looks like a good solution for the customer, but since water returning to the DH network has quite a high temperature, it is not desirable from the perspective of the DH network and the customer may be charged for insufficient cooling. This solution is not considered a good solution for LTDH, so we do not elaborate more on it.

3. Methods

3.1. Experimental measurement of IHEU

3.1.1. Experimental setup and instruments

The IHEU is a type of district heating substation that consists of a HEX without a storage tank. DHW is produced instantaneously in the HEX only it is tapped. It is then supplied directly to DHW taps. The IHEU was equipped with an external bypass and a PTC2-P controller for DHW preparation. The PTC2-P is a flow-compensated temperature controller with integrated differential pressure control [27]. The space heating system uses direct connection without the heat exchanger. Compared to a traditional IHEU, the low-temperature unit has a HEX (X37H-40) using plates with dimpled pattern which ensure optimal operation with a lower supply temperature. The water volume on both the primary and secondary sides was 1.1 L and the heat exchanger was not insulated. The experiments carried out focused on the dynamic and static performance of the substation in relation to DHW production, so no space heating loop was connected and the space heating valves in substation were closed. The temperature requirements for the DHW prepared were in accordance with those specified in DS439. We measured the temperature of four water flows passing through the substation and the primary and secondary flow rates. On the primary side, it was the temperature of the DH water supplied to the substation \((T_{11})\) and the temperature of the DH water returning back to the DH network \((T_{12})\), and on the secondary side, it was the temperature of the cold water entering the substation \((T_{21})\) and the temperature of the DHW produced \((T_{22})\). All temperatures were measured by type T thermocouples installed directly in pipes, in the water stream, so there is no practical time delay for the measurements. The time constant to reach 90% of step change was less than 1 s. The distance of thermocouples from the substation flanges was 5 cm and thermocouples were calibrated beforehand. We also measured the air temperature in the testing room. Temperatures were measured and collected by a multifunction acquisition unit (Agilent 34970A) every second. The DH network with a constant temperature of 50 °C was simulated by the university DHW system, which has a large enough capacity to supply 50 °C continuously. Moreover, a thermostatic fixture was used to keep the supply

![Fig. 2](image-url). Effect of the thermal capacity of an AluFlex 20/20/110 service pipe on the temperature of water delivered after substation idling.
temperature to the substation at 50 °C. To prevent pipes supplying DHW to laboratory cooling down in periods when there was no flow through the substation, a small guard flow drained directly to a sink just before the entrance to the substations was kept to maintain DHW at a constant 50 °C.

3.1.2. Experimental procedure

The DHW controller (PTC2+P) was adjusted to provide 47 °C on the DHW side with a supply temperature of DH water at 50 °C. The substation was left idle for a long time so that whole parts of the unit including water in the HEX were at room temperature, i.e. substation recovery time needed to deliver DHW with a temperature to the substation at 50 °C. Then we opened a tap on the DHW side and measured the DHW recovery time needed to deliver DHW with a temperature of 40 °C. Then we opened a tap on the DHW side and measured the substation recovery time needed to deliver DHW with a temperature of 40 °C and 45 °C at the substation’s outlet from the moment when DHW tap was opened. The DHW water flow was 8.4 L/min, i.e. a flow representing showering [25]. The results from measurements are presented in Fig. 5 and are denoted “measured”. All the measurements are described in detail in [16].

3.2. Numerical model for IHEUs

A numerical model of the substation was developed in the commercially available software, Simulink [30]. The model is based on work by Persson [31], where it is well described. The philosophy of the modelling is fundamentally based on an energy balance between the primary (hot) and secondary (cold) side including heat transfer through the wall separating the two sides, described together by Equations (1)–(3). This approach accounts some simplifications: no heat conduction among the sections, negligible heat resistance in the HEX walls and no heat losses to the surroundings, but the influence on the accuracy is for our application negligible. Convective heat transfer coefficient $\alpha$ [$\text{W/(m}^2\text{ K)}$] is calculated based on the Equation (4) which is an empirical equation from manufacturer of HEX. Nomenclature for the equations is listed in Table 3.

$$\frac{d}{dt}(m_c \cdot c_p \cdot T_{c, out}) = \frac{m_c \cdot c_p}{A_c} \cdot \left( T_{c, in} - T_{c, out} \right) - \frac{\alpha_{c/w} \cdot A_c}{2} \cdot \left( T_{c, in} - T_{c, out} - T_w \right)$$ (1)

Equation (1) rearranged and written for HEX-section 1

$$\frac{d}{dt}(T_{c, out}) = \frac{1}{m_c} \cdot c_p \cdot \left[ T_{c, in} - T_{c, out} \right] - \frac{\alpha_{c/w} \cdot A_c}{2} \cdot \left( T_{c, in} - T_{c, out} - T_w \right)$$ (5)

Energy balance equation for hot medium

$$\frac{d}{dt}(m_h \cdot c_p \cdot T_{h, out}) = \frac{m_h \cdot c_p}{A_h} \cdot \left( T_{h, in} - T_{h, out} \right) - \frac{\alpha_{h/w} \cdot A_h}{2} \cdot \left( T_{h, in} - T_{h, out} - T_w \right)$$ (2)

Energy balance equation for HEX plates

$$\frac{d}{dt}(m_w \cdot c_p \cdot T_w) = \frac{m_w \cdot c_p}{A_w} \cdot \left( T_{w, in} - T_{w, out} \right) - \frac{\alpha_{w/w} \cdot A_w}{2} \cdot \left( T_{w, in} - T_{w, out} - T_w \right)$$ (3)

Fig. 3. Various IHEU bypass strategies: left — no bypass; middle — external bypass (cold HEX); right — internal bypass (warm HEX).

Fig. 4. Description of IHEUs numerical model with three sections.

Fig. 5. Comparison of experimentally measured and numerically simulated temperatures $T_{22}$ and $T_{12}$ for an IHEU equipped with a PTC2+P controller for the same input data.
Convective heat transfer coefficient $\alpha$

$$\alpha = 0.2 \cdot \lambda_w / (d_p \cdot 2) \cdot Re^{0.67} \cdot Pr^{0.4}$$

(4)

Our model consists of three sections and each section consists of cold and hot side and the wall (see Fig. 4). The three sections model is considered to be enough accurate to model the performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance in the first few seconds. Each of three sections is described by system of Equations (1)–(3) and by their rearranging to form of Equation (5) the system can be solved in Simulink. Additional information about numerical modelling of DH substations can be also found in [32].

As a first step, we verified the HEX model we developed using real measured data from an experiment and the results of design software from the manufacturer [33]. The differences in outlet temperatures for static performance, i.e. when all input values are constant in time, were maximally ±0.2 °C. We considered this sufficient accuracy since the model is aimed to be used as tool for holistic evaluation rather than as an expert tool for the detailed evaluation of substations.

As a second step, we modelled and added the PTC2+P controller and ran tests under dynamic conditions. The numerical model of the controller was built as a physical model of a real PTC2+P controller, i.e. it includes equations describing all springs, $k_t$ values of valves, etc. [29]. The model was verified again with real measured data. Fig. 5 shows a comparison between experimentally measured and numerically simulated output temperatures $T_d$. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31]. The model does not consider heat losses to the ambient, because our aim was to investigate the dynamic performance of the HEX [31].

For the static phase the differences are negligible. The difference is caused by the fact that the numerical model does not include the exact dynamic heating of all components (pipes and valves) in the substation through which the water should pass before reaching the HEX; this causes the deviation between the simulated and the measured values during the transient heating-up period.

The difference in results from simulation and measurement can be also seen in Table 4 by comparing cases M and 0.

### 3.3. Realistic modelling of external bypass

Theoretically, an external bypass maintains a continuous flow through the SP to keep the thermostatic valve at the desired constant set-point temperature. The real solution is slightly different because thermostatic valves have an on/off range (a so-called dead band) to avoid stability problems. The dead band will mean that the bypass flow will not be continuous but intermittent.

An external bypass was modelled with following parameters: set-point temperature 35 °C, dead band ±2.5 °C, and bypass flow rate 3 L/min. The model assumes that the thermostatic valve is controlled by the forward temperature at the end of the SP. In reality, the thermostatic valve is controlled by the temperature measured in the valve itself, which can be strongly affected by the position of the valve in the substation and the insulation of the substation. The results for external bypass performance were obtained by a combination of Matlab code for the SP dynamics and graphs of the cooling of DH water during idling. Both methods, mentioned above in the Introduction, are fully reported by Dalla Rosa [26,28].

<table>
<thead>
<tr>
<th>Case Description</th>
<th>Recovery time of IHEU [s] to produce DHW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without SP</td>
<td></td>
</tr>
<tr>
<td>1 Pure recovery time, $T = 50$ °C, HEX20</td>
<td>3.7, 4.5, 6.5, 7.3, 9.4</td>
</tr>
<tr>
<td>2 Pure recovery time, $T = 50$ °C, HEX40</td>
<td>4.1, 5.5, 7.4, 8.55, 10.9</td>
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<tr>
<td>With SP</td>
<td></td>
</tr>
<tr>
<td>3 Continual bypass 50 °C–35 °C</td>
<td>5.7, 7.7, 10, 11.3, 14.1</td>
</tr>
<tr>
<td>4 Ext. bypass before 2nd bypass flow</td>
<td>5.9, 8.7, 11.0, 12.4, 15.5</td>
</tr>
<tr>
<td>5 Ext. bypass before 3rd bypass flow</td>
<td>7.2, 9.5, 11.9, 13.2, 16.4</td>
</tr>
<tr>
<td>6 Without bypass — 35 °C</td>
<td>7.2, 9.8, 12.3, 13.6, 16.9</td>
</tr>
<tr>
<td>7 Without bypass — 20 °C</td>
<td>11.0, 12.7, 15.2, 16.7, 20.9</td>
</tr>
</tbody>
</table>

### 3.4. Modelling of continual bypass

In addition to a traditional bypass, we also modelled an "ideal external bypass" with continuous flow, i.e. an external bypass without a dead band. It is assumed that a continual bypass flow will be possible used in a floor heating system in bathrooms and thus increasing comfort for users and utilising energy which would otherwise be lost [14]. The results for the temperature profile along the SP were obtained by solving the differential Equation (6) [34] applied for AluFlex 20/20/110 in Matlab [30].

$$T_d = \frac{M}{K} + T_u \cdot \frac{M}{K} \exp \left( -\frac{K}{V} \right)$$

(6)

where $T_d$ is the downstream fluid temperature [K], $T_u$ is the upstream temperature [K], $L$ is the pipe
length [m], \( T_a \) is the ambient temperature [K], \( C_h \) is the overall heat transfer coefficient [W/(m²°C)], \( C_p \) is the specific heat capacity at constant pressure of the fluid [J/(kg°C)], \( A \) is the cross sectional area of pipe [m²], \( V \) is the velocity [m/s], \( \partial P/\partial x \) is the pressure gradient [N/m³]/m, \( D \) is the hydraulic diameter [m], \( f \) is the friction factor [–], and \( \rho \) is the density of fluid [kg/m³].

3.5. Definition of SP operational modes

To investigate the effect of various SP operation modes on the recovery time of an IHEU equipped with a PTC2-P controller, we defined the following cases:

Cases 1 and 2 represent the pure recovery time of the IHEU cooled completely to an ambient temperature of 21.6°C. In these cases, the DH water entering the substation is at a temperature of 50°C already, so there is no SP influence. These cases are not realistic in normal operation (supplied water will never be 50°C from the very first moment), but they are valuable for comparison with other types of substation. The difference between Cases 1 and 2 is that the HEX has 20 plates in the former and 40 plates in the latter.

Case 3 is defined as a continual external bypass with a flow rate of 0.024 L/min, which ensures an almost linear temperature drop in the SP (see Fig. 6 – equation of linear regression) from 50°C at the beginning to 35°C at the end of the SP.

Cases 4 and 5 represent a traditional bypass with a set-point temperature of 35°C with a dead band of 2.5°C. The temperature profile along the SP in Case 4 is the situation just before the 2nd bypass flow and in Case 5 just before the 3rd bypass flow.

Cases 6 and 7 represent the SP without any kind of bypass. In Case 6, the whole SP is at a temperature of 35°C and in Case 7 at a temperature of 20°C. Respectively, these correspond to idling periods of 1 and 4 h for an AluFlex 20/20/110 pipe buried in soil with a temperature of 14°C.

As input data for our simulations, we used temperature profiles along the SP (see Fig. 6) from Matlab simulations as described above. All the simulations take the thermal capacity of the SP into account.

4. Results and discussion

The results presented are valid for summer conditions when there is no need for space heating and with a cold water temperature of 13.2°C. We are aware that during the winter the cold water temperature will be lower, but the additional power needed for increasing the lower water temperature is expected to be covered by increased the supply temperature of DH.

The results of the experimental measurements and numerical simulations for all the cases described are listed in Table 4, which shows the recovery time for an IHEU equipped with a PTC2+P controller to produce DHW with the desired temperature in relation to SP operation mode. For all simulations, except the verification of the model (Cases M and 0, where the primary flow was 14.1 L/min), we used a maximum primary flow rate of 17.3 L/min defined in the IHEU by the differential pressure available (0.5 bar).

As shown above, the numerical model is sufficiently accurate, so it can be used for the simulation of the recovery time in relation to the various SP operation modes.

4.1. Pure recovery time of IHEU

Case 2 represents the pure recovery time of the substation, excluding any SP effect and with an inlet temperature to the substation of 50°C from time \( t = 0 \). It can be seen that the substation needs 7.4 s to produce DHW with a temperature of 40°C and 10.9 s to produce DHW with a temperature of 45°C. In addition to the recovery time, we also need to add the time needed for DHW to reach a fixture and take some temperature drop in the feeding pipe into account. These results are also valid for evaluation of performance of DHSU units, because DH water is supplied from the buffer tank almost immediately, which matches this situation.

IHEUs are normally produced with a HEX with 40 plates, but to reduce the price of the unit, we simulated an IHEU with a 20-plate HEX. Table 4 shows that the recovery time for the 20-plate HEX is
less than for a HEX with 40 plates, e.g. the difference is 1 s for 40 °C. The reason is that the volume of water in the HEX is reduced to one half, so the transportation delay in the HEX is reduced and there is a reduction in the thermal mass of the HEX. Moreover, reducing the number of plates decreases the price of the HEX, because less material is used. On the other hand, the average temperature of primary water returning to the DH network during showering (a DHW flow of 8.4 L/min) is slightly higher, i.e. 21.8 °C instead of 19.1 °C. The pressure drop of the HEX at a nominal power of 32.3 kW and a supply temperature of 50 °C increased from 3.1 kPa on the primary to 2.6 kPa on the secondary side to 13.5 and 8.6 kPa, which are values in the desired range for the optimal operation of the HEX. The low pressure drop for the HEX with 40 plates shows an unnecessary over-dimensioning of the HEX, but ensures better cooling of DH water to 19 °C instead of 22.3 °C. Pressure drop values were calculated by the software provided by the HEX manufacturer [33].

4.2. External and continual bypass

Fig. 6 shows changes in temperature profiles along the SP for cycles of external bypass operation during a summer day (ground temperature 14 °C, no space heating demand in the building). The profiles are a combination of results from SP cooling from COMSOL software (Fig. 1) and calculations of SP dynamics (Fig. 2). At the beginning, the temperature in the whole SP is 50 °C because DHW tapping has just finished. Since there is no tapping during the next 82 min, water standing in the SP cools down to 32.5 °C and this is the opening temperature for external bypass (35 °C-2.5 °C). The bypass flow starts to move cooled water in the SP towards the substation until the temperature at thermostat valve reaches 37.5 °C (i.e. 35 °C + 2.5 °C). At that moment, the temperature profile in the SP is represented by the graph “after first bypass”. This assumption is not fully realistic because the thermostat valve has some thermal capacity which will prolong the bypass flow period, but from an engineering point of view this assumption is enough. When the bypass flow stops, the substation idles again so that water in the SP cools again until the end point of the SP reaches a temperature of 32.5 °C and the bypass flow starts again (the graph “after 2nd cooling”). The last control volume in the SP cools from 37.5 °C to 32.5 °C in 35 min. Fig. 6 shows the temperature profiles until the 4th cooling of the SP. It can be seen that after the 4th cooling the temperature profile is similar to the temperature profile after the 2nd cooling and the temperature profile after the 5th cooling (not shown in the figure) is similar to the temperature profile after the 3rd cooling. The temperature profiles after the 2nd and following bypasses are also very similar, only the inflection point is moved in horizontal direction, to the left (for odd number of bypasses) or to the right (for even number of bypasses). The temperature profiles after the 2nd and 3rd cooling thus represents the most and least favourable situations for the recovery time of the substation. Finally, we have in this way obtained the supply temperatures needed for the simulation of the effect of an external bypass on the recovery time of the substation. All the calculations take the thermal capacity of the SP into account.

In the case of ideal “continuous” bypass, the bypassed water in the SP cools down continuously with an almost linear profile from 50 °C at the beginning of the SP to 35 °C at the entrance to the substation. The flow needed to keep this temperature drop with a ground temperature of 14 °C is only 0.024 L/min.

4.3. The effect of the SP on IHEU recovery time

The recovery time of the IHEU depends strongly on the temperature profile in the SP defined by its operation mode. The longest recovery time is for the case when the SP is operated without a bypass, i.e. the water standing during idling in the SP simply cools down. In this case, the SP water can be in the range between 50 °C and 14 °C (representing temperature of the ground). Case 7 represents such a situation, when idling occurs for 4 h and water in the SP cools down to 20 °C. Case 6 is a similar situation, except that idling occurs for only 1 h and water in the SP cools down to 35 °C. The recovery time of the substation to produce 40 °C DHW, allowing for the effect of the SP operation, is 15.2 and 13.6 s, respectively. These results can be used for customers with very short SPs or for customers with low requirements for comfort. Shorter recovery times are achieved with the traditional external bypass solution, which ensures that the entry-to-substation temperature is always in the range of 32.5–37.5 °C. The recovery time to produce 40 °C DHW is reduced to 11.9 or 11.0 s in Cases 3 and 4 depending on the phase of bypass when tapping was started. The traditional external bypass solution was simulated as an ideal operation without taking into account the thermal inertia of the thermostatic valve and the effect of the ambient conditions. In reality, the temperature sensor measures the water temperature in the thermostatic valve and not at the end point of the SP. It is expected that the real flow in an external bypass will be higher than simulated and will occur for a longer time. Further improvement in the reduction of recovery time can be achieved by the implementation of a continual bypass, which ensures a constant entry-to-substation temperature of 35 °C, i.e. the recovery time to produce 40 °C DHW is reduced to 10 s. The necessary flow of 0.024 L/min can be used for whole-year floor heating in the bathroom and thus ensure an energy-efficient bypass that prevents recirculation of DH water back to the return pipe. In non-circular-shaped DH networks, the bypass should be installed at least at the end of a street, and this solution is more favourable for the DH network and customers than a traditional bypass because it is expected to prevent the need for other types of bypass to keep the DH network at the proper temperature level. The effect of the “comfort bathroom” concept on whole DH network has recently been studied in detail [14].

4.4. Waiting time for DHW

If the requirement is a maximum 10 s waiting time for DHW at the least favourable fixture, i.e. in our case the wash basin (see Table 2), DHW should leave the DH substation with a temperature of 40 °C within 8.4 s of the start of tapping, because it will take 1.6 s to reach the tap. This requirement cannot be met in reality, because if we take the effect of SP operation into account, even in the fastest case (continual external bypass) the recovery time for 40 °C DHW is 10 s. On the other hand, Table 4 shows that DHW with temperature of 30 °C leaves the substation within 5.7 s. DHW with this temperature is not sufficient for taking a comfortable shower, for which a temperature of 37 ± 1 °C is preferred, but should be enough for washing hands. Customers requiring DHW in a very short time whether continuously or discontinuously (only during rush hours) can ensure almost no tap delay by keeping DHW in pipes at the desired temperature using electric trace heating.

5. Conclusion

The paper describes state-of-the-art DH, i.e. LTDH. The first full-scale demonstration site at Lystrup in Denmark proved that the LTDH concept is a promising solution for energy-efficient and secure supply of low-energy buildings with heat harvested from renewable sources of energy. DHW can be produced with a temperature below 50 °C with no increased risk of Legionella because the whole DHW system is designed with an overall volume...
below 3 L and is therefore excluded from traditional DHW temperature requirements.

We have shown that a low-temperature IHEU is able to produce DHW with a temperature of 46 °C from DH water with a temperature of 50 °C with good cooling and with a sufficiently short waiting time. Although the recovery time for 45 °C DHW is longer than the suggested 10 s, the substation produces 35 °C DHW in 5.5 s which can be regarded as warm enough to start washing hands or showering. Nevertheless, the recovery time could be further decreased by optimizing the design of the HEX with an adequate number of plates, pressure drop, and improved heat convection transfer, all resulting in reduced thermal mass and therefore in faster recovery time. Work on such a design is in progress.

The numerical model of an IHEU we developed was successfully validated with real measurements and in combination with a model of an SP was used for the evaluation of various SP operation modes in relation to DHW comfort for customers and the overall economy of LTDH.

The results show that IHEUs installed in a DH network should be equipped with a bypass to keep the recovery time within a range acceptable for users. Nevertheless, the use of a traditional bypass with a set-point temperature of 35 °C cannot guarantee a short waiting time. The most efficient solution might be a continual bypass always ensuring an entry-to-substation temperature of 35 °C during idling. The continual bypass flow with a constant temperature can be used in floor-heating systems in bathrooms and thus bring benefits for the customer as well as for the network.

The numerical model of the IHEU is an important tool for the further optimization of the whole LTDH concept and its individual components. Moreover, it can be extended with a storage tank module and then be used for the optimization of DHSUs.

The next step in our research will be the consideration of real conditions in DH networks, i.e. dropping the assumption of 50 °C DH water at the beginning of the SP and using real temperature profiles, which will be result of the simulation of a real DH network. Other research should also focus on the introduction of the LTDH concept in existing single-family and multi-storey buildings, because these buildings represent the majority of the building stock.

Heat Plan Denmark 2008 concluded that DH is the main solution to achieve fossil-free heat supply in Denmark by 2050 with regard to energy efficiency, economy and environment. But to make this happen, DH must be transformed into LTDH. More research and improvements are still needed to achieve the strong position of LTDH predicted for the future, but we now have the first show-case example of Low-Temperature District Heating and the main task has become its implementation on the large scale.

References

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ISI paper II

Energy-efficient and cost-effective in-house substations bypass for improving thermal and DHW comfort in bathrooms in low-energy buildings supplied by low-temperature district heating

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ENERGY-EFFICIENT AND COST-EFFECTIVE IN-HOUSE SUBSTATIONS BYPASS FOR IMPROVING THERMAL AND DHW COMFORT IN BATHROOMS IN LOW-ENERGY BUILDINGS SUPPLIED BY LOW-TEMPERATURE DISTRICT HEATING

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ABSTRACT

Using a bypass to redirect a small flow through the in-house district heating (DH) substation directly to the return pipe is a commonly used but energy-inefficient solution to keep the DH network “warm” during non-heating seasons. Instead, this water can be redirected to the bathroom floor heating to cool down further and thus reduce the heat lost from bypass operation while tempering the bathroom floor heating as well as guaranteeing fast provision of DHW. We used the commercial software IDA-ICE to model a reference building where we implemented various solutions for controlling the redirected bypass flow and evaluated their performance. The effect on the DH network was investigated using Termis software. Bypass flow redirected into bathroom floor heating during the non-heating period resulted in a comfortable increase in floor temperature without causing overheating, additional average cooling of bypass water by 3.9°C, and reduced heat loss from the DH network by 13% in comparison with the reference case. The use of the bypass flow in bathroom floor heating is a cost-effective solution, both for DH utilities (reduced heat loss from the DH network and higher revenues), their customers (improved thermal comfort), and society (reduction of greenhouse gas emissions).

Keywords: efficient external bypass, low-temperature district heating, floor heating, DHW waiting time, network heat loss reduction, IDA-ICE

LIST OF ABBREVIATIONS AND SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>( \dot{m} )</td>
<td>mass flow</td>
</tr>
<tr>
<td>( T_{\text{bypass}} )</td>
<td>bypass set-point temperature</td>
</tr>
<tr>
<td>( T_{\text{floor}} )</td>
<td>floor temperature</td>
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<tr>
<td>( T_{\text{op}} )</td>
<td>operative temperature</td>
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<tr>
<td>( T_{\text{ret}} )</td>
<td>return temperature</td>
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<tr>
<td>( T_{\text{soil}} )</td>
<td>temperature of soil</td>
</tr>
<tr>
<td>( \Delta T_{\text{DB}} )</td>
<td>deadband</td>
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<tr>
<td>CB</td>
<td>comfort bathroom</td>
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<td>DH</td>
<td>district heating</td>
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<td>DHW</td>
<td>domestic hot water</td>
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<td>FH</td>
<td>floor heating</td>
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<td>FJVR</td>
<td>TRV controlled by the fluid temperature</td>
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<td>HEX</td>
<td>heat exchanger</td>
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<td>IHE substation</td>
<td>instantaneous heat exchanger substation</td>
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<td>SH</td>
<td>space heating</td>
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<td>SUB</td>
<td>DH in-house substation</td>
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<td>TRV</td>
<td>thermostatic regulation valve</td>
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1 INTRODUCTION

The heat demand in buildings drops outside the heating season because the only heating requirement users usually have is in connection with Domestic Hot Water (DHW). This is very discontinuous and is generally needed for a total of less than 1 h/day in a typical single-family home [1]. The lack of heat load causes the undesirable cooling of some parts of the network to temperatures that are insufficient to ensure the prompt provision of DHW through the Instantaneous Heat Exchanger (IHE) substations. If proper control strategies are not implemented, the time needed for low-temperature IHE substations to produce DHW with a temperature of 40°C can increase by 40% up to 15 s [2]. That is why bypass valves are installed at in-house substations (see Figure 1) and/or other suitable locations in the network: their purpose is to redirect a relatively low water flow from the supply media pipe to the return media pipe, so that the temperature at the District Heating (DH) substation inlet is maintained within the required range of operation. The effect of the bypass is a certain flow achieved during low heat load periods, ensuring sufficient supply temperatures in the network. For example, in the low-temperature DH pilot project in Lystrup [3], the set-point temperature for the external bypass is 35°C.

![Figure 1: Principle of bypass valve installed at in-house DH substation.](image)

This operation, although necessary, results in increased heat loss and higher return temperatures in the DH network, and this is particularly critical in the case of low-energy demand building areas. The share of heat losses due to the bypass operation can reach up to 55% of the heat demand (i.e. for heating DHW) of low-energy buildings outside the heating season. Moreover, in low-energy buildings, the heating season is shorter than in traditional buildings, so this means the bypass is needed for a longer period. Furthermore, even during the heating period there are sunny days when space-heating is not required (for our reference case, as much as 6%). All this underlines the need for an energy-efficient bypass solution. In recent years, R&D and demonstration studies on low-temperature DH systems for energy-efficient buildings have shown that the concept fits the vision of a sustainable building sector in Denmark, by integrating end-user energy savings, energy-efficient distribution networks, and low-grade sources and renewable energy on the supply side [1]-[9]. This paper deals with the modelling of bypass operation strategies in DH service pipes supplying low-energy buildings and the option of using the redirected water in bathroom Floor Heating (FH) to increase thermal comfort for customers. The effect of the bypass operation at the level of the DH distribution network is also reported.

2 TRADITIONAL EXTERNAL BYPASS IN AN IN-HOUSE SUBSTATION

The traditional bypass solution is widely used, but there is still a lack of precise knowledge on bypass performance, i.e. how much energy/water is sent back to the DH network, how often the bypass opens, and for how long. To evaluate the performance of the traditional external bypass, we modelled and measured a traditional external bypass embedded in an instantaneous low-temperature DH substation.

2.1 MODELLING OF BYPASS FLOW IN SERVICE PIPES

The bypass is generally controlled by a thermostat with an adjusted set-point temperature, $T_{\text{bypass, set}}$, and the amplitude of the “deadband”, $\Delta T_{\text{DB}}$, which also defines a “top temperature”, $T_{\text{bypass, top}} = T_{\text{bypass, set}} + \Delta T_{\text{DB}}/2$, and a bottom temperature, $T_{\text{bypass, bottom}} = T_{\text{bypass, set}} - \Delta T_{\text{DB}}/2$. The bypass control ensures that the temperature is kept within the range of operation set by the deadband.
Our investigations compared the energy performance of two theoretical bypass operations. The first case is the situation with an “ideal” thermostatic valve without a deadband (ΔT_{red}=0) where there is a continuous bypass flow through the service pipe to maintain \( T_{bypass, set} \) at the service pipe outlet (the inlet to the in-house substation), where the bypass control is assumed to be located. The second case is a “pulse” bypass operation (caused by the deadband of the self-acting controller) modelled as controller that is acting like an on/off switch. This means that when the temperature of bypass water at the outlet of the service pipe reaches a specific value, \( T_{bypass, top} \), the bypass flow instantaneously stops; the pipe is now in “stand-by” mode, meaning that there is no flow in the media pipe, and the water gradually cools down; after a certain time, the temperature at the service pipe outlet has decreased to the value of \( T_{bypass, bottom} \), the bypass valve opens, and the water flows again. A pulse bypass cycle consists of a period of water flow from the main distribution line to the service pipe (called the “bypass” period), and a period when there is no flow and the water inside the supply service pipe cools down (the “stand-by” period); after that, another cycle starts, in a process that can be modelled as periodical. The Matlab® code modelling the DH service pipes developed in [7], [8] was applied to study the transient, coupled fluid-thermal phenomena during the bypass mode; the cooling of the water during the stand-by period was evaluated using regression curves derived from 2-D transient heat transfer simulations in COMSOL Multiphysics®, see Figure 2. Given the geometry and the materials of the service pipe, a specific supply water temperature, and certain boundary conditions, it is possible to calculate the transient temperature field in the pipe, see Figure 3. The model of the DH pipes in the heat transfer simulations was built in accordance with the methodology explained in [9].

We modelled both ideal and pulse bypass operation with code developed in [8] at three locations in the DH network (see Figure 10). The numerical code used was successfully verified for dynamic calculation in the service pipe, but first without the effect of the thermostatic bypass valve because it would be very difficult to make this test in laboratory conditions. We subsequently carried out full-scale measurements at a low-temperature DH substation connected to the DH network at the Danish Technological Institute, but even using DH energy meters with the highest available flow resolution (signalling every measured litre), the highly dynamic performance of the pulse bypass involves volumes below this resolution, so the results were not used.
The DH supply temperature expected at the beginning of the service pipe in the non-heating season during periods without DHW tapping will differ in each location. The further down the DH network the house is situated, the lower the temperature at the inlet to service pipe, in this case: 50°C, 40°C and 37.3°C. The service pipe in the investigation was always considered as 10 m long twin pipe Aluflex 20/20/110 (i.e. media pipe inner Ø = 16 mm, casing pipe outside Ø = 110 mm) [10], surrounded by soil with temperature 8°C. This is a typical connection between the IHE substation and the single-family house.

The continuous flow needed in the case of ideal bypass operation to keep the end of the service pipe at 35°C for each location depends on the temperature at the beginning of the service pipe: if this temperature was 50°C, 40°C or 37.3°C, it was calculated to 1.77 kg/h, 4.68 kg/h and 9.36 kg/h, respectively. Pulse bypass operation was modelled with a set-point temperature of 35°C, a deadband of 3°C and a bypass flow of 0.5 kg/min only for locations with a continuous flow of 1.77 kg/h and 4.68 kg/h. The results show that the intermittent bypass in both cases operates approximately every 15 minutes. For the first location, the bypass opens approximately 70 s in each cycle and bypasses 0.65 kg, which corresponds to 2.6 kg/h. For the second location, the bypass opens for 213 s and bypasses roughly 1.8 kg, i.e. 7.1 kg/h.

Comparing the results, it can be concluded that keeping the inlet of the substation at 35°C, the pulse bypass operation circulates approximately 50% more water than the continuous bypass. The higher bypass flow with pulse operation results in 10-35% higher heat loss from service pipes than with the continual flow. Based on this, it would be beneficial to replace the traditional thermostatic bypass using pulse operation with a solution that keeps the flow constant. This could be realised by using a small needle valve. However, this does not take DHW tapping into account, as explained later in Section 3.2.2.

To improve the energy efficiency of the DH network, it might be interesting to use the energy delivered by the bypass e.g. for FH in rooms where it is desirable to have a warm floor even outside the normal heating season, as can be the case with bathrooms in buildings situated in Scandinavian climate regions. This option is described and investigated in the following paragraphs as the “comfort bathroom” concept.

### 3 THE “COMFORT BATHROOM” CONCEPT

The Comfort Bathroom (CB) is a concept of redirecting insufficiently cooled DH water from the external bypass of an in-house substation to the bathroom FH during a non-heating period. The outcome is additional cooling of the bypassed water, resulting in a reduction of heat loss from the DH network, higher efficiency of DH heat sources, and the sensation of a warm floor for occupants, increasing their comfort. In the case of our reference house (described later in Section 3.2.2), the average floor temperature in the bathroom during non-heating period is only 22.4°C. Based on [11], the floor temperature preferred in bathrooms is 28±0.3°C, and this justifies the option of increasing floor temperature comfort in the bathroom.

It is true that a similar effect can be achieved by an FH loop controlled with a traditional TRV [12] (thermostatic valve controlled by operative temperature, e.g. for controlling radiators) modified for use in FH loop or an FJVR valve [13] (thermostatic valve controlled by temperature of fluid), but in comparison CB utilizes only the “free” energy available from DH water bypassed anyway through the external bypass. The CB concept is therefore not meant to give the same level of comfort as FH controlled by TRV or FJVR, but to utilize part of the heat lost by bypass operation in the DH network to heat up the floor in bathrooms at discount price.

#### 3.1 TECHNICAL SOLUTIONS FOR CB

FH in the bathroom can be controlled either by an FJVR valve, or by a TRV valve controlled traditionally by self-acting thermostatic sensor (e.g. [13]) or by an electronic actuator (e.g. [14]) with wired/wireless remote temperature sensor (not shown in Figure 4). The first two solutions is usual in houses heated by radiators with FH installed only in the bathroom, while for buildings with all rooms heated by FH, every FH loop usually has own control valve with remote temperature sensor.

Figure 4 shows the implementation of the CB concept with various types of space heating (SH). For this purpose, SH systems can be divided into those with the option of changing the supply temperature to the FH loop (directly connected SH with mixing loop or indirect SH systems) and those without (directly connected SH without mixing loop).
Reference case without CB

Figure 4a shows the original connection scheme of the SH system used in the low-temperature DH showcase in Lystrup, where we plan to test the CB concept in full scale as part of the EUDP 2010 research project [3]. One goal of the designer was obviously to make a simple and cheap solution, with directly connected radiators, FH in the bathroom and no mixing loop (i.e. the SH system is supplied with 50°C also during moderate outdoor temperatures). The FH is controlled by an FJVR valve [13] mounted on the return pipe from the FH loop. The FJVR valve is situated in the technical room, just before the manifold collecting individual SH loops. The substation is kept “ready for use” by the thermostatic external bypass (also an FJVR valve, with a set-point of 35°C) redirecting bypassed water back to the DH return. This SH design is simple and widespread, so there is high potential for using the CB concept in many existing buildings that have these systems.

3.1.1 SH systems without mixing loop

**CB realised with a needle valve**

The solution with a needle valve was investigated on the basis of results of previous work [15] reporting lower heat loss from the service pipes using continuous bypass flow than using an FJVR valve with a deadband. CB is realised by installing a needle valve on the return pipe of FH in parallel with the existing FJVR valve (see Figure 4b) keeping the constant bypass flow through the bathroom FH. The external bypass valve located in the substation is removed. The needle valve is placed in the loop of the differential pressure controller, so the flow remains constant also during oscillations of differential pressure in the DH network. For periods when the bypass flow through the needle valve is not enough to heat the bathroom, the additional flow can pass through the originally installed FJVR. The advantage of this solution is the simple and cheap installation of the needle valve in the substation, (i.e. no need to take any action in the FH loop). The drawback of the solution is the lack of an “automatic stop” of the bypass flow when the supply temperature is above set-point temperature, i.e. 35°C in our case. This means that the CB will be supplied with water with a temperature above 35°C for about 45 minutes after DHW tapping, resulting in higher energy use possibly leading to overheating and also higher return temperature, which is undesirable. Similar problems also arise in cases with the FH loop controlled with a traditional TRV valve, where the implementation of CB is by installing the needle valve in the bathroom parallel with the TRV valve.

**CB realised with an FJVR**

In FH loops controlled by TRV valves, it is possible to prevent the supply of bypass water with a temperature above the bypass set-point by replacing the needle valve with an FJVR valve as an external bypass in parallel with the TRV...
ventilation system is designed to supply 216 m$^3$/h and the heat recovery has 85% efficiency, accounting for a drop in and the weather file Design Reference Year (DRY) for Denmark was applied. In accordance with BR10, the concept was implemented in both of them. The software used for the simulations was advanced level of IDA-ICE 4.2

number of hours with an operative temperature above 26°C to less than 100 hours a year.

temperature in the house in all rooms is always below 26°C, so the house fulfils BR10 [16] requirements limiting the

occupants are not at home, i.e. working days between 8 am and 3 pm. The internal heat gains in the whole house above 24°C and stops when the air temperature drops below 22°C. The windows cannot be opened when the

by a 0.5 m roof-overhang. Venting by opening of the windows starts when the air temperature in the room rises above 24°C and stops when the air temperature drops below 22°C. The windows cannot be opened when the occupants are not at home, i.e. working days between 8 am and 3 pm. The internal heat gains in the whole house were modelled as 5 W/m$^2$ constantly, but the heat gains from both bathrooms were transferred to the living room and kitchen, because based on [19] the internal heat gains produced in the bathrooms are negligible. The operative temperature in the house in all rooms is always below 26°C, so the house fulfils BR10 [16] requirements limiting the number of hours with an operative temperature above 26°C to less than 100 hours a year.

3.2 MODELLING OF CB

3.2.1 Reference house

The CB was modelled in a 157 m$^2$ single-family house fulfilling the requirements of low energy class 2015 in accordance with the Danish Building Regulation 2010 (BR10) [16]. This means that the energy needed for SH, DHW heating and the operation of HVAC systems should be below 37 kWh/(m$^2$.y) after accounting for primary energy factors. The house is described in more detail in [17]. The house has two bathrooms (8.3 and 4.3 m$^2$) and the CB concept was implemented in both of them. The software used for the simulations was advanced level of IDA-ICE 4.2 [18] and the weather file Design Reference Year (DRY) for Denmark was applied. In accordance with BR10, the ventilation system is designed to supply 216 m$^3$/h and the heat recovery has 85% efficiency, accounting for a drop in efficiency for very cold outside temperatures. The windows in the house are shaded with external blinds (g value of 0.14) drawn when the solar irradiation on the window increases above 300 W/m$^2$. Moreover, the windows are shaded by a 0.5 m roof-overhang. Venting by opening of the windows starts when the air temperature in the room rises above 24°C and stops when the air temperature drops below 22°C. The windows cannot be opened when the occupants are not at home, i.e. working days between 8 am and 3 pm. The internal heat gains in the whole house were modelled as 5 W/m$^2$ constantly, but the heat gains from both bathrooms were transferred to the living room and kitchen, because based on [19] the internal heat gains produced in the bathrooms are negligible. The operative temperature in the house in all rooms is always below 26°C, so the house fulfils BR10 [16] requirements limiting the number of hours with an operative temperature above 26°C to less than 100 hours a year.
3.2.2 Difference in continuous and intermittent bypass

As mentioned above, from the on-site measurements we could not establish whether the FJVR valve operates in continuous or pulse mode because of the low resolution of flow meters. So we modelled both pulse and continuous operation of the bypass and investigated its importance for the performance of FH. We did this at two different locations in the DH network defined with continuous flow 1.77 and 4.68 kg/h as described in Section 2.1.

CB supplied with a continuous flow of 1.77 kg/h was modelled first with the needle valve, which is unable to stop the bypass flow when the temperature is increased by DHW tapping, and then with an FJVR valve blocking the bypass flow during and for 45 minutes after every DHW tapping. The FJVR valve was modelled in three modifications. First the FJVR valve was considered as an ideal controller with no deadband, providing continuous bypass flow in the same magnitude as the needle valve. Second, we considered a realistic use of an FJVR with a 3°C deadband, resulting in pulse operation and an increase of bypassed volume of roughly 50%, i.e. 2.6 kg/h. Finally, we also considered pulse operation of the valve but without the 50% increase of bypassed volume.

The second location in the network was modelled in the same way, but with the basic continuous flow of 4.68 kg/h, i.e. 7.1 kg/h when considering the 50% increase of bypassed volume.

Figure 5 compares the floor temperature $T_{\text{floor}}$ and the temperature of water returning from CB $T_{\text{ret}}$ for the CB controlled by the needle valve and the FJVR valve in the two locations in the DH network. For the sake of comparison, the floor temperature for the bigger bathroom, but without CB concept, is also shown. The values in Figure 5 are reported in time steps of 3.6 s. The cases can be divided into three groups. First, the reference case without CB, second the case defined as continuous flow of 1.77 kg/h, and third the case with continuous flow of 4.68 kg/h.

![Figure 5: Comparison of: a) $T_{\text{floor}}$ and b) $T_{\text{ret}}$ in CB for two locations in the DH network modelled with four different control strategies of bypass flow, output for results in steps of 3.6 s.](image)

It can be seen that for the case of 1.77 kg/h the maximal difference in $T_{\text{floor}}$ (see Figure 5a) for all four control strategies (one with a needle valve and three with an FJVR valve) is 0.25°C. The difference increases maximally to 0.5°C for the case of 4.68 kg/h. The maximum differences in $T_{\text{ret}}$ are 0.3°C and 0.7°C, respectively. Moreover, the curve of $T_{\text{ret}}$ also documents differences between individual flow controls. While the $T_{\text{ret}}$ for CB realised with needle valve increases slightly after each DHW tapping (see Figure 5b, time h=4488), the $T_{\text{ret}}$ for all FJVR valves decreases because the flow to the bathroom is stopped during and for 45 minutes after DHW tapping. Pulse operation of the FJVR valve caused by the deadband can be seen as an increase of $T_{\text{ret}}$ every 15 minutes however the average value is comparable with the performance of the ideal FJVR valve modelled without the deadband.

Based on the Figure 5, it can be concluded that the difference in performance between the FJVR valve modelled with continuous flow or pulse flow is negligible and the total mass flow and the ability to block bypass water with a temperature above 35°C is more important. Pulse operation of the FJVR valve makes no noticeable changes in $T_{\text{floor}}$, because the heat delivered in pulses is averaged by the thermal mass of FH. The difference in weighted averaged $T_{\text{ret}}$ is also negligible.

This finding allows us to model the pulse FJVR as continuous. This saves computational time because the simulation of pulse bypass needs time steps small enough to catch the nature of intermittent bypass (around 30 seconds) while in some situations time steps for other heat transfer phenomena in buildings can be increased to steps ranged in
hours. Moreover, output data reported every 30 seconds makes it very difficult to analyse results because of huge amount of data.

3.2.3 Specification of modelled cases

The usual set-point temperature of external bypass valves installed in low-temperature DH substations with the instantaneous principle of DHW preparation is 35°C [5]. Nevertheless, the assumption of CB supplied during the non-heating period constantly with bypass flow of 35°C is a considerable simplification and does not reflect the influence of DHW tapping. During each DHW tapping, the temperature of DH water at the inlet of the substation increases to 50°C (the supply media pipe is filled with hot water at 50°C). It takes approx. 45 minutes for water standing in the service pipe to cool back down to 35°C [2]. Moreover, during the heating season, the DH substation is supplied with DH water with temperatures of 50°C or higher (peaking in very cold winters) continuously. So the DH supply temperature to the substation can be considered to be 50°C during a heating period and for the rest of the year (the non-heating period) to reflect the effect of DHW tapping. We assume a 5-minute period of DHW tapping every 3 hours (daily between 6:00-24:00), which means that the temperature in the service pipe is 50°C for 5 minutes during DHW tapping and then drops linearly for 45 minutes to 35°C. This means that while in the case of CB realised with an FJVR valve, the valve stops the supply of FH when the temperature of the bypass water is above 35°C, in the case of CB realised with a needle valve and in the reference case with an FH loop traditionally controlled by an FJVR, the FH loop is supplied by bypass water linearly decreasing from 50°C and 35°C for a period of 45 minutes.

As described in Section 2.1, we investigated three different locations in the DH network, resulting in three different bypass flows. The cases investigated are listed in Table 1 and described below in detail.

Table 1: Matrix of simulated cases.

<table>
<thead>
<tr>
<th>case abbreviation</th>
<th>flow [kg/h]</th>
<th>Tsup to CB &gt;35°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 noCB - FJVR</td>
<td>yes</td>
<td>✓ ✓ x</td>
</tr>
<tr>
<td>2 noCB - external BYP</td>
<td>no</td>
<td>✓ ✓ ✓</td>
</tr>
<tr>
<td>3 CB - needle valve</td>
<td>yes</td>
<td>✓ ✓ ✓</td>
</tr>
<tr>
<td>4 CB - FJVR</td>
<td>no</td>
<td>2.6 7.1 14.0</td>
</tr>
<tr>
<td>5 CB - electr. step valve</td>
<td>no</td>
<td>✓ ✓ ✓</td>
</tr>
<tr>
<td>6 CB - FJVR + mixing loop</td>
<td>no</td>
<td>x 7.1 x</td>
</tr>
</tbody>
</table>

**Case 1 – FH whole year**

Case 1 is a reference case without implementation of CB, but the FH is on duty the whole year. The DH substation is equipped with a traditional external bypass valve. FH loops in both bathrooms are equipped with an FJVR valve at the return pipe (one valve for each), controlling the mass flow to keep the return water from each loop at the set-point temperature of 25°C. If the total flow rate is not enough to keep the inlet of substation at 35°C, the traditional external bypass opens. On the other hand, the flow needed in both bathrooms during the non-heating period can be higher than the minimal bypass flow, resulting in temperatures above 35°C at the inlet of the substation. The temperature of the DH water supplied by the service pipe during the non-heating period in this case changes as a function of the flow rate. The relationship between flow and temperature supplied by a service pipe has been calculated by [2]. Moreover, by logging the flow through the external bypass, it can be seen whether the bypass flow needed in both FH loops controlled by FJVR valves is high enough to be used instead of the external bypass.

**Case 2 – FH only during the heating period**

This example represents a second reference case, i.e. a substation equipped with the external bypass and no heating demand in the bathrooms during the non-heating period, i.e. FH in bathroom is controlled by an FJVR or TRV valve, but the valves are closed during the non-heating period.

**Case 3 – CB realised with a needle valve**

This case represents CB realised with a needle valve installed in parallel with a TRV or FJVR valve. The flow rate through the needle valve for three different locations in the DH network has been calculated by [8], taking into account boundary conditions for the non-heating period. Flow through the needle valve is stopped during a heating period. The case does not consider changes in the temperature of bypassed water caused by changes in ground temperature during a non-heating period.

**Case 4 – CB realised with an FJVR with a deadband**

This case represents an improved solution realising the CB concept by replacing the needle valve with an FJVR valve installed in parallel with the TRV valve. The difference is in the automatic shut off of the “bypass” flow in the
periods when the bypassed water has a temperature above the bypass set-point. This means that the CB is turned off during a space heating season and also for 5 plus 45 minutes (needed for DH water in service pipe to cool down from 50°C to 35°C) after every DHW tapping during the non-heating season. In this case, the FJVR valve is modelled as continuous flow, controlled by a deadband of 3°C, expecting a 50% increase in the volume of bypassed water (Section 2.1) compared to the needle valve solution.

**Case 5 – CB realised with an electronic step valve**

This case is modelled in the same way as the previous one, but with an FJVR valve without a deadband. This means that the bypass flow is the same as in the case of a needle valve, but the bypass flow is stopped if its temperature exceeds the set-point temperature of 35°C. The FJVR valve without a deadband (in fact, there is always a deadband, but it can be reduced) can be realised using an electronic step valve controlled by a temperature sensor. However, the effect of reduced deadband on the control stability of the valve should be considered.

**Case 6 – CB with mixing loop (direct SH) or heat exchanger (indirect SH)**

This case is an implementation of CB into the indirect SH system or direct SH system with a mixing loop. Considering the thermal efficiency of the HEX for SH equal to 1, both systems can be represented by one model. The flow rate through the CB FH loop is distributed proportionally to the floor area, i.e. 80 and 40 kg/h for the bigger and smaller bathroom, respectively. The temperature of the water supplied to the SH system during a heating season is controlled by a P-controller measuring the operative temperature in the bathroom and adjusting the supply temperature proportionally in a range from 21 to 29°C. The supply temperature to the CB loop during a non-heating period is a result of mixing the bypass flow required by the location of the building in the DH network with the flow returning from the CB loop in a ratio such that the total flow through the CB loop (both bathrooms) is 120 kg/h. The mixing loop is not provided as default in IDA-ICE and should be modelled by the user.

4 RESULTS AND DISCUSSION

4.1 PERFORMANCE OF FLOOR HEATING WITH/WITHOUT COMFORT BATHROOM

The figures below show the operative temperature $T_{\text{op}}$ and floor temperature $T_{\text{floor}}$ reported with time steps of 0.3 h and, if not specified, only for the bigger bathroom.

**4.1.1 HEATING PERIOD**

Figure 6 compares the performance of FH in the bigger bathroom during a heating period. This is directly connected FH without a mixing loop controlled by a TRV valve with a set-point $T_{\text{op}} = 20°C$, an FJVR valve with a set-point temperature of 25°C (sensing temperature of water returning from FH loop), and FH with a mixing loop equipped with TRV. It can be seen that for the FH loop controlled traditionally by an FJVR valve, $T_{\text{op}}$ is constantly around 21.5°C and $T_{\text{floor}}$ around 23.5°C, while for the TRV valve the values are 20.5°C and 21.5°C, respectively. The average $T_{\text{floor}}$ is slightly higher for the case with the mixing loop than for the directly connected solution because a lower supply temperature and higher flow rate in the FH loop results in a better temperature distribution, but on the other hand in worse cooling and therefore also in slightly higher $T_{\text{ret}}$.

![Figure 6: Performance of FH in the bathroom during heating season.](image-url)
The heat demand for both bathrooms is 90% higher for the reference case with FJVR during heating season than for the case with TRV (see Table 2), as a result of $T_{op}$ being higher by 1.1°C on average. But most of the heat is recovered by the ventilation system, so the overall additional heat demand of the house during the heating period is 93 kWh, i.e. 4%. The heat demand during the non-heating period increases by 17%, which can be expressed as an additional cost of 322 DKK for the customer. We are assuming a heat price of 650 DKK/MWh (currency exchange rate 1 DKK = 0.13 EUR).

4.1.2 Floor heating without CB

Figure 7 shows a two-day period in two locations in the DH network during a non-heating season and compares $T_{floor}$ and $T_{op}$ for the reference case without FH heating (Case 2), the case with FH traditionally controlled by an FJVR (Case 1), and for the sake of comparison also with the CB concept realised with an FJVR valve (Case 4). It can be seen that $T_{op}$ and $T_{floor}$ are lowest for the reference case without FH, with very small differences between both temperatures. On the other hand, FH controlled with an FJVR valve in the traditional manner results in the highest $T_{op}$ and $T_{floor}$. As can be seen, curves for both cases with a traditionally operated FJVR valve are very similar, confirming that in this case the amount of heat delivered by the FH does not depend on its location in the DH network. Furthermore, the set-point temperature of 25°C on the return pipe results in $T_{op}$ over 24°C for some time periods and it triggers opening of a window. The window is closed again when $T_{op}$ drops to 22°C, but the window opens only when occupants are at home. The window is opened for 415 hours, i.e. 8% of the non-heating period, despite the building’s location in the DH network. The flow of DH water from the service pipe for the location defined as continuous flow of 1.77 kg/h is below the minimal value for 3% of the non-heating period, and for the location defined as having a continuous bypass flow of 4.68 kg/h is below the minimal value for 17% of non-heating period, which means that FH controlled by an FJVR with a set-point temperature of 25°C cannot always ensure the desired bypass flow for the whole non-heating period, so the traditional bypass valve should be used anyway.

$T_{op}$ and $T_{floor}$ for both CB cases lie between the reference cases and depend on the water flow rate defined by the position of the building in the DH network. Average $T_{op}$ and $T_{floor}$ for the reference case without FH during the non-heating period are both 22.4°C, meaning that the CB concept can be used. Results for all cases are summarised in Table 2.

![Figure 7: $T_{op}$ and $T_{floor}$ in bathroom during non-heating period, considering various control of floor heating system.](image)

4.1.3 Floor heating with CB

Figure 8 shows $T_{floor}$ for a two-day period during a non-heating season for the CB solution realised with a needle valve and an FJVR valve with a 3°C deadband and compares them with both reference cases without CB. Based on the finding in Section 3.2.2, we modelled the bypass flow controlled by the FJVR valve only as continuous flow.

It can be seen that $T_{floor}$ and also $T_{op}$ (not shown in Figure 8) depends not only on the bypass control strategy, but also on the bypass flow rate defined by the location in DH network. If we compare the same bypass strategy for different locations in the DH network, it can be seen that the further the end-user is located from the heat production plant, the more heat is transferred to the bathroom FH expressed as higher $T_{floor}$. For both CB strategies, $T_{floor}$ in the
location defined with continuous flow 4.68 kg/h is roughly 0.75°C higher, and in the location defined as 9.36 kg/h 1.5°C higher than in the location with a continuous bypass flow of 1.77 kg/h.

If we compare $T_{\text{floor}}$ for CB realised with a needle valve and an FJVR with a deadband in the same locations in DH network, $T_{\text{floor}}$ is very similar, even though the nominal bypassed volume was 50% higher in the case of an FJVR valve. The explanation lies in the shut off of the bypass flow with the FJVR valve after each DHW tapping, which results in a final difference in bypassed volume of around 6% (see Table 2). Similar can be concluded also for CB realised in the system with a mixing loop.

![Figure 8: $T_{\text{floor}}$ during a non-heating period in a bathroom with CB realised with a needle valve or an FJVR valve.](image)

Figure 9 also shows $T_{\text{floor}}$ and $T_{\text{op}}$ for CB located in three different places in DH network, but compares CB realised with a needle valve with an electronic step valve, i.e. an ideal FJVR valve with a reduced deadband resulting for the same locations in a bypass flow rate equal to that for the needle valve. If the bypass flow rate in the three locations investigated is reduced by 50% while retaining the blocking function when the bypass water is above 35°C, $T_{\text{floor}}$ is reduced by 0.1°C, 0.25°C and 0.25°C in comparison to CB realised with a needle valve (or an FJVR valve with a deadband as shown in Figure 8).

![Figure 9: $T_{\text{floor}}$ during a non-heating period in a bathroom with CB realised with a needle valve or electronic step valve.](image)

**4.1.4 COMPARISON OF ALL CASES INVESTIGATED**

Table 2 summarizes the performance of all the cases investigated for the non-heating period, i.e. from 15 April to 15 November in the three different locations in DH network. The locations are defined as locations where the DH water at the beginning of the service pipe has a temperature of 50°C, 40°C and 37.3°C, respectively. The average return
temperature reflects only the performance of the external bypass/CB and does not take account of the effect of DHW tapping. Taking the DHW production corresponding to 3200 kWh/person into account with design cooling the DH water from 50°C to 20°C will decrease the average return temperature, but from the perspective of heat loss reduction from the DH network this effect can be neglected because DHW tapping accounts only for 3% of the day (40 minutes).

Table 2: Comparison of simulated cases for non-heating period 15/4 – 15/11, i.e. 5160 hours.

<table>
<thead>
<tr>
<th>case #</th>
<th>nominal bypass flow [kg/h]</th>
<th>T_{up} avg. [°C]</th>
<th>T_{down} avg. [°C]</th>
<th>T_{ret} avg. [°C]</th>
<th>bypassed volume [m³]</th>
<th>average heat output from FH [W]</th>
<th>energy delivered by SP [kWh]</th>
<th>energy used in FH [kWh]</th>
<th>heat demand incl. FH [kWh]</th>
<th>increase of heat demand [%]</th>
<th>CB cost for customer [DKK]</th>
<th>bypass cost for DH company [DKK]</th>
</tr>
</thead>
<tbody>
<tr>
<td>location 1</td>
<td>1</td>
<td>1.77</td>
<td>23.5</td>
<td>24.4</td>
<td>25.2</td>
<td>24.3</td>
<td>97</td>
<td>1212</td>
<td>500</td>
<td>2945</td>
<td>17%</td>
<td>325</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>2.6</td>
<td>22.4</td>
<td>22.4</td>
<td>35.0</td>
<td>9.7</td>
<td>0</td>
<td>435</td>
<td>0</td>
<td>2352</td>
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<td>0</td>
</tr>
<tr>
<td></td>
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<td>23.0</td>
<td>23.0</td>
<td>23.3</td>
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<td>32</td>
<td>430</td>
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<td>2518</td>
<td>7%</td>
<td>107</td>
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<tr>
<td></td>
<td>4</td>
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<td>22.7</td>
<td>23.0</td>
<td>23.2</td>
<td>9.7</td>
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<td>22.9</td>
<td>22.9</td>
<td>6.9</td>
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<td>2348</td>
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<td>0</td>
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<td>0%</td>
<td>0</td>
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<td>25.8</td>
<td>48.3</td>
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<td>1428</td>
<td>418</td>
<td>2766</td>
<td>15%</td>
<td>272</td>
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</tbody>
</table>

Table 2 shows that the CB solutions realised with a needle valve (case 3) and an FJVR (case 4) for the same locations are comparable from the perspective of $T_{up}$, $T_{floor}$ and weighted average return temperature of bypassed water $T_{ret}$. In all three locations investigated, the FJVR valve results in an increase of roughly 6% in water volume bypassed during the non-heating period. The last column in Table 2 represents the running costs for the DH company for bypass operation in the 10 m long service pipe during the non-heating season. The cost is calculated as the cost of heat lost in the service pipe during bypass operation minus the heat used in CB, which is paid for by the customer. The cost calculation takes into account the different temperatures at the beginning of the service pipe for different locations in DH network, namely 50°C, 40°C and 37.3°C.

Although the CB realised with an FJVR valve results in a slightly higher volume of bypassed water than with a needle valve, the average supply temperature is lower (because the bypass is stopped if $T_{sup}$ is over 35°C), which means less heat transferred to the bathroom floor heating making the FJVR solution cheaper for the customer. The difference in thermal environment is minimal. On the other hand, CB realised with a needle valve will be a more beneficial solution for the DH company because the customer will be paying for more heat. However, it can be concluded that CB realised with an FJVR valve is better because, in comparison with the needle valve, the adjustment of an FJVR valve is very simple (adjustment of the desired temperature, not the flow rate), the valve automatically changes the needed bypass flow based on the actual conditions in the DH network, and CB is shut down automatically during a heating period.

The traditional external bypass solution realised with the same FJVR valve but without redirecting the bypassed water to the CB results in same bypassed volume, but increases the average return temperature to 35°C, meaning reduced efficiency of the heat plant and higher heat loss from the DH network. Moreover, this solution is the most expensive for the DH company, and the customer has a cold floor in the bathroom.

The traditional external bypass solution realised with the same FJVR valve but without redirecting the bypassed water to the CB results in same bypassed volume, but increases the average return temperature to 35°C, meaning reduced efficiency of the heat plant and higher heat loss from the DH network. Moreover, this solution is the most expensive for the DH company, and the customer has a cold floor in the bathroom.
reference case without FH during the non-heating period. For the DH company, this solution means reduced cost for bypass operation.

Further saving potential can be seen in the application of an electronically controlled step valve with a reduced deadband. Such a solution will result in reduced total bypassed volume and therefore lower running costs to keep the DH network ready for use. The volume of bypassed water will be reduced by roughly 30% compared to using an FJVR with a 3°C deadband.

The heat additionally consumed by the customer using CB realised with an FJVR valve ranges between 150 and 571 KWh for the non-heating period, which represents an additional cost of between 98 and 371 DKK, which is an increase of between 6-20% in total heating cost, as can be seen in Table 2. For the DH company, this income means a reduction in costs related to heat loss in service pipes.

But the main advantage should be seen in making good use of heat which would otherwise be lost in the DH network and in the increase of the thermal efficiency of the heat plants. So customers using the CB system should get some kind of discount on the heat used in CB.

4.2 EFFECT OF COMFORT BATHROOM ON HEAT PRODUCTION AND DISTRIBUTION

Implementation of the CB concept gives a valuable advantage at network level thanks to the lower return temperature. The layout of an existing low-temperature DH network described in [3] was chosen for a case study and the results are reported in Figure 9 and Table 3. It is important to stress that although the DH network layout was adopted from [3], the investigation was made on the reference house described in Section 3.2.1 and not on the houses built in [3].

![Figure 10: Plot of the supply temperature in the network with continuous bypass flow and the “CB concept” during the non-heating season. The ΔT_supply-return in the FH was set to 8°C and flows needed to keep a constant 35°C at the bypass.](image)

The network was modelled in the software Termis® [20] and steady-state simulations were carried out, in which it was ensured that the supply water temperature at the entry point of each in-house substation was kept at 35°C±1.5°C using a continuous bypass flow that was then delivered to the bathroom FH system. This condition represents CB realised with a needle valve. However, since the total volume bypassed by an FJVR valve with a deadband is only 6% higher, the results are considered valid also for the case of FJVR with a deadband.

The heat load in each bathroom due to the radiant FH was set at 30 W for case B. This value was chosen as an average value during the non-heating period assuming that customers do not open the windows in the bathrooms, resulting in an increase of T_r and reduced cooling of bypassed water. Results for this simulation are therefore on the conservative side. As a consequence, the DH water was further cooled down in each house by 7.5°C on average (max.: 8.0°C, min.: 4.0°C, standard deviation 0.8°C). In comparison to the case without CB, the lower return temperature along the distribution pipelines decreases the heat loss from the return pipe by 35%; Table 3 shows that the distribution heat loss from the supply pipe increased slightly. This phenomenon is due to the twin pipe configuration: a slightly greater heat transfer from the supply media pipe towards the surrounding ambient derived from the lower return temperature, because the media pipes are embedded in the same insulation and thus “thermally coupled”. As an overall effect, the heat loss reduction due to the CB concept (by the additional cooling of returning DH water) is approximately 13% and it is therefore possible to conclude that 40% of the heat used by the CB is “free” heat, due to the lower distribution heat loss (in the example, 0.5 kW heat loss reduction with a heat load of 1.2 kW). Moreover, there is a possible improvement on the heat generation side thanks to the lower return
temperature: outside the heating season, the return temperature to the plant is 23.8°C on average, instead of 27.7°C when there is no SH demand.

Table 3: Results from the network simulation with the application of only the “continuous bypass” (A) or “CB concept” in the summer season (B).

<table>
<thead>
<tr>
<th></th>
<th>case</th>
<th>difference</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>Heating Power [kW] (plant)</td>
<td>3.8</td>
<td>4.5</td>
</tr>
<tr>
<td>T_{supply} [°C] (from the plant)</td>
<td>55</td>
<td>55</td>
</tr>
<tr>
<td>T_{return} [°C] (to the plant)</td>
<td>27.7</td>
<td>23.8</td>
</tr>
<tr>
<td>Heating Load [kW]</td>
<td>-</td>
<td>1.2</td>
</tr>
<tr>
<td>Total Heat Loss [kW]</td>
<td>3.8</td>
<td>3.3</td>
</tr>
<tr>
<td>Heat Loss Supply [kW]</td>
<td>2.25</td>
<td>2.3</td>
</tr>
<tr>
<td>Heat Loss Return [kW]</td>
<td>1.55</td>
<td>1</td>
</tr>
<tr>
<td>Heat loss/production [%]</td>
<td>100</td>
<td>72.4</td>
</tr>
</tbody>
</table>

4.3 COST-EFFICIENCY

From the perspective of cost efficiency, CB controlled with an FJVR valve with a 3°C deadband operated during the non-heating period costs the customer in the house located further into the DH network 371 DKK, while close to the main line it will only cost 98 DKK. It is true that $T_{floor}$ is on average 1.6°C higher meaning greater comfort for the first customer, but he could not be forced to pay more just because his house is located further down the DH network. So it is suggested that the DH company will deduct the energy used in the CB during the non-heating period and instead charge the customer with a fixed price for the use of CB. This price should be the same for all customers whatever their location in the DH network and should reflect the fact that the heat used in CB costs less than heat used for SH or DHW preparation. Moreover, with regard to operational costs, the example in the paper demonstrated that a significant part of the additional heat demand for the FH of the bathrooms is counteracted by the lower distribution heat losses and by the benefit of lower return temperatures to the heating plants.

The additional investment costs for implementation of CB can be minimized if incurred during the in-house substation installation phase. In fact, it only involves the hydraulic reconnection of the FJVR valve (which is part of the substation anyway) and the possible installation of additional pipe to the FH loop.

In Denmark, district heating distribution companies are non-profit, generally owned by the local authorities, so costs caused by the heat losses are hidden in the final users’ heat price. Finally, most end-users could be willing to pay the limited extra investment and operational costs, given the improvement in the thermal comfort in their homes.

5 CONCLUSIONS

The flow from the external bypass redirected in the non-heating period to the floor heating in bathrooms gives customers the sensation of a warm floor by increasing the average floor temperature by 0.6–2.2°C, while also reducing the return temperature of otherwise insufficiently cooled bypass water. Furthermore, an increase in operative temperature of 0.3–1°C will contribute to the reduction of relative humidity in the bathroom, usually high in the Danish summer. The cost for the user in the non-heating season would be 98–371 DKK, depending on the location in the DH network, but we suggest an equal charge for all customers whatever their location in the DH network.

Depending on the location in the DH network, the temperature of the bypassed water is reduced from 35°C in the traditional external bypass solution to 23.2–25.6°C. For the case study of a low-temperature DH network supplying 40 low-energy houses, this meant a 13% reduction in heat loss from the DH network during the non-heating period. That corresponds to 40% of the SH demand in the bathrooms during the summer. Moreover, reduced temperature of DH water means better efficiency for heat plants.

The flow of redirected bypassed water can be controlled either by a needle valve providing continuous flow or by a thermostatic FJVR valve with a 3°C deadband resulting in nominal flow rate 50% higher than with a needle valve. However, if we take into account the temperature increase of DH water in the service pipe after each DHW tapping and the ability of an FJVR valve to stop bypass flow when the water temperature is above the set-point temperature, the two solutions are comparable. The solution with an FJVR valve should be preferred because it does not need precise adjustment of the bypass flow and reacts instantaneously to changes in the supply temperature of DH water.

The CB concept can be further improved and heat loss from the district heating network can be further reduced by using an electronic step valve, removing the disadvantage of the deadband in the self-acting FJVR valve. This solution will combine the advantages of needle and traditional FJVR valves, i.e. bypassing only the small flow
necessary to keep the service pipe warm. This would save roughly 30% of the bypassed volume in comparison to using a needle valve and further decrease the average return temperature by 0.3-0.8°C depending on the location in the network.

Redirecting the bypass flow into bathroom floor heating will be more beneficial in non-low-energy buildings where the cooling of bypassed water is expected to be higher.

Using the bypass water for bathroom FH is a cost-effective solution for the DH utilities, the end-users, and society as a whole. The utilities could earn money supplying heat that otherwise would be wasted in distribution heat losses and they also benefit from lower return temperatures. End-users can increase the comfort standard in their houses in an economical way. Society as a whole would benefit from the opportunity to include a larger share of low-grade heat and renewable energy in the heating system, thus decreasing greenhouse gas emissions and contributing to the country’s energy security.

6 ACKNOWLEDGEMENTS

The research is supported by Danish Energy Agency in the Energy Technology Development and Demonstration Programme (EUDP) as a part of the project “Low-temperature district heating in existing buildings”. This financial support is gratefully acknowledged.

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ISI paper III
Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment
Brand M, Svendsen S
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Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment

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Abstract

Denmark is aiming for a fossil-free heating sector for buildings by 2035. Judging by the national heating plan, this will be achieved mainly by a further spread of district heating (DH) based on the renewable heat sources. To make the most cost-effective use of these sources, the DH supply temperature should be as low as possible. We used IDA-ICE software to simulate a typical Danish single-family house from the 1970s connected to DH at three different stages of envelope and space heating system refurbishment. We wanted to investigate how low the DH supply temperature can be without reducing the current high level of thermal comfort for occupants or the good efficiency of the DH network. Our results show that, for a typical single-family house from the 1970s, even a small refurbishment measure such as replacing the windows allows the reduction of the maximum DH supply temperature from 78°C to 67°C and, for 98% of the year, to below 60°C. However for the temperatures below 60°C a low-temperature DH substation is required for DHW heating. This research shows that renewable sources of heat can be integrated into the DH system without problems and contribute to the fossil-free heating sector already today.

Keywords: low-temperature district heating, existing buildings, refurbishment, supply temperature curve, domestic hot water, energy simulation, IDA-ICE, radiators, renewable heat sources,

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1 Introduction

To reduce CO₂ emissions and increase security of supply, Denmark has decided that all buildings will have 100% fossil-free heating by 2035. To reach this goal, Denmark needs to implement energy-saving measures on the demand side, increase efficiency on the production side, and replace fossil fuels with various sources of renewable energy [1]. District heating (DH) is seen as the main solution for achieving the goal because it offers faster and cheaper integration of renewable sources of heat than individual heating sources. Based on the study, Heat Plan Denmark 2008 [2], it is planned to increase the share of DH from the current figure of 46% to 53-70%, with the remaining heating demand, which is mostly in areas with low heating demand density, being supplied by individual heat pumps.

However, this widespread integration of renewable sources of heat with high efficiency and thus reasonable cost will require further reductions in DH supply and return temperatures [3]. Moreover, it will also be necessary to reduce the ratio between DH network heat losses and heat consumption in buildings. This is currently increasing due to reductions in heating demand thanks to [4] the refurbishment of existing buildings and the increasing number of low-energy buildings. The solution is to reduce DH heat losses by using twin-pipe geometry (two media pipes in one casing), thicker insulation, and reduced supply and return temperatures. This philosophy lies behind the concept of low-temperature DH, in which the supply temperature is reduced to 55-50°C and the return temperature to 30-25°C [5,6]. The minimum supply temperature of 50°C is defined as the lowest primary temperature needed to supply the required 45°C domestic hot water (DHW) at tapping points [7]. Low-temperature DH is the optimal concept for the integration of 100% renewable sources of heat.

Reducing heat loss from DH networks makes economic sense for the whole system and also enables supply to areas with low heating demand density, e.g. low-energy housing areas. The economic feasibility and high level of comfort for occupants has been demonstrated in a pilot low-temperature DH project in Lystrup, Denmark, where low-temperature DH supplies an area with 42 low-energy single-family houses [8]. The heat loss from the Lystrup DH network, with design supply/return temperatures of 50/25°C, is only one quarter of what it would be if the network had been designed with the traditional temperatures of 80/40°C [9,10]. The houses are equipped with radiators designed for 50/25°C supply/return temperatures. The 50°C supply temperature to the space heating (SH) system is enough for newly built buildings, designed mainly with floor heating or low-temperature radiators, and there is still the option to boost the supply temperature during the coldest periods. In fact, the DH supply temperature can be even lower, but then it needs an additional system to heat up DHW to the desired temperature level. Such a system has been recently tested in Birkerød in Denmark, where four single-family houses are supplied with DH with a temperature of 40°C. This temperature is enough for the space heating system, but the DHW is additionally heated by a newly developed micro heat pump supplied with heat from the DH network [11].

However, low-energy buildings still comprise only a small share of the building stock while the majority are older buildings with considerably higher heating demand. Existing buildings are usually equipped with SH and DHW systems designed with supply temperatures of around 70°C or higher, so a reduction of DH supply temperature would be expected to cause discomfort for the
occupants. These buildings will continue to make up a large share of the building stock for many years (in Denmark, about 85-90% in 2030 [3]), so the question arises as to whether such buildings can cope with low-temperature DH with supply temperatures of 55-50°C and, if not, what renovation measures need to be carried out on the building envelope and the building heating and DHW systems, or how the DH network should be operated.

Some scientific research has focused on DH in relation to the refurbishment of existing buildings and reporting how much CO₂ emission will be saved, how the reduced heating demand will affect the DH companies from the perspective of heating sold and the reduction in peak heat output of boilers, and suggesting new price tariffs [12]. But we found no papers focusing on the possible reduction of DH supply temperature to existing buildings, and considering their DHW and SH systems from the perspective of integration of renewable heat sources into the DH network.

The maximum supply temperature needed can be reduced by improving the building envelope or by changing the original SH system to low-temperature system. From the long-term perspective, the preferred solution is to reduce the energy demand by improving the building envelope, but due to the cost not every house owner is willing to do this. Changing the SH system is a cheaper and faster solution, but it does not bring any energy savings; it just allows existing buildings to be supplied by DH with reduced supply temperatures. The refurbishment measures carried out on existing houses, e.g. those built in the 70s, vary from no measures (original state) to extensive renovation, including replacing the windows and wall and roof insulation. Replacing the windows is the most typical refurbishment carried out on these houses, because the window lifetime of 30 years has passed and a relatively small investment brings considerable heat savings.

1.1 Proper design and operation of space heating system
To evaluate the possible reduction of supply temperature to existing buildings, we should start with the proper design of a SH system and with realistic operation conditions. According to Danish Standard DS418 [13], SH systems should be designed for an operative temperature of 20°C as indoor temperature, a steady-state outdoor temperature of -12°C, and no internal and solar heat gains. The operative temperature involves both air and mean radiant temperature, and thus defines how the occupants perceive the environment. Nevertheless, it is very probable that SH systems in the 70s were in fact designed with the air temperature alone, because the hand calculation of operative temperature is simply very complicated. In low-energy buildings, the difference between operative and air temperature is very small, so error caused by using air temperature is negligible, but in older buildings, constructed without good insulation properties, the difference can be rather high due to cold surfaces. This means that, even when the air temperature is at a comfort level of 20°C, the occupants can feel cold because the operative temperature is lower (e.g. 18°C). Moreover occupants tend to set the operative temperature to 22°C instead of 20°C [10]. This raises the question of how an SH system designed on the basis of air temperature will perform when the occupant increases the set-point temperature and how this will affect the DH network. Both will result in the need for higher heat output from the SH system and thus increased water flow and higher return temperatures from the SH and also DH system and, if one of the systems does not
have enough hydronic capacity, this can cause thermal discomfort for customers. This clarification is therefore important in any investigation into the possible reduction of the DH supply temperature.

1.2 DHW system

Considering the possibilities of reducing the DH supply temperature to 50°C, it should be kept in mind that DH is used also for DHW heating. For DH supply temperature reduced below 60°C, the recently used DHW substations need to be replaced with specially designed low-temperature DHW substations. Such substations with compact and very effective heat exchangers are already in use in Lystrup [5] and provide a high level of comfort for occupants as well as good cooling of DH water. For the design conditions, i.e. 13.2 L/min of DHW heated from 10°C to 45°C, the substation provides cooling on the primary side from 50°C to 20°C, [14]. The only change the customers experience is the maximum DHW temperature reduced to about 47°C which is still enough. However since the DHW temperatures is below 55°C a special attention should be paid to the risk of Legionella, increased mainly in the temperature range 30°C-50°C. Most of the national DHW standards therefore requires minimal temperature of DHW to 55°C, simply not reachable by low-temperature DH. Nevertheless due to the German standard DVGW 551 [15] there is no requirement for the minimal DHW temperature for “small DHW systems with volume below 3L” (excluding volume of the heat exchanger and DHW circulation loop) giving the possibility to use low-temperature DH anyway. In fact most single-family houses with a modern DHW system will fulfil this requirement, because 3L for ½” pipe (DN 15) means 15 m of pipes, which should be enough. If the volume of water in the DHW pipes is above 3L, the piping can be changed from ½” to 3/8”, which will increase the maximal length of the pipes to 25 m.

Where buildings currently use an in-house substation with DHW storage, a 3L requirement can be fulfilled by replacement of the original substation with a 120L storage tank for DH water [16]. The principle is shown in Figure 4. The storage tank acts as buffer tank for DH water, and DHW is heated instantaneously in the heat exchanger only when needed. In such a solution, there is no storage of DHW, which otherwise should follow the rules about a minimum temperature of 55°C. DHW circulation is not prohibited for either type of DH substation, but it is not recommended because of large heat losses.

![Figure 4 – Principle of low-temperature DH substation with buffer tank for DH water, based on [16].](image-url)
For multi-storey buildings with a traditionally designed DHW system with vertical risers, low-temperature DH can only be used if some kind of DHW disinfection is provided, because the volume of the DHW system is above 3L. Thermal disinfection is a well-known concept, but the connection of low-temperature DH means there is a need for an additional source of heat, because efficient thermal disinfection needs at least 60°C while the supply temperature of low-temperature DH is only 50°C. The higher the disinfecting temperature is, the shorter time water needs to stay at that temperature to be disinfected [17]. If we exclude fossil-fuel-based heat sources, one solution could be an electric resistance heater coupled with the heat exchanger for heat recovery, cooling the DHW to ensure that all flats get a temperature of 45°C in kitchen taps. Another possibility is the use of Advanced Oxidation Technologies (AOT), where UV lamps form free radicals by irradiation of a catalytic surface. The radicals then break down contaminants in the water. The process occurs only inside the purifier and leaves no harmful residuals in the water [18]. One UV lamp is installed on the cold water supply to the DHW heat exchanger and another UV lamp on the DHW circulation just before the heat exchanger. The solution was tested in [17] with good results. The disadvantages of such a solution are considerable investment and running costs, because the lamps need to be changed once a year. Over the lifetime of 20 years, it means around DKK 3800 per annum [19]. This solution could work for DHW systems supplying several flats where the tenants share the costs, but as an alternative for use in single-family houses it is currently too expensive. However a state-of-the-art solution for multi-storey buildings will be use of individual in-house substations in each flat [20], so called “flat stations”, resulting in the same requirements for DHW system as for the individual connection of single-family houses. However need of a complete renovation of the DHW and SH system makes this solution expensive and therefore not commonly used for the existing buildings.

1.3 Objective
This paper describes research into the possible reduction of DH supply temperatures for a typical single-family house from the 70s at three different stages of building envelope refurbishment, so that we can evaluate the possibility of integrating existing buildings into low-temperature DH networks based 100% on renewable heat sources. The investigation takes into consideration the proper designing of the SH system and the effect of the indoor set-point temperature on the annual heating demand, maximum heat output, and hydronic conditions in SH and DH systems.

2 Methods
For our investigation, we chose a typical one-storey single-family house with pitched roof built in 1973, with an area of 157m². The house was part of the Realea renovation project [21], in which four typical single-family houses built in 70s underwent different levels of refurbishment to see how much energy can be saved with the different kinds of refurbishment measures. All the houses from the Realea project are connected to the traditional district heating network and thus feasible for our investigation. Moreover all the houses were extensively measured for two years before and after refurbishment, which gave enough information to build and calibrate the numerical model which we used for the investigation. We modelled the house as a multi-zone model (each room was modelled as an individual zone plus one zone representing the attic) in the commercially available software
IDA-ICE 4.22 [22]. The difference in the heating demand measured in the house and simulated with our model was 2.5% for the non-renovated house. For this comparison, we used the real weather data measured during the period of house measurements. All the subsequent simulations were made using the weather file, Design Reference Year (DRY) for Denmark.

First we designed the space heating (SH) system for the house in its original state from 70s, i.e. the non-renovated house, to define the maximum heating output for the SH system and the limit flow rate of DH water, both needed for dimensioning the space heating (SH) system and the DH network. The SH system was designed as a two-pipe system with the radiators connected directly to the DH network and without mixing loop. Therefore the supply temperature to the radiators is equal to the DH supply temperature. The design was made in accordance with Danish Standard DS418 [13], with temperature levels of 70/40/20°C (supply/return/air temperature), but based on the indoor air temperature. The reason for this approach was to evaluate the effect on the SH system and DH in cases when the SH system really was designed on the basis of air temperature instead of operative temperature.

To simulate realistic conditions, we later introduced constant internal heat gains 4.18 W/m² [23] and we ran an annual simulation for the DRY weather file. As the first step, we found the minimum DH supply temperature needed to ensure operative temperatures of 20°C and 22°C during the coldest period of the year. The operative temperature 22°C was chosen to make the investigation valid for realistic human behaviour. In the next step, we introduced control of DH supply temperature based on the outdoor temperature so that the DH supply temperature could be low as possible but at the same time without the DH water flow exceeding the flow limit defined for the design conditions based on DS418. This relationship between outdoor temperature and DH supply temperature is called the supply temperature curve. Using the same approach, we found the supply temperature curve also for both a lightly and an extensively renovated house, but we kept the SH system unchanged, i.e. in the state as it was originally designed. This represents the situation, when the housing envelope is improved (windows, insulation on the ceilings, etc.), but the SH system is left unchanged.

In the second step, we also replaced the original radiators with low-temperature radiators. The length and the height of the radiators were the same, but the depth was changed. The original radiators had two panels (plates filled with water) and one convector plate increasing the heat transfer area (i.e. radiator type 21) [24], now the radiators had three panels and three convector plates (type 33). The power curve coefficient n, which is needed for recalculation of the heat output based on the supply temperature, was changed accordingly.

As the last step, we also considered the possibility of increasing the limit for heating water flow rate, both the in DH network and in the SH system. The results are reported separately for the different stages of building envelope refurbishment.
3 Results and Discussion

3.1 Non-renovated house from 1973

The SH system was designed for an outdoor temperature of -12°C [13], but taking into consideration an air temperature 20°C, our example of a non-renovated house results in a heat loss of 9.2kW. The DH supply temperature of 70°C theoretically corresponds to maximum water flow (in both the SH and DH systems) of 264 L/h (with cooling in the SH system of 70/40°C). However, real radiators are available only in certain nominal power and thus a slight over-dimensioning by a total of 550 W results in a maximum water flow rate of 245 L/h with slightly more cooling, i.e. the return temperature is 37.6°C instead of the designed 40°C (see Table 1).

Table 1 – Comparison of heating demand, peak heat output (P_{max}) and weighted average return temperature (T_{RW}) for all the investigated cases.

<table>
<thead>
<tr>
<th>T_{operative}</th>
<th>internal heat gains</th>
<th>T_{supmax}</th>
<th>RAD</th>
<th>P_{max}</th>
<th>TRW heating demand</th>
<th>P_{max} reduction</th>
<th>heating demand reduction</th>
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<td>°C</td>
<td>W/m²</td>
<td>°C</td>
<td></td>
<td>kW</td>
<td>°C</td>
<td>[MWh/a] [%]</td>
<td>[%]</td>
</tr>
</tbody>
</table>

**non-renovated house - basic**

- 20b: 0 70.0 O 9.2 37.6 x - -
- 20: 0 70.0 O 9.4 40.2 x - -
- 20: 4.18 73.0 O 9.9 30.1 20.0 - -
- 22: 4.18 78.0 O 10.5 32.9 24.6 -6% -23%
- 22: 4.18 67.3 LT 10.5 27.6 24.6 -6% -23%

**non-renovated house - advanced**

- 22c: 4.18 70.0 O 10.5 35 24.5 0% 0%
- 22d: 4.18 78.0 O 10.5 33.1 24.6 0% 0%
- 22e: 4.18 78.0 O 7.8 32.5 21.7 25% 12%

**light renovation - new windows**

- 20: 0 70.0 O 7.7 33.0 x
- 20: 4.18 62.0 O 7.8 27.1 14.9 21% 26%
- 22: 4.18 67.0 O 8.3 30.4 18.4 21% 25%
- 22: 4.18 59.0 LT 8.3 25.7 18.4 21% 25%

**extensive renovation**

- 20: 0 70.0 O 5.8 28.2 x
- 20: 4.18 53.1 O 5.47 24.7 9.9 45% 50%
- 22: 4.18 57.0 O 5.80 28.1 12.4 45% 49%
- 22: 4.18 50.0 LT 5.82 24.1 12.4 44% 50%

\(a\): O = original radiators, LT = low-temperature radiators
\(b\): dimensioned based on air temperature
\(c\): maximum flow limit increased to 400 L/h
\(d\): time delay in DH supply temperature control
\(e\): simulated with “measured weather data input”
The air temperature for all rooms was 20°C, but the minimum operative temperature was only 18.4°C. This was due to cold walls and windows, so the occupants would be not satisfied. To reach 20°C operative temperature in all rooms (with the same SH system), the maximum heat output would have to increase to 9.4kW and the maximum water flow rate to 280 L/h (see Table 1). An increase of 0.2kW in maximum heat output and of 35 L/h in water flow rate is not a problem for the DH network or for the SH system, so it makes no practical difference whether the SH system is designed for an air or operative temperature. The only thing that matters is to be aware of the difference between air and operative temperature from the perspective of the thermal comfort.

Simulation with an SH supply temperature of 70°C (design condition), the DRY weather file, and internal heat gains (4.18W/m²) representing the realistic use of the building increases the maximum needed heating output to 9.9 kW and the maximum flow rate 324 L/h. Furthermore, occupants tend to operate the SH system with an operative temperature of 22°C, which increases the maximum heating output further to 10.5 kW and the maximum flow rate to 432 L/h (not reported in Table 1). This corresponds to only a 14% increase of the maximum heat output, but to a 64% increase in the design water flow rate. This high increase in flow rate is caused by the fact that the SH system has to operate above the design conditions, and it results in reduced cooling and increased flow of the heating water. This shows that realistic use of the SH system results in the need for greater heat output than it was designed for, and the most common solution is an increase of DH supply temperature above 70°C to avoid problems with enough flow in DH network and to satisfy all customers.

3.1.1 Supply temperature curve construction

The results presented so far were simulated with a constant supply temperature of 70°C and the option of exceeding the design flow rate limit. Now we limit the maximum DH flow rate to 264L/h (value based on DS418 design) and modify the DH supply temperature curve based on the outdoor temperature to the lowest possible level without violating the flow limit and a high level of thermal comfort.

The optimal supply temperature curve (see Figure 1) for a non-renovated house was found by using the following procedure. First, we found the lowest constant supply temperature that gave an operative temperature of 22°C during the whole heating period (DRY weather file) without exceeding the maximum flow rate of 264 L/h. This result defined the supply temperature for the coldest period and for $T_{op}$ 22°C is represented by point [-20;78] where the coordinates are outdoor temperature and DH supply temperature, respectively. Then we changed the supply temperature to 50°C (low-temperature DH concept), removed the DH flow limitation and the result revealed the outdoor temperature when the flow rate with 50°C supply temperature exceeds the 264 L/h limit, i.e. point [5.5;50]. Next we connected the two points obtained in the previous two steps and defined the supply temperature curve (see Figure 1). The curve is valid for the case of a non-renovated house heated to an operative temperature of 22°C, and the highest needed supply temperature is 78°C.
Figure 1 shows that, for the situation with the supply temperature curve defined only by two points ([-20;78] and [5.5;50]), the flow exceeds for some periods the limit of 264 L/h (Figure 1 - diamond markers). This situation occurs roughly for 170 hours over a year and will result in reduced thermal comfort for occupants. The reason the water flow is above the limit just for some periods is the thermal capacity of the building and the history of outdoor temperature. When the outdoor temperature increases fast, resulting in a fast drop of the DH supply temperature, the inside of the building construction is still “cold” from the period with lower outdoor temperatures, so it is still receiving heat from the indoors to establish a new heat balance condition (i.e. to heat up).

The simplest solution would be to raise the whole supply temperature curve until the maximum flow rate remains below 264 L/h, but this will result in long periods with a higher supply temperature than needed and thus higher heat losses from the DH network and reduced efficiency in the use of heat sources. A better solution is to find the supply temperature curve that over the whole heating season results in a flow close to the limit value. Figure 1 shows such a supply temperature curve, defined by the addition of just one additional point [-10;71]. The DH flow rate (triangle markers) is exceeded only three times (totalling around 1 hour), which can be negligible considering the thermal inertia of the building.

Figure 2 shows the percentage of hours in a year when the DH supply temperature needs to be increased above 50°C to satisfy the heating demand of the house. The figure also reports results for the other cases investigated, which are described in Sections 3.2 and 3.3. It can be seen that for the non-renovated house, the supply temperature to the SH system needs to be increased above 50°C for 41% of hours in a year. For 22% of hours in a year, the temperature needed to be above 55°C, and so on.
Figure 2 – Percentage of hours during a year with supply temperature higher than 50°C.

Figure 2 shows that, in the least favourable conditions represented by the non-renovated house in its original condition from the 70s and an operative temperature 22°C, the DH supply temperature needed to be increased above 50°C for 41% of the year, 8% of hours above 60°C, 1% hours above 70°C and 0.2% of hours (20 hours) above 75°C.

An alternative solution to the optimised supply temperature curve (constructed from more than two points) could be the introduction of a time delay in the DH supply temperature control for periods when the outdoor temperature increases. We simulated the non-renovated house with an operative temperature of 22°C and the original temperature supply curve defined by just two points, but introduced a time delay of six hours in the periods when the outdoor temperature was increasing. This case is in figures denoted “non-renovated 22°C Tout SHIFT”. For the periods when the outdoor temperature drops, the current value of the outdoor temperature was used. The time delay reduced the maximum flow rate to 280 L/h, but increased the period with a supply water temperature of 50-60°C by 12%. Further increases in the time delay would decrease the maximum water flow rate, but increase the periods with a supply temperature above 50°C.
3.1.2 Measured weather data input
The results above are based on the DRY weather file, which contains extreme values for designing buildings and buildings services. With real data (denoted in Table 1 with upper case “e”) measured during 2009 close to the site of the Realea project, the heating demand of the house decreases by 12% and the number of hours with a supply temperature above 50°C decreases by 10% in comparison with the DRY weather file.

3.1.3 Limited DH supply temperature
The scenario presented in 3.1 considered the availability of a DH heat source providing supply with a temperature of up to 78°C. However, this cannot always be assured, so it is interesting to investigate conditions when the supply temperature is limited to 70°C. The lower supply temperature requires an increase in the limit in water flow rate in both the SH and the DH system.

In the case of the non-renovated house, the 70°C supply temperature limitation results in a maximum water flow rate of 432 L/h, i.e. 64% more than in the original case. By proper adjustment of the supply temperature curve, this flow rate can be kept for the whole heating period. In comparison with the original case (max. supply temperature 78°C, max. flow rate 264 L/h), the percentage of hours with the supply temperature over 50°C decreases from 41% to 29% hours, and the period with the supply temperature above 60°C drops from 3% to 2% of hours. Increased maximum flow rate also slightly increases the weighted average return temperature from the SH system, from 32.9°C to 35°C. But the really important question is whether the DH network and the house SH system can handle a 64% increase in maximum flow rate.

Hydronics of the SH system
The standard approach to designing the diameter of pipes for SH systems in the 70s was the method of economical velocity with values suggested between 0.8 and 1.2 m/s [25]. Taking into account the pressure drop on the path to the critical radiator (usually the most distant one) and by including the pressure drop in the radiator, the thermostatic radiator valve and the local pressure losses, the whole SH pipe network in the house can be built from steel pipes with a diameter of 3/8” (DN10). The maximum velocity of the heating water in this case with an operative temperature set-point of 20°C is calculated to 0.42m/s and the total pressure drop to 9.4 kPa. In fact, the SH piping network is usually made from ½” (DN15) pipes and this reduces the maximum velocity to 0.25m/s and the total pressure drop to 2.8 kPa. For directly connected SH systems, the additional pressure drop of 4.8 kPa from differential pressure controller (k-v value 1.6 for flow 264 L/h) should be added, and for indirectly connected SH systems, the pressured drop of the SH heat exchanger should be taken into account.

A 64% increase in the maximum flow rate (to 432 L/h) in 3/8” pipes means a maximum velocity of 0.71 m/s and a total pressure drop of 25.3 kPa (2.53 mH2O); in ½” pipes, this means a maximum velocity of 0.42 m/s and a total pressure drop of 13.7 kPa (1.37 mH2O). Both these figures include a 7.3 kPa pressure drop from the differential pressure controller. Directly connected SH systems typically have 50 kPa (0.5 bar) of available differential pressure, so even the less favourable solution with 25.3 kPa pressure drop does not represent any problem for SH systems. For indirectly
connected systems, the maximum available pressure depends on a circulation pump, which can easily be changed if the increased pressure drop exceeds the head pressure of the pump. But it should be emphasised that, to avoid noise problems, the maximum suggested velocity of water in pipes should be below 1 m/s.

3.1.4 Low-temperature radiators

Furthermore, the low-temperature DH concept can be extended to existing buildings by replacing existing radiators with low-temperature radiators and thus change the design operating conditions from 70/40/20 to e.g. 50/25/20. The DH flow limit is still kept at 264 L/h, but bigger radiators mean an opportunity to reduce the supply temperature. In our example, we decided to keep the length and height of the radiators and only changed their depth. This meant that while all the original radiators were type 21, the low-temperature radiators were type 33 [24]. In this way, replacing the radiators is very easy, because the radiators can be connected to the existing piping system without any changes.

Figure 3 – Supply temperature curves for all the investigated cases.

Figure 3 shows that the low-temperature radiators allow the non-renovated house to reduce the original supply temperature curve from a maximum value of 78°C down to 67.3°C. At the same time, the outdoor temperature when the DH supply temperature needed to be increased over 50°C (also called the breaking point) moved from 5.5°C to 1°C, which is 4.5°C lower than in the original case. Lowering the supply temperature curve also reduces the number of hours with supply temperature above 50°C by 83% to 7% of hours in the year (see Figure 2) and by 94% to 0.15%
hours in year for the supply temperature above 65°C. The weighted average return temperature from the SH system decreases from 32.9°C to 27.6°C.

3.1.5 One-pipe system
The simulations do not consider one-pipe heating system accounting for ten per cent of the houses in Denmark. One-pipe heating system is not seen as a good solution for low-temperature DH because it can be characterised as system with need of a higher supply temperature, low cooling and therefore higher flow to DH than a two-pipe system. Increased flow can lead to problems with the hydraulic capacity of the DH network and higher supply and return temperatures result in higher heat losses from the DH network and lower efficiency of heat sources. Therefore the reduced cooling is usually fined by the DH company. The cost of replacing a one-pipe system with a two-pipe system for single-family house is estimated at DKK 60 000 [26]. However one-pipe systems should be still analysed to evaluate their influence on low-temperature DH.

3.2 Light renovation – changed windows
As a typical light renovation measure, we chose replacement of the windows, because the expected lifetime of 30 years is over, so most of these houses have already done so. This assumption is further strengthened by the fact that for a relatively small investment house owners get considerable energy savings. In the period around year 2000, the original windows from the 70s (overall $U_w$ value of 3.2 W/(m².K)) were replaced with “energy-glazing” windows with a $U$-value of 1.2 W/(m².K), resulting in an overall $U_w$ value of 1.5 W/(m².K). The new windows also reduce air infiltration by 15% to 0.41 h⁻¹ (0.278 L/(m².s)) [21].

Peak heat output and heating demand are reported in Table 1. First we can see that the heating demand for SH decreased from 20.0 MWh to 14.9 MWh for an operative temperature of 20°C and from 24.6 MWh to 18.4 MWh for an operative temperature of 22°C. In both cases, this corresponds to a reduction of 27%. The maximum heating power needed for the SH system with an operative temperature of 22°C is reduced from 10.5 kW to 8.3 kW, which makes it possible to define a new supply temperature curve. From the original 78°C for a non-renovated house, the maximum value decreased to 67°C with the breaking point of 0.5°C. Replacing the original radiators with low-temperature radiators shifts the supply temperature curve down by 8°C and the maximum supply temperature is therefore further reduced from 67°C to 59°C. In comparison with the non-renovated house, the number of hours with a supply temperature above 50°C (see Figure 2) decreases by 60% to 16.5% of hours over the year and by 95% to 2% of hours with low-temperature radiators.

This reduction in the heating demand brings closer the possibility of using 50°C all year round. This becomes possible if the DH network can guarantee a maximum flow rate for SH of 520 L/h. For directly connected SH systems designed with ½” pipes and differential pressure controller with k-v value 1.6, this means a maximum velocity of 0.51 m/s and a design pressure drop of 19.2 kPa (19.2 m H₂O), both of which are in an acceptable range. This solution would also be possible for 3/8” piping, but would result in a maximum velocity of 0.85 m/s and a maximum pressure drop of 46.2 kPa (4.6 m H₂O), which are very close to the maximum values.
3.3 Extensive renovation

The extensive renovation of the house was modelled on the Realea project [21]. This was carried out by adding 300 mm insulation above the ceiling \((\lambda_{\text{ins}}=0.56 \text{ W/(m}^2\text{K})\) including the effect of wooden beams), and by insulating the wooden beams that bear the roof construction with 125 mm of insulation \((\lambda_{\text{ins}}=0.039 \text{ W/(m}^2\text{K})\) and 13 mm gypsum board. The overall heat transfer coefficient \(U\) for the ceiling construction was reduced from 0.48 to 0.14 W/(m\(^2\)K) and for the insulated beams from 1.1 to 0.24 W/(m\(^2\)K). Moreover, the thermal bridges between the inner and outer wall around the windows were reduced by inserting 30 mm of polystyrene (in simulations modelled as reduced linear heat loss from 0.0736 to 0.0192 W/(K.m)). Windows facing west and north were replaced with triple-glazed low-energy windows with an overall \(U_w\) value of 0.9 W/(m\(^2\)K) and a \(g\) value of 0.5. The change of the windows reduced air infiltration by 15\%, as in the case of light renovation, i.e. to 0.613 h\(^{-1}\). Table 2 show the impact of the individual renovation measures on the annual heating demand.

Table 2 – Impact of individual refurbishment measures on annual heating demand; measured weather data input, set-point temperature \(t_{\text{air}}=20^\circ\text{C}\).

<table>
<thead>
<tr>
<th>Heating demand</th>
<th>Heating demand reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>[MWh/a]</td>
<td>[MWh/a]</td>
</tr>
<tr>
<td>non-renovated house</td>
<td>18.8</td>
</tr>
<tr>
<td>improved thermal bridges on windows</td>
<td>18.3</td>
</tr>
<tr>
<td>additional 300 mm insulation to the ceiling</td>
<td>14.4</td>
</tr>
<tr>
<td>reduced air infiltration with new windows from 0.327 -&gt; 0.27795 (L/m(^2)s)</td>
<td>13.6</td>
</tr>
<tr>
<td>new low-energy windows on W + N, (U_w) from 3.2 -&gt; 0.9 W/(m(^2)K)</td>
<td>10.3</td>
</tr>
<tr>
<td>added 125 mm insulation to wooden bearings for the roof construction</td>
<td>9.4</td>
</tr>
</tbody>
</table>

It can be seen that the largest reduction in heating demand was achieved by insulating the ceiling: roughly 21\%. The next most effective solution was replacement of the windows (only west and north façades), contributing to a reduction in annual heating demand of roughly 18\%. The change of the windows would have had greater effect if all the windows had been changed. The other refurbishment measures each contributed less than 5\%. The numbers given describe only the heating demand saving potential and say nothing about the economic feasibility of the individual measures, but these refurbishment measures were chosen in Realea project as the most cost-effective solutions.

Compared with its original state, extensive renovation of the house reduces the annual heating demand for SH from 20.0 MWh to 9.9 MWh for an operative temperature of 20\(^\circ\)C and from 24.6 MWh to 12.4 MWh for an operative temperature of 22\(^\circ\)C. In both cases, this is a reduction of about 50\%. Moreover, the peak heating demand in each case is reduced from 9.4 kW and 10.8 kW to 5.5 kW and 5.8 kW, respectively.

The flow rate limit 264 L/h for the extensively renovated house represents a supply temperature curve with a maximum value of 60\(^\circ\)C, and this fits in with the philosophy of low-temperature DH with increased supply temperature during winter. A supply temperature in the range 50-55\(^\circ\)C is
needed only for 3% (295 hours) of a year and for just 20 hours in range 55-60°C. The weighted return temperature is 28°C. Furthermore, replacing the radiators with low-temperature radiators means that the extensively renovated house can be supplied by DH water with a constant temperature of 50°C during the whole year without exceeding the maximum flow of the original DH network and a weighted return temperature 24.1°C.

3.4 Impact of DHW on reduction of annual weighted return temperature
DHW demand was expected to be 79m³/a of 45°C DHW (corresponding to 800kWh/(a.person)) [27], which is only 8% of the volume needed for SH in case of non-renovated house heated to 22°C. The return temperature from the DHW heat exchanger changes with varying DH supply temperatures and cold water temperatures. It can be precisely calculated, but as a rough estimation we can expect an average return temperature of 20°C. In the example of the non-renovated house, it reduces the annual weighted average return temperature from 32.9°C to 32°C, which corresponds to an 8% decrease.

4 Conclusion
The results show that a typical single-family house built in 70s and recently still without any renovation measures can be heated by low-temperature DH with supply temperature 50°C to an operative temperature of 22°C roughly for 59% (3600 hours) of year. However to avoid compromising of thermal comfort or exceeding the design flow rate in the DH network the DH supply temperature should be raised above 60°C for roughly 8% of year (700 hours). Considering the average DH supply temperature of 79°C during the heating season 2010 solely from the perspective of the SH systems, the DH supply temperature can be considerably decreased even for non-renovated houses, representing the most unfavourable conditions.

However in reality, most houses from the 70s have already replaced their original windows, which mean that the maximum value and the duration of increased DH supply temperature can be further reduced. In our example, it means a reduction from 8% to only 2% (175 hours) of hours in the year when the temperature is above 60°C. Therefore it shows that for most houses from the 70s it is possible to decrease the DH supply temperature below 60°C for almost whole year and integrate renewable sources of heat with high efficiency, thus contributing to the fossil-free heating sector already today.

Furthermore, by installing low-temperature radiators (with the same outer dimensions as the original ones), the maximum supply temperature can be reduced to 59°C so that there is no period with a DH supply temperature over 60°C. The same supply temperature curve is also valid for the extensively renovated house (new low-energy windows and attic insulation) with the original SH system. If the extensively renovated house also replaces its space heating system with low-temperature radiators, it can then be supplied all year around with a DH supply temperature of 50°C. The duration of periods with a DH supply temperature above 50°C is reported for an operative temperature of 22°C to model a realistic set-point temperature preferred by occupants. The durations for an operative temperature of 20°C will be shorter.
However, the reduction of the DH supply temperature to below 60°C does require changing DHW heat exchangers to special low-temperature heat exchangers and traditional DHW storage tanks to low-temperature DH storage tanks. We suggest that DH utilities should start requiring the replacement of existing DH substations with low-temperature DH substations already today, because this will ensure that in 20 years (the typical lifetime of a DH substation) all newly installed DH substations will be ready for low-temperature DH.

The DH supply temperature curve, which represents changes in DH supply temperature based on the actual outdoor temperature, needs to be defined by more than two points. Alternatively, the outdoor temperature input needs to be delayed. Otherwise the DH network will not be used in an optimal manner, which will result in greater heat losses and poor cost-effectiveness. Moreover the supply temperature curve can be further shifted to lower temperatures if the maximum guaranteed DH flow rate is increased with additional head pressure from pumps in the DH network. This is documented in the example of the non-renovated house where the maximum supply temperature decreased from 78°C to 70°C while the annual weighted average return temperature increased only by 3°C. This solution will therefore make it easier to integrate renewable sources of energy, but the impact on DH networks needs further investigation.

The consequences of using air temperature instead of operative temperature when designing a SH system were shown to be marginal, both for the SH and the DH system. In reality, the radiators will tend to be over-dimensioned because the designers want to be sure that the system provides enough heat and this will make it possible to use slightly lower supply temperatures. Over-dimensioning of the DH network by 20-30% is also expected and this will contribute to better integration of existing buildings into low-temperature DH networks. However, we cannot rely on the over dimensioning.

Percentage reduction in the heating demand is not the same as the reduction in peak heat output. Light renovation results in a 25% reduction in heating demand but only to 20% reduction in peak heat output. Similar is valid also for extensive renovation, with 50% and 45% reduction of heating demand and peak heat output respectively.

The heating demand of existing buildings is expected to decrease linearly to 50% of its present value by 2050. This reduction in heating demand, however, will cause no difficulties, if the present DH concept is changed to low-temperature DH. The low-temperature DH concept still requires further optimisation, and more work is needed on DH network design and operation to take into account the integration of renewable sources of energy, but the low-temperature DH concept can be introduced already today because existing buildings do not represent such big problems as might have been expected.

5 ACKNOWLEDGEMENTS

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6 REFERENCES


[18] www.walleniuswater.com, reached 27.3.2013

[19] Personal correspondence with Efsen Engineering AS, representative for Wallenius Water AS in Denmark
7 List of Figures – captions

Figure 1 – Construction of weather-compensated supply temperature curve for non-renovated house, set-point temperature 22°C.

Figure 2 – Percentage of hours during a year with supply temperature higher than 50°C.

Figure 3 – Supply temperature curves for all the investigated cases.

Figure 4 – Principle of low-temperature DH substation with buffer tank for DH water [16].
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Table 2 – Impact of individual refurbishment measures on annual heating demand; measured weather data input, set-point temperature $t_{\text{air}}=20^\circ\text{C}$.
Schedule for Internal Heat Gains
### Overview of internal heat gains

<table>
<thead>
<tr>
<th>List of Internal HG</th>
<th>Equipment / event</th>
<th>Continuous power [W]</th>
<th>Energy [kWh/24h or cycle]</th>
</tr>
</thead>
<tbody>
<tr>
<td>fridge</td>
<td>32</td>
<td>0.77</td>
<td></td>
</tr>
<tr>
<td>washing machine</td>
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<td>1</td>
<td></td>
</tr>
<tr>
<td>tumble dryer</td>
<td>1850</td>
<td>1.85</td>
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</tr>
<tr>
<td>dish washer</td>
<td>1000</td>
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<td></td>
</tr>
<tr>
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<tr>
<td>TV</td>
<td>200</td>
<td></td>
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</tr>
<tr>
<td>Computer</td>
<td>100</td>
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<td></td>
</tr>
<tr>
<td>Breakfast</td>
<td>1+2kW in 5 minutes</td>
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</tr>
<tr>
<td>Ironing</td>
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<tr>
<td>IHEU</td>
<td>36</td>
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<table>
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<tr>
<th>addition to routine events [kWh/week]</th>
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<td>event</td>
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<td>vacuuming</td>
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<tr>
<td>ironing</td>
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<tr>
<td>total</td>
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</table>

<table>
<thead>
<tr>
<th>EQUIP + LIGHTS ... difference my SCH vs. constant 3.5W/m2</th>
<th>people ... difference my SCH vs. Constant 1.5W/m2</th>
<th>total internal HG</th>
</tr>
</thead>
<tbody>
<tr>
<td>EQUIP + LIGHTS: const. 3.5 W/m2</td>
<td>EQUIP + LIGHTS: model</td>
<td>people: const. 1.5 W/m2</td>
</tr>
<tr>
<td>differen ce</td>
<td></td>
<td>people: model</td>
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<tr>
<td>week [kWh/week]</td>
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<td>40.07</td>
</tr>
<tr>
<td>const. power [W]</td>
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<td>1.5</td>
</tr>
<tr>
<td>year [kWh/year]</td>
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<td>2084</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6945</td>
</tr>
</tbody>
</table>

| week [kWh/week]                                         | 93.5                                              | 71.39             |
| const. power [W]                                        | 3.5                                               | 0.8               |
| year [kWh/year]                                         | 4862                                              | 22.1              |

| const. power [W]                                        | 3.5                                               | 1.5               |
| year [kWh/year]                                         | 4862                                              | 1149              |
|                                                          |                                                   | 2130              |

| total internal HG                                       | 93.5                                              | 133.6             |
|                                                      |                                                   | 5842              |
## Detailed schedule of internal heat gains for working day

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<th>heat emitted from people [W/ wd]</th>
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<th>320</th>
<th>320</th>
<th>1280</th>
<th>5.12</th>
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</table>

### Week days (wd)

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<th>No one at home</th>
<th>Working</th>
<th>Cooking/res</th>
<th>Eating</th>
<th>Resting</th>
</tr>
</thead>
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<td>room</td>
<td>23:00 - 7:00</td>
<td>7:00 - 8:00</td>
<td>8:00 - 15:00</td>
<td>15:00 - 17:00</td>
<td>17:00 - 18:00</td>
<td>18:00 - 19:00</td>
<td>19:00 - 23:00</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
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</tr>
<tr>
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<td>1</td>
<td>1</td>
<td>1</td>
<td></td>
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<tr>
<td>Tech. room 9.1</td>
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<tr>
<td>Room 9.1</td>
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</tr>
<tr>
<td>Bathroom 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Bedroom 15.0</td>
<td>2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Kitchen 9.4</td>
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<td>Corridor</td>
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<td>7</td>
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<td>1</td>
<td>1</td>
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### Heat emitted [W] week days (wd)

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<thead>
<tr>
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<th>1100</th>
<th>200</th>
<th>200</th>
<th>200</th>
<th>545</th>
<th>318</th>
<th>477</th>
<th>586</th>
<th>1818</th>
<th>218</th>
<th>3472</th>
<th>7.4</th>
<th>kWh/ wd</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fridge + IHEU</td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td>68 kWh/ wd</td>
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<tr>
<td>Washing machine</td>
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<tr>
<td>Tumble dryer</td>
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<tr>
<td>Dish washer</td>
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<td></td>
<td></td>
<td>250 kWh/ wd</td>
</tr>
<tr>
<td>Cooking</td>
<td></td>
<td></td>
<td></td>
<td>1100</td>
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<tr>
<td>TV</td>
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<td></td>
<td></td>
<td>200 kWh/ wd</td>
</tr>
<tr>
<td>Computer</td>
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</tr>
<tr>
<td>Eating (kettle, toaster)</td>
<td></td>
<td>250</td>
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<tr>
<td>Lights</td>
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<td></td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Total EQ + LIGHT in the house [W]</th>
<th>545</th>
<th>318</th>
<th>477</th>
<th>586</th>
<th>1818</th>
<th>218</th>
<th>3472</th>
<th>7.4</th>
<th>kWh/5 wd</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant value 3.5W/m2</td>
<td>13.4</td>
<td>kWh/ wd</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Difference</td>
<td>5.9</td>
<td>kWh/ wd</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Total EQUIP and LIGHT</td>
<td>29.6</td>
<td>kWh/5 wd</td>
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</tr>
</tbody>
</table>


# Detailed schedule of internal heat gains for weekend day

## Heat emitted from people (W/weekend)

<table>
<thead>
<tr>
<th></th>
<th>2560</th>
<th>320</th>
<th>1280</th>
<th>320</th>
<th>960</th>
<th>320</th>
<th>320</th>
<th>1280</th>
<th>7.68</th>
</tr>
</thead>
<tbody>
<tr>
<td>HG people</td>
<td>[kWh/weekend]</td>
<td></td>
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<td></td>
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<td></td>
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</tbody>
</table>

### Weekend (weekend)

<table>
<thead>
<tr>
<th>Room</th>
<th>23:00-07:00</th>
<th>07:00-08:00</th>
<th>08:00-12:00</th>
<th>12:00-13:00</th>
<th>13:00-14:00</th>
<th>14:00-17:00</th>
<th>17:00-18:00</th>
<th>18:00-19:00</th>
<th>19:00-23:00</th>
</tr>
</thead>
<tbody>
<tr>
<td>room 11.6</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>room 11.9</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Tech. room 9.1</td>
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<tr>
<td>Entrance 4.1</td>
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<td></td>
</tr>
<tr>
<td>Room 9.1</td>
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<td></td>
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</tr>
<tr>
<td>Bedroom 15.0</td>
<td>2</td>
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<td></td>
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</tr>
<tr>
<td>Kitchen 9.4</td>
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</tr>
<tr>
<td>Bathroom 1</td>
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<td></td>
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<tr>
<td>Corridor</td>
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</tr>
<tr>
<td>Living room 32.0</td>
<td>2</td>
<td>1</td>
<td>2</td>
<td>1</td>
<td>4</td>
<td></td>
<td></td>
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<tr>
<td>Dining room 20.1</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
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</tr>
</tbody>
</table>

### Time for the activity (hours)

<table>
<thead>
<tr>
<th>Room</th>
<th>8</th>
<th>1</th>
<th>4</th>
<th>1</th>
<th>1</th>
<th>3</th>
<th>1</th>
<th>1</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total</td>
<td>24</td>
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<td></td>
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<td></td>
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</tbody>
</table>

### Heat emitted (W)

<table>
<thead>
<tr>
<th>Activity</th>
<th>Weekend (weekend)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fridg + IHEU</td>
<td>68</td>
</tr>
<tr>
<td>Washing machine</td>
<td>250</td>
</tr>
<tr>
<td>Tumble dryer</td>
<td>1850</td>
</tr>
<tr>
<td>Dish washer</td>
<td>250</td>
</tr>
<tr>
<td>Cooking</td>
<td>1100 1100</td>
</tr>
<tr>
<td>TV</td>
<td>200 200 200 200</td>
</tr>
<tr>
<td>Computer</td>
<td>200 200 200 200</td>
</tr>
<tr>
<td>Eating (kettle, toaster)</td>
<td>250</td>
</tr>
<tr>
<td>Lights</td>
<td>16.7 250 150 150</td>
</tr>
</tbody>
</table>

### Total EQ + LIGHT in the house (W)

<table>
<thead>
<tr>
<th>Weekend (weekend)</th>
</tr>
</thead>
<tbody>
<tr>
<td>545 318 2872 3418 68 1454 1818 218 2672 13.4 kWh/weekend</td>
</tr>
</tbody>
</table>

### Constant value (3.5 W/m²)

<table>
<thead>
<tr>
<th>Weekend (weekend)</th>
</tr>
</thead>
<tbody>
<tr>
<td>13.4 kWh/weekend</td>
</tr>
</tbody>
</table>

### Difference

<table>
<thead>
<tr>
<th>Weekend (weekend)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.0 kWh/weekend</td>
</tr>
</tbody>
</table>

### Total EQUIP and LIGHT

<table>
<thead>
<tr>
<th>Weekend (weekend)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.1 kWh/2 weeks</td>
</tr>
</tbody>
</table>