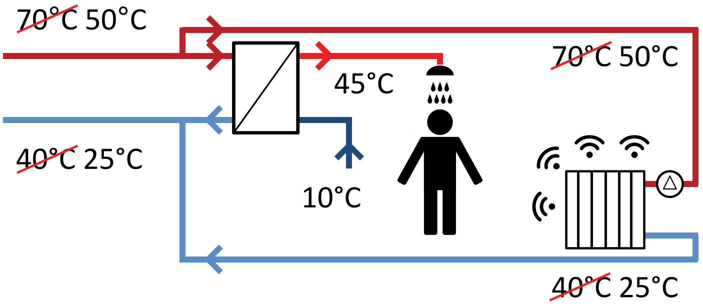
Heating and Domestic Hot Water Systems in Buildings Supplied by Low-Temperature District Heating



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PhD Thesis

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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, results and discussion and specific conclusions, but they share a common introduction, hypothesis, general 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.

- 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. in Energy 2012, vol. 41(1), p. 392-400.

The first paper argues that the supply service pipe 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 service pipes and whole network while making good use of the “waste” heat to take the chill off the floor in bathrooms.

- 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, Brand M, Dalla Rosa A, Svendsen S. in Energy 2014, vol. 67, p. 256-267.

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 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. in Energy 2013, vol. 62, p. 311-319.

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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 Wollerstrand 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.

One of my biggest thanks is to my colleague Alessandro Dalla Rosa for being always there to share matters of research and everyday life issues as well, and Jakub Kolařík for reading through and commenting on this thesis. And I also want to acknowledge all the staff from the section of Building Physics and Services for providing a friendly and inspiring environment. Special thanks to Diana Lauritsen and other Danish colleagues for having the patience to talk to me in Danish and thus helping me to learn the language.

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 Lída, 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 utilities. 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 utility, 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 5 W/m^2 to 4.2 W/m^2, 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 and reflecting reduced internal heat gains and higher desired indoor temperature leads to up to 56% greater heat demand than values suggested in the Danish national calculation tool Be10, and in 20% lower connection power than for an SH system dimensioned in accordance with DS 418. Furthermore the connection heat power is usually by DH utility increased by additional “safety factor” of 20-30%, resulting in total over-dimension for space heating up to 60%. Use of safety factors and 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

Fjernvarme (fjv.) systemer baseret på vedvarende energikilder er et væsentligt element i løsningen for at opnå fossil fri varmeforsyning til bygningssektoren frem mod 2035. For at kunne opnå dette mål, skal forsyningstemperaturen reduceres, for derved at tilgodese bygningernes fremtidige lavere energiforbrug og den øgede mængde af vedvarende energi som ved den lavere temperatur bliver til rådighed. Begge behov er imødekommet for det fornylig lancerede lav temperatur fjv. konceptet, som opererer ved forsyningstemperaturer på 50°C/25°C sammen med højt isolerede fjv. rør. Sammenlignet med traditionelle fjv. systemer (80°C/40°C), er distributions varmetabet reduceret til 1/4. Imidlertid medfører de reducerede temperaturer en række udfordringer for det varme brugsvand og for varme forsyningen. Dette både ift. slut-forbrugeren men også ift. varmeværket. Formålet med dette arbejde er at identificere, analysere og opstille anbefalinger og løsninger hertil.

Første del af arbejdet fokuserer på mulighederne for forsyning af varmt brugsvand uden øget risiko for legionalla bakterier, samt funktionalitet og performance af lav temperatur fjv. stationen.

Af Dansk Standard DS 439 fremgår at varmt brugsvand skal leveres indenfor acceptabel tid (komfort) og uden unødvendig temperatur variationer (komfort) samt uden øget risiko for vækst af legionella bakterier (hygiejne). Mht. komfort kræves en brugsvands 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 stiller krav til temperaturen ift. hygiejne. Temperaturkravet er således bestemt af komfortkravene i DS439, dvs. 45°C til rådighed for køkkenvasken.

Funktionalitet og performance for lav temperatur fjv. stationen er blevet analyseret vha. udviklede dynamiske Matlab-Simulink modeller og verificeret op imod laboratorier målinger. Disse viste, at der ved specificeret flow kunne opnås en brugsvandstemperatur på 47°C ved 50°C forsynings temperatur, med en tilhørende retur temperatur på 20°C. Resultaterne fra den dynamiske model viste, at det er muligt at komme fra system tomgang (ingen tapning over længere tid, 10 m delvis nedkølet stikledning, by-pass temperatur på 35°C og ingen varme behov) til 40°C brugsvandstemperatur på 11 sekunder. Ifølge DS 439 betragtes en passende ventetid på varmt brugsvand 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 styret 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 brugsvand, idet fjv. stationen bliver forsynet umiddelbart med varmt fjernvarmevand også uden for perioder med tapning af varmt brugsvand. Forskellige løsnings forslag for by-pass er blevet analyseret og valideret vha. IDA-ICE simulerings 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 reduceret fjernvarmenettets 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 sommeren 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 forsyningstemperatur 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 gulvvarmen for badeværelset i sommer perioden for derved at holde stikledningen varm og udnytte en del af den energi som ellers ville være gået tabt in 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 forsyningstemperatur 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. Energirenoveres med nye vinduer, hvilket er oplagt idet de eksisterende har udstået deres levetid, bliver den maksimale forsyningstemperatur reduceret til 62°C og varigheden for temperaturer over 50°C hhv. 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 forsyningstemperatur 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 varmetilskud 4.2 W/m^2 i stedet for 5,0 W/m^2 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 retvisende 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

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. in Energy 2012, vol. 41(1), p. 392-400.

- Marek Brand performed the full-scale measurements of district heating house substation at DTU; based on the model of heat exchanger and individual components provided by Danfoss build and calibrated numerical model in Matlab Simulink and wrote the whole article.
- Jan Eric Thorsen from Danfoss provided numerical models of components, supported further development on the numerical model and proof-read the article and suggested comments.
- Svend Svendsen proof-read the article and suggested comments.

Article 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, Brand M, Dalla Rosa A, Svendsen S. Svendsen S. in Energy 2014, vol. 67, p. 256-267 .

- Marek Brand developed technical solutions for comfort bathroom concept; performed detailed simulation in IDA-ICE and wrote the article.
- Alessandro Dalla Rosa wrote part "Traditional external bypass in a in-house substation" and provided code for calculation of bypass performance needed as an input data for simulations of bypass use in bathroom floor heating. Moreover based on the results from building simulations, provided by Marek Brand, performed simulation of the influence of bypass use in the floor heating on the district heating network in software Termis and partly wrote chapter "effect of comfort bathroom on heat production and distribution".
- Svend Svendsen developed the original concept of comfort bathroom which was explored by Marek Brand with help of Alessandro Dalla Rosa.

Article III

Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment, Brand M, Svendsen S. in Energy 2013, vol. 62, p. 311-319.

- Marek Brand performed all work.
- Svend Svendsen proof-read the article and suggested comments.

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

- A Direct Heat Exchanger Unit Used for Domestic Hot Water Supply in a Single-Family House Supplied by Low Energy District Heating. Brand M, Thorsen J E, Svendsen S, Christiansen CH. In proceedings of 12th International Symposium on District Heating and Cooling. 5-7 September 2010. Tallinn, Estonia
- Performance of Low Temperature District Heating. Brand M, Dalla Rosa A, Svendsen S. In proceedings of IEA ANNEX 49 conference "The Future for Sustainable Built Environments with High Performance Energy Systems". 19 - 21 October 2010. Munich, Germany
- Experiences on low-temperature district heating in Lystrup – DENMARK. Thorsen J E, Christiansen C H, Brand M, Olsen P K, Larsen C T. In proceedings of International Conference on District Energy. March, 2011. Portoroz, Slovenia
- Energy-Efficient and Cost-Effective Use of District Heating Bypass for Improving Thermal Comfort in Bathrooms in Low-Energy Buildings, Dalla Rosa A, Brand M, Svendsen S. In proceedings of 13th International Symposium on DHC. September 3-4, 2012. Copenhagen, Denmark
- Space heating in district heating-connected low-energy buildings. Lauenburg P, Brand M, Wollerstrand J. In proceedings of 13th International Symposium on DHC. September 3-4, 2012. Copenhagen, Denmark
- Optimal Space Heating System for Low-Energy Single-Family House Supplied by Low-Temperature District Heating. Brand M, Lauenburg P, Wollerstrand J, Zboril V. In proceedings of PassivHus Norden 2012.
- Results and experiences from a 2-year study with measurements on a new low-temperature district heating system for low-energy buildings. Christiansen C H, Dalla Rosa A, Brand M, Olsen P K. In proceedings of 13th International Symposium on DHC. September 3-4, 2012. Copenhagen, Denmark

RESEARCH PROJECTS

- EUDP 2008 "CO_2-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
- EUDP 2010 "Danish Energy Agency, "EUDP 2010 Low-temperature district heating in existing buildings"
  - 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
- IEA DHC Annex X – contribution to chapters 3, 4 and 5 of the report “Toward 4th Generation District Heating: Experiences with and Potential of Low-Temperature District Heating”

NOMENCLATURE

List of abbreviations

CB	comfort bathroom
ECL	electronic controller from Danfoss
DHWC	DHW circulation
DHWSU	DHW storage unit
DH	district heating
DHSU	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_{bypass}	[°C]	bypass set point temperature
T_{floor}	[°C]	floor temperature
T_{op}	[°C]	operative temperature
T_{ret}	[°C]	return temperature
T_{soil}	[°C]	temperature of soil
ΔT_{DB}	[°C]	Deadband
P_{max}	[kW]	maximum heating power
TRW	[°C]	weighted average return temperature

TABLE OF CONTENTS

1 INTRODUCTION

To reduce CO_2 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).

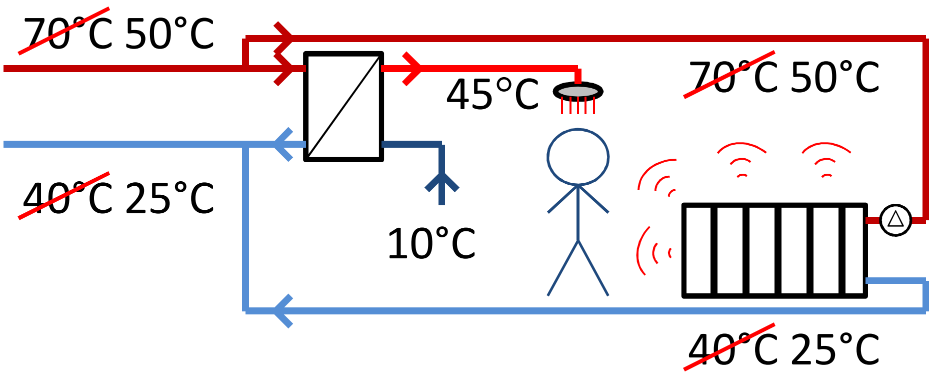


Figure 1.1 – Concept of low-temperature district heating

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 and 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 heating 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 and thus in comparison with traditional bypass solutions 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.

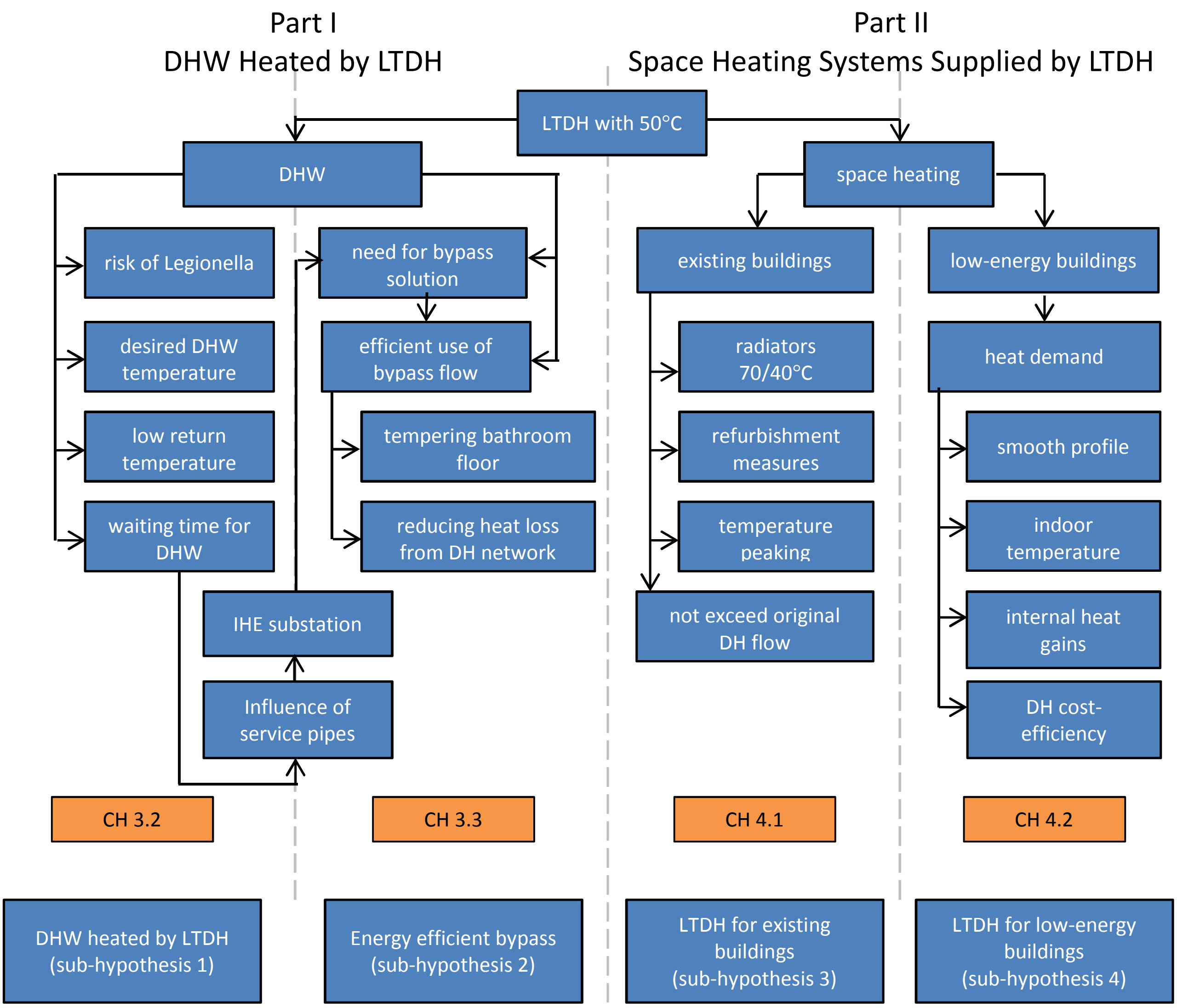


Figure 1.2 – Relation between main hypothesis and sub-hypothesis

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 CO_2 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 CO_2 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.

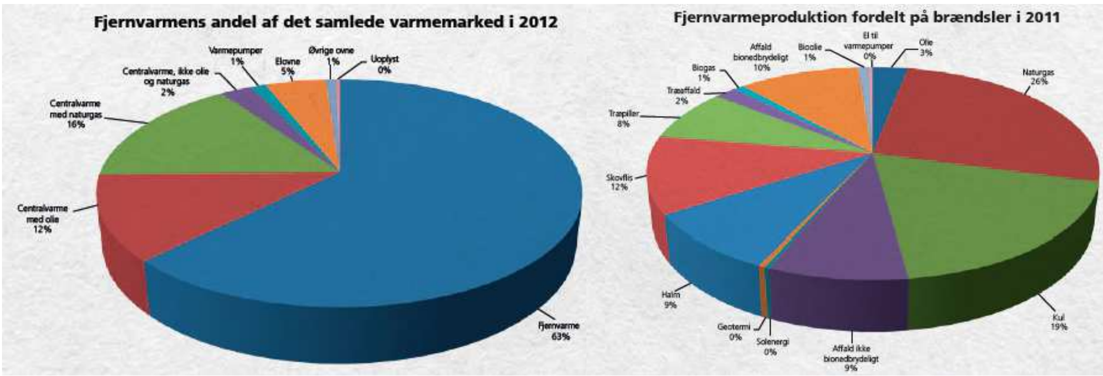


Figure 2.1 – left: Share of DH in heat delivery in Denmark 2012; right: Share of heat sources in DH for 2011 [18]

In 2010, the average DH supply (T_{sup}) and return (T_{ret}) temperatures in Denmark were:

- heating season T_{sup} 78.7°C and T_{ret} 41.4°C
- non-heating season T_{sup} 73.3°C and T_{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.

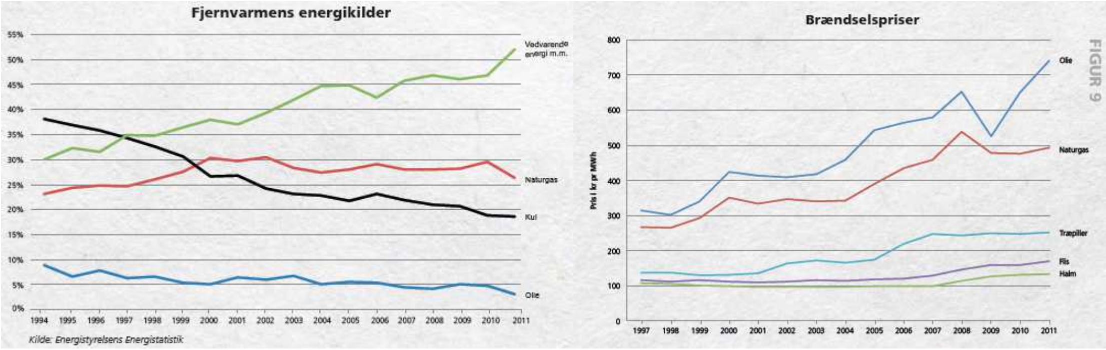


Figure 2.2 – left: Development of heat source shares in DH in the period 1994-2012 [19] *(green – RES, red – natural gas, black – coal, blue – oil)*; right: Development of fuel prices for DH in the period 1997-2011 [18] *(blue – oil, red – natural gas, azure – wood pellets, violet – wood chips, green - straw)*

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 to 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 [31]. 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.

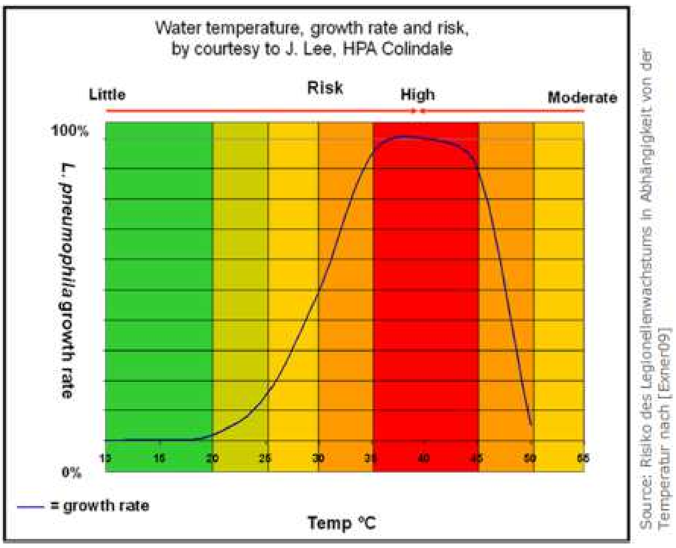


Figure 3.1 – Risk of Legionella proliferation related to the DHW temperature [32]

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 [33] 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 [34], 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:

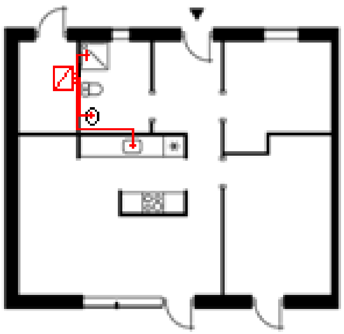
- Minimum DHW temperature of 45°C, based on the comfort requirements

- 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.



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

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, connected together with pipes and fittings, controlling the heating of the DHW to the desired temperature.

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), typical design heating power 32.3 kW [22]
- Substations with a DHW storage tank, design heating power depends on size of the DHW storage tank

A substation based on the instantaneous principle of DHW heating produces DHW only when needed (see Figure 3.3 left), 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, often controlled by the outdoor temperature and 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.

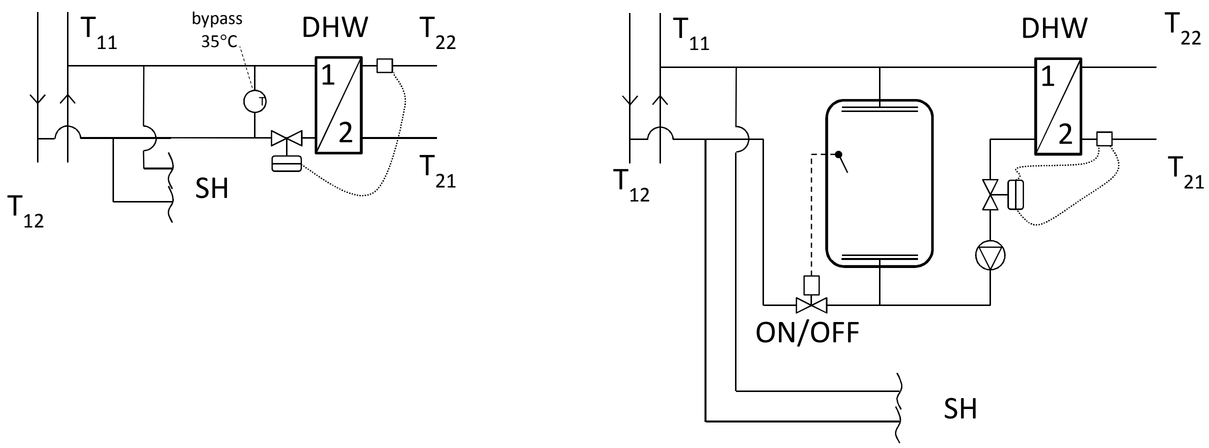


Figure 3.3 – Low-temperature DH substations; left: instantaneous DHW principle, i.e. IHEU [35], right: storage tank for DH water, i.e. DHSU [25], [36]

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 [37], 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)

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 [35] or Akva Les II S [38].

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 (period without DHW tapping), e.g. PTC2+P [39]. 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 (pipe connecting the DH pipe in the street with the DH substation in the building), 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 across the control valve and thus enables the control valve to operate on whole stroke (lift) giving the full control range.

DHW storage tank

To follow the German standard DVGW 551 [33], 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 right) 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 supply service pipe 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 supply service pipe, 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.

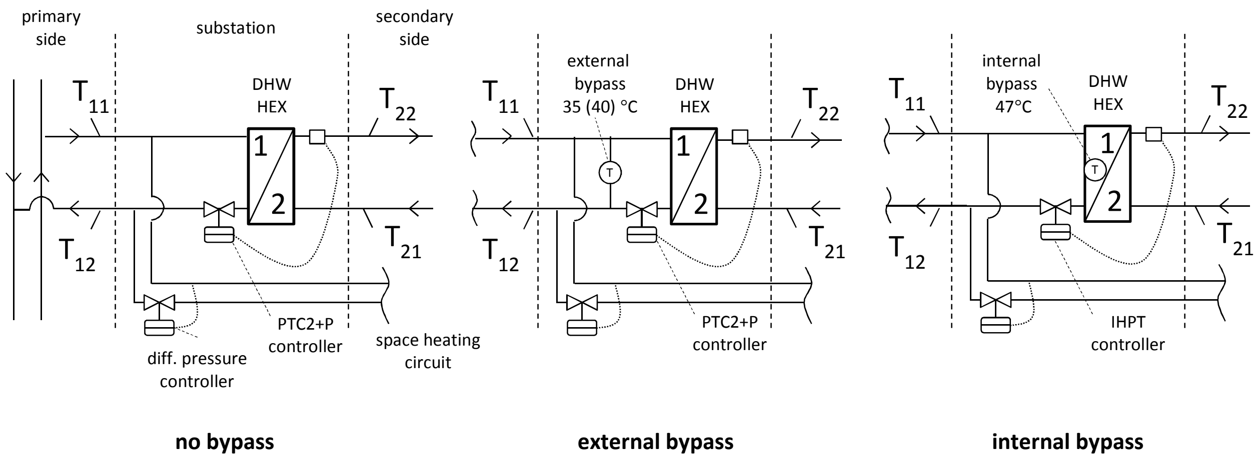


Figure 3.5 – Various bypass strategies for IHEU; left: no bypass; middle: external bypass (cold HEX) with set-point temperature 35°C; right: internal bypass (warm HEX) with set-point temperature 47°C (defined by DHW set-point 45°C)

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 required by missing subsequent customers. 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 reduction in efficiency of the HEX developing in time 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. There is no flow and the supply service pipe (SP) 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 [40], 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.

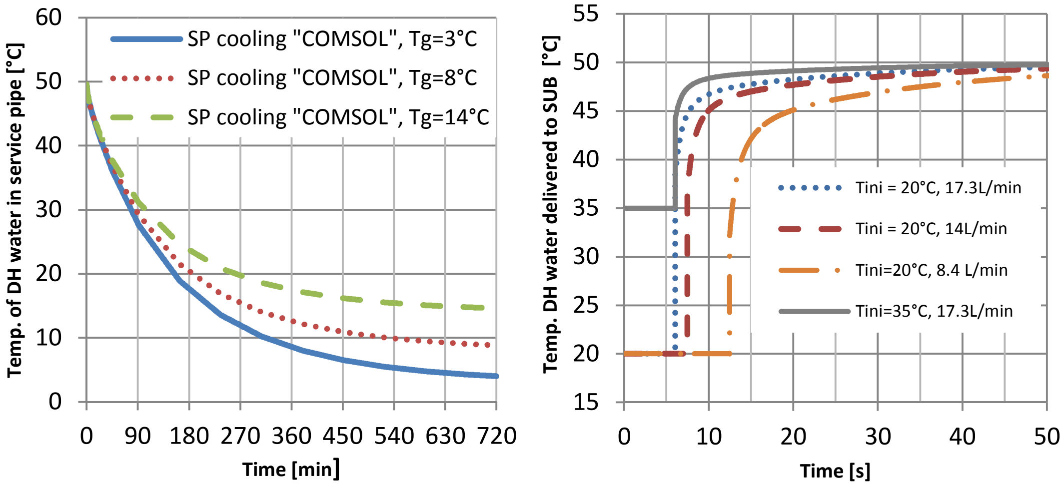


Figure 3.6 – left: Cooling down of DH water standing in an AluFlex 20/20/110 service pipe during idling. The initial temperature of water in pipe is 50°C, and the initial temperature of the insulation is 15°C ; right: Effect of the thermal capacity of an AluFlex 20/20/110 service pipe during reheating of the pipe

After 180 minutes, the customer opens the DHW tap again and "fresh" DH water at 50°C starts to flow to the 10 m long 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. It means that 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 (defined by DHW controller) 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), showing results based on code of Dalla Rosa reported in [40]. During this period the substation will be supplied with DH water cooled by standing in the supply 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 [39] the initial DH flow rate is 17.3 L/min and 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 in case of low-temperature DH 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], [41]. 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.

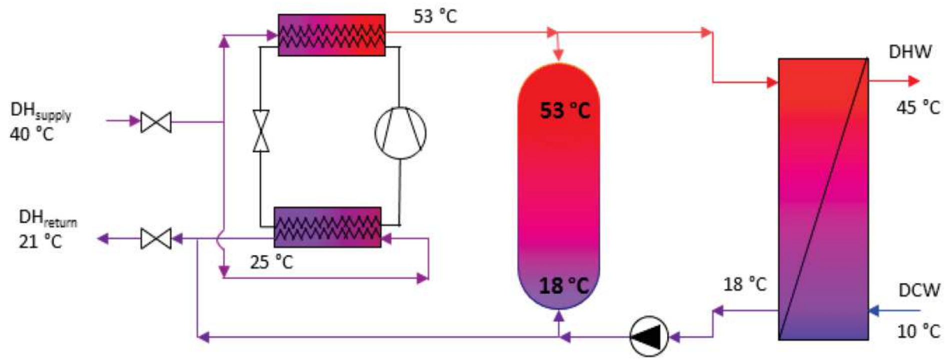


Figure 3.7 – Principle of low-temperature DH substation combined with micro heat pump [41].

During the development phase of the substation, few variants were analysed [41]. 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. As the solution with the best performance was evaluated 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 [33]. 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. The low-temperature DH substation could be either instantaneous heat exchanger unit (IHEU) [35] or district heating storage unit (DHSU) [36], [25], depending mainly on the requirements of the DH utility.

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 the volume of DHW is more than 3 L and 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, easily reaching up to 16 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 [42] (see Figure 3.8 - right), where each apartment has own substation, and the DHW and SH pipes are in the individual flats 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.

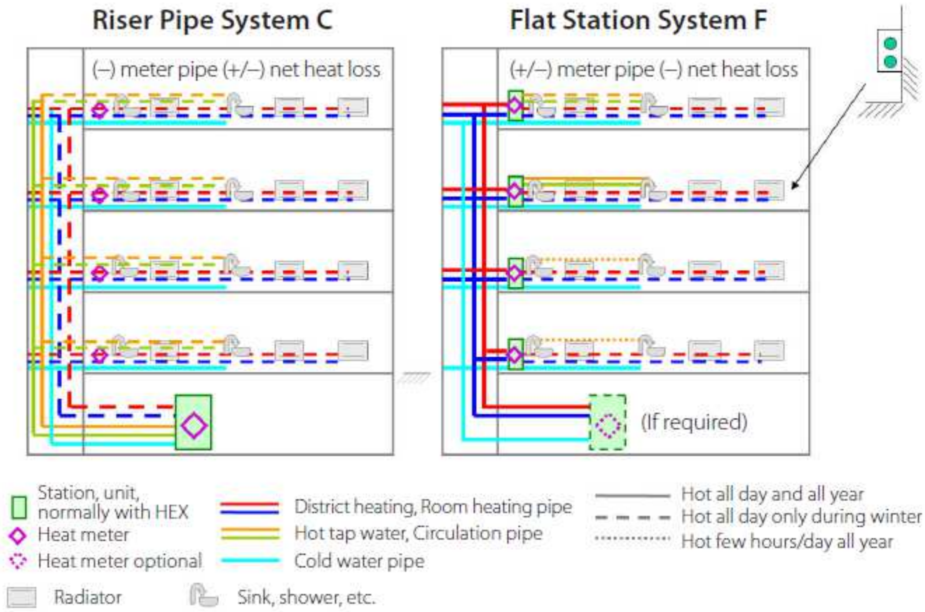


Figure 3.8 – Comparison of DHW systems in multi-storey building [42]; left – traditional system with vertical risers; right – flat station system

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 ultraviolet (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 [43]. 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.

As an alternative solution in the DHW systems with DHW circulation could be seen a thermal disinfection of circulating DHW with heat recovery (see Figure 3.9). Since the DHW can be with LTDH heated only approximately to 47°C, the additional source of heat is needed. This can be either small gas boiler or electric resistance heater, boosting the temperature of DHW up to required temperature level (position 1). The required temperature will depend on the time period the DHW will spent in the “reaction chamber” (position 2).

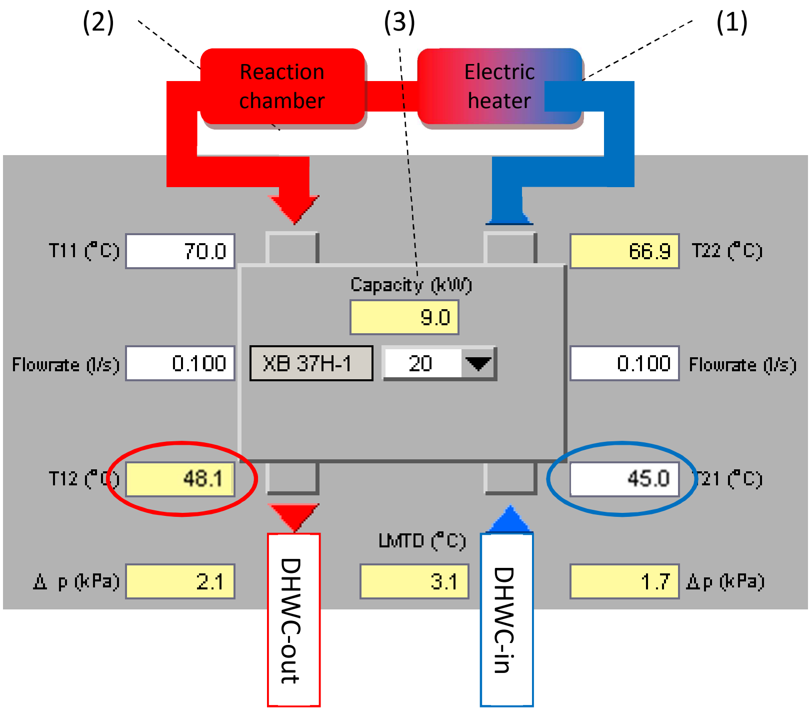


Figure 3.9 - Principle of thermal disinfection with heat recovery for LTDH. Calculation made by Danfoss HEXcalc software [44]

The higher the temperature the shorter the needed reaction time [26]. While for 70°C the reaction time to eliminate 99% of Legionella is 1 minute, for 60°C it is 5 minutes and for 50°C it is 80 minuts. After the thermal disinfection, the DHW will flow through the heat recovery (heat exchanger, position 3) where the disinfected DHW will be cooled down by 47°C warm DHW returning from the DHW recirculation.

However even small DHW circulation flow as 0.1 L/s, requires considerable amount of energy, i.e. 1.3 kWh/h, corresponding to 31 kWh/day and 11.4 MWh/a. Seen this value from the perspective of 800 kWh/a as energy needed for DHW heating for one person per year it corresponds to the annual energy demand for DHW for 14 people, having very high cost. Considering the price of electricity 2 DKK/kWh the cost for such solution is around 23000 DKK/year, while the expected annual running cost for UV disinfection technology is 3800 DKK (energy demand 42W). The solution with UV disinfection should be therefore preferred.

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 “Numerical modelling and experimental measurements for a low-temperature district heating substation for instantaneous preparation of DHW with respect to service pipes” [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 (IHEU) 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 [35] equipped with a PTC2+P controller for DHW was measured in well-defined conditions in the laboratories of the Technical University of Denmark. The experimental setup is shown in Figure 3.10.

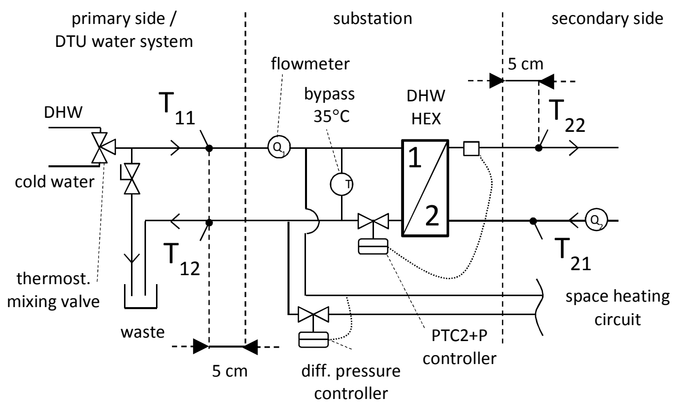


Figure 3.10 – Experimental setup for measurements of IHEU

We measured temperature of four water flows passing through the substation, flow rates on primary and secondary sides and air temperature in the laboratory. It was supply temperature T_{11} and return temperature T_{12} on primary side and the supply temperature T_{21} (cold domestic water) and return temperature T_{22} (DHW) on the secondary side. All temperatures were measured with previously calibrated thermocouples type T. Water flow rates Q were measured on both sides with traditional analogue flow meters. The substation was connected to the campus DHW system by thermostatic mixing valve, providing 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 combined few numerical models in one and use it for this purpose. The philosophy of the model can be seen in Figure 3.11.

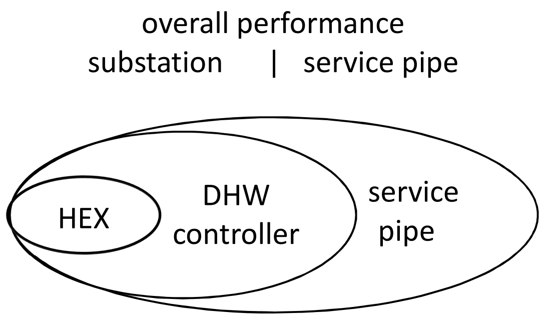


Figure 3.11 – Structure of model investigating waiting time for DHW in the DH substation without a storage/buffer tank

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 utilities.

The numerical model of the IHEU substation was developed in MATLAB and-Simulink [45] based on combination and update of earlier models of individual components 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 the time delays in the pipes caused by the transportation delay and their thermal capacity.

The model of the HEX is based on the model in Persson [46], 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 3.12).

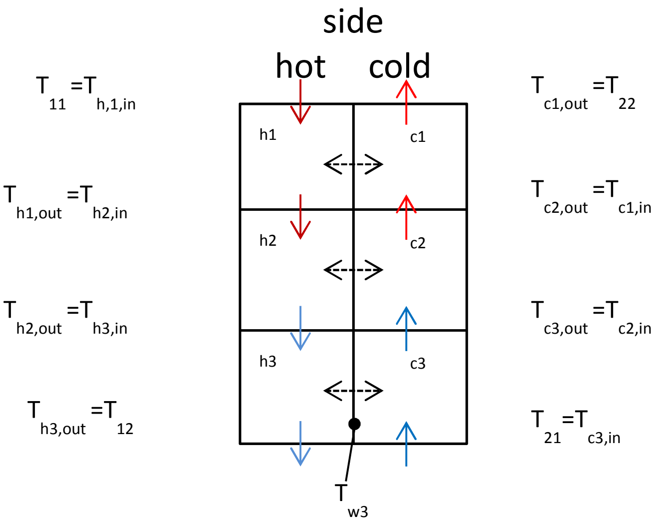


Figure 3.12 – Description of the numerical model of the HEX with three sections

The three sections model is considered to be accurate enough to model the overall performance of a HEX [46]. 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:

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

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

$$\frac{d}{d\tau}(T_{c1,out}) = \frac{1}{m_{c1} \cdot c_{p,c}} \cdot [\dot{m}_c \cdot c_{p,c} \cdot (T_{c1,in} - T_{c1,out}) - \alpha_{c,w} \cdot A_c \cdot (\frac{T_{c1,in} + T_{c1,out}}{2} - T_{w1})] \quad (2)$$

Energy balance equation for hot medium:

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

Energy balance equation for HEX plates:

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

where:

A_c	total plate area on the cold side in one section	[m^2]
$c_{p,c}$	specific heat capacity of water	[J/(kg.K)]
m_c	mass of water in the section	[kg]
$\dot{m}_c$	mass flow of water through the section	[kg/s]
m_w	total mass of plates in the section	[kg]
Pr	Prandtl number	[-]
Re	Reynolds number	[-]

$T_{c,in}$	temperature of water coming into the section	[°C]
$T_{c,out}$	temperature of water leaving the section	[°C]
T_w	average wall temperature in the section	[°C]
$\alpha_{c,w}$	convective heat transfer coefficient	[W/(m^2.K)]
δ_p	distance between individual HEX plates	[m]
λ_w	thermal conductivity of water	[W/(m.K)]

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 original model was updated with input parameters representing HEX XB37H used in the tested substation.

The adopted model of PTC2+P DHW controller (see Figure 3.13) was modelled as a numerical description of all individual mechanical parts (springs, bellow elements, valves, friction resistance, etc.) with the parameters given by the manufacturer.

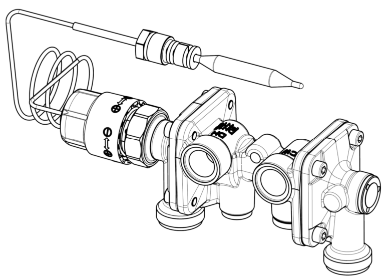


Figure 3.13 – PTC2+P DHW controller [39], courtesy of Danfoss A/S.

The model is property of manufacturer and the manufacturer doesn't want to present the model in more details. Part of the implementation of IHEU to the MATLAB Simulink can be seen in Figure 3.14.

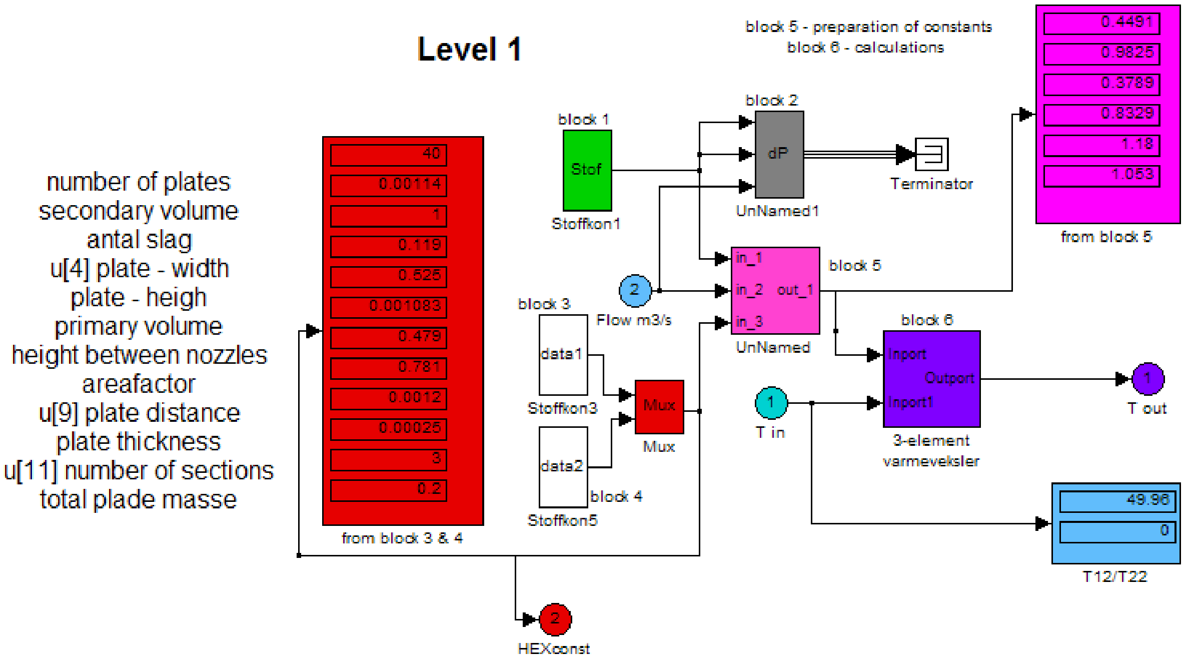


Figure 3.14 –Part of the Simulink model of the IHEU substation – detail on HEX model

The overall model of IHEU was successfully verified with the data from laboratory measurements, under both steady-state and dynamic conditions (see Figure 3.15 and Table 3-1). Figure 3.15 shows the results obtained from the measurements of recovery time for the IHEU and compare 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 (compare curves T_{22}sim with T_{22}meas and T_{12}sim with T_{12}meas) and can therefore be used. Numerical values are reported in Table 3-1: case M for measurements and case 0 for the simulation.

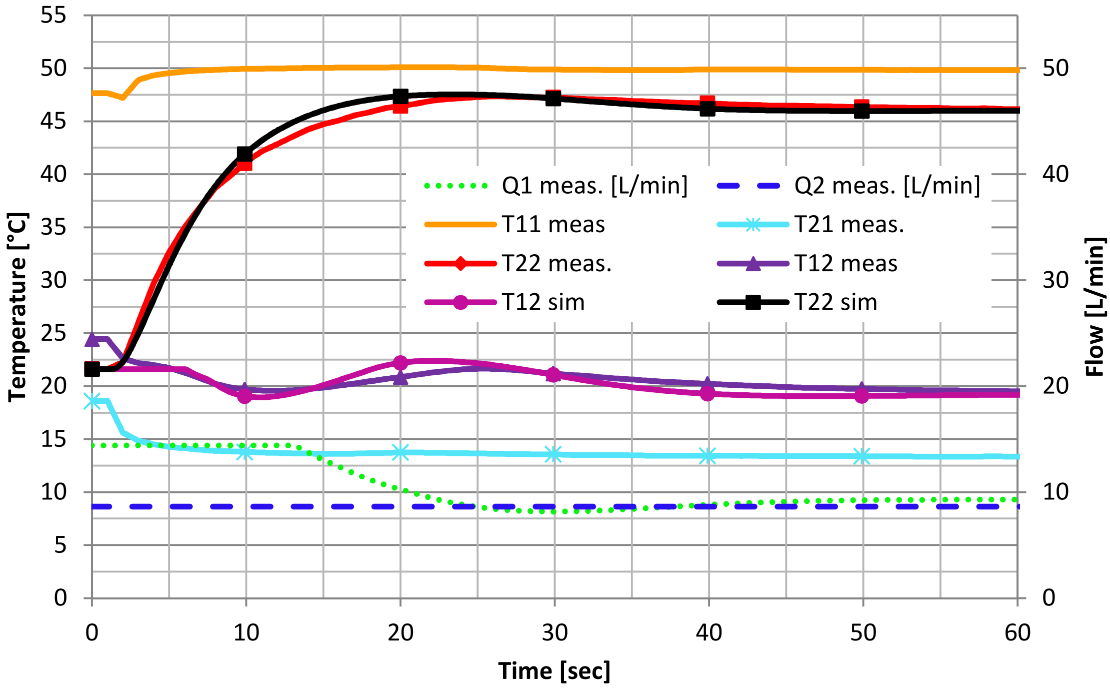


Figure 3.15 – Comparison of experimentally measured “meas” and numerically simulated “sim” temperatures T22 and T12 for an IHEU equipped with a PTC2+P controller for the same input data

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 [47], controlled on the basis of the temperature of the passing fluid, and installed just after the inlet to the IHEU (see Figure 3.18a). To ensure the stability of the control process, the valve has a neutral zone (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. The neutral zone therefore results in intermittent operation of the valve also called “pulse” mode.

By recurrent application of the cooling curves for cooling of DH water in service pipes combined with the numerical code for the dynamic heat transfer in the service pipes [40], both developed by Dalla Rosa (already described in section 3.1.3), we obtained a temperature profile along the 10 m long AluFlex 20/20/110 service pipe (Figure 3.16) 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.

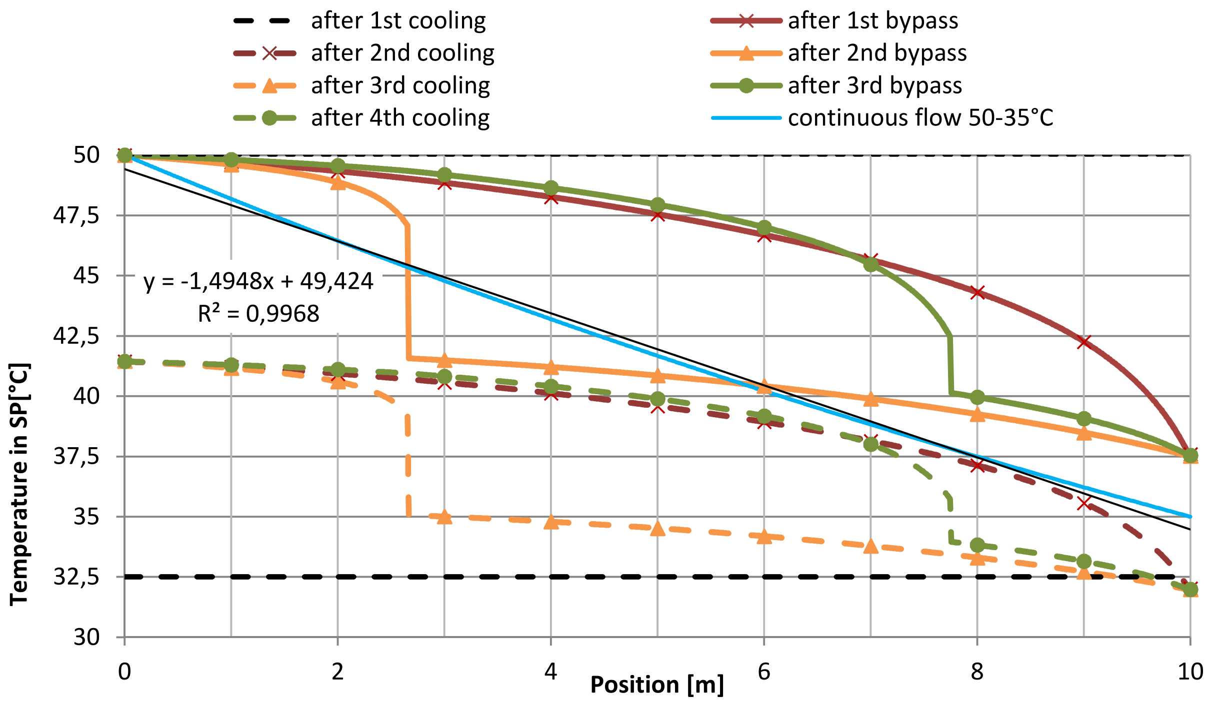


Figure 3.16 - Temperature profile along a 10 m long SP AluFlex 20/20/110 for an IHEU equipped with an external bypass in traditional and continual operation modes. Traditional bypass: set-point 35°C, deadband ±2.5°C, bypass flow rate 3 L/min, T_{ground}=14°C. Continual bypass: temperature drop 50-35°C, flow rate 0.024 L/min, T_{ground}=14 C.

The figure shows development of the temperature profile along 10 m long supply service pipe for few time slices. It can be seen that the temperature of DH water standing in the pipe varies depending on how many times the bypass valve has opened since the previous tapping. First, the pipe is full of 50°C hot water as result of just finished DHW tapping. The water in the pipe cools down uniformly to the temperature of 32.5°C in 82 minutes (curve “after first cooling”), which opens the external bypass and DH water starts to flow into the substation. Considering the fact that the service pipe was prior to bypass valve opening at 32.5°C and now the water with 50°C started to flow results is cooling of the bypass water front. The bypass flow stop after the water front with temperature 37.5°C reaches the bypass valve and stops the flow (curve “after first bypass”). Then the water in the service pipe cools down again (curve “after 2nd cooling”) and the procedure is repeated.

Knowledge on DH water temperature in the service pipe in every time makes possible to use water temperature history as an input data for the IHEU model taking in to the consideration also influence of service pipe and external bypass.

Figure 3.16 also shows the temperature profile for a theoretical bypass controller providing continuous flow without the deadband. The temperature profile along the pipe in steady state conditions was obtained from equation [49]

$$T_{d(x)} = T_g + (T_u - T_g)\exp\left(-\frac{U.x}{\dot{m}.c_p}\right) \quad (5)$$

where $T_{d(x)}$ represents temperature of bypassed water at the location in question, T_u temperature of bypassed water at the beginning of the service pipe and T_g represents ground temperature, x represents x-coordinate of the investigated point measured from the beginning of service pipe [m], $\dot{m}$ is the mass flow rate [kg/s] of bypassed water, U is thermal transmittance of the pipe [W/(m.K)], c_p is specific heat of water [J/(kg.K)].

First the needed mass flow $\dot{m}$ was calculated for the boundary conditions T_d=35°C, T_u=50°C, T_g=8°C, x=10 m, U=0.089 W/(m.K) and afterwards the mass flow was used to calculate the temperature profile along the pipe (shown in Figure 3.16).

The results of Dalla Rosa et al. [50] show that continuous bypass without deadband bypasses only needed amount of water to keep the inlet to the substation on 35°C, meaning roughly 30% less water volume than traditional external bypass with dead band, and thus saving up to 30% 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 including the service pipe was simulated for the following cases:

1) No bypass – water in supply service pipe cools down during non-heating season
2) Traditional external bypass controlled by FJVR valve and 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 during non-heating period 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 hypothetic thermostatic controller without the deadband, keeping inlet to substation with continuous flow on 35°C, which was expected to result in lower heat loss from the service pipe.

3.2.2 Results and Discussion

Regarding the risk of Legionella and comfort for occupants 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

The laboratory measurements confirmed that the tested low-temperature DH substation with instantaneous heat exchanger principle can from 50°C DH water produce DHW with 47°C, while cool down DH water to 20°C.

Table 3-1 shows the recovery times of the IHEU for DHW with various temperature and different boundary conditions as results from the numerical model. The first two lines compares recovery time for experimental measurements with numerical model with the same input conditions and it documents that the model describes with good agreement performance of the IHEU. The lower part of Table 3-1 reports the recovery time for the IHEU with the influence of the service pipe (SP), while the upper part excludes this influence.

Table 3-1 – Recovery time for IHEU for various boundary conditions

	Case	Description	recovery time of IHEU [s] to produce DHW with temperature				
			30°C	35°C	40°C	42°C	45°C
Without serivce pipe	M	experimental measurement	4.1	6.0	9.1	10.8	15.8
	0	verification of num. model	4.6	6.3	8.6	10.0	13.1
	1	pure recovery time, T11=50°C, HEX20	3.7	4.9	6.5	7.3	9.4
	2	pure recovery time, T11=50°C, HEX40	4.1	5.5	7.4	8.55	10.9
With service pipe	3	continual bypass 50°C - 35°C	5.7	7.7	10	11.3	14.1
	4	ext. bypass before 2nd bypass flow	5.9	8.7	11.0	12.4	15.5
	5	ext. bypass before 3rd bypass flow	7.2	9.5	11.9	13.2	16.4
	6	without bypass – water in SP on 35°C	7.2	9.8	12.3	13.6	16.9
	7	without bypass – water in SP on 20°C	11.0	12.7	15.2	16.7	20.9

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 to 6.5 s 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 the price of the HEX is reduced.

Taking into account influence of 10 m long service pipe (Aluflex 20/20/110), filled with DH water cooled to 20°C by a long period of idling (case 7), 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 (case 4 and 5), the recovery time decreases by 3-4 s, depending on the bypass phase (see Figure 3.16). Using continuous flow with a set-point temperature of 35°C (case 3) will decrease the recovery time by another 1 s.

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 has also 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 of DHW waiting times with traditional DH substations is not available.

For tested substation, 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 the components 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 [51]. 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.

3.2.3 Conclusions

- Based on the literature study it is concluded that in the DHW systems with overall volume below 3L is no requirement of minimal DHW temperature regarding increased risk of Legionella. Therefore the minimal DHW temperature is defined by requirements for comfort, i.e. 45°C.

- Laboratory test of low-temperature DH substation confirmed that the substation can produce DHW with 47°C from 50°C warm DH while cooling the DH water to 20°C. These results therefore justify temperature level of 50°C as a minimal supply temperature of low-temperature DH for the buildings without additional on-site heating source for DHW.

- Developed numerical model of low-temperature DH substation with instantaneous type of DHW production, including influence of service pipes was successfully validated with laboratory measurements and documents importance of accounting the effect from service pipes to the overall evaluation of the substation performance.

- To prevent increase of the waiting time for DHW, the supply service pipe should be kept warm by bypassing small flow of DH water. With external thermostatic bypass set to 35°C, the 10 s waiting time for 50°C DHW recommended by Danish DHW standard is doubled, but DHW with sufficient temperature 40°C is prepared in 11 s and just slightly exceeding the required value. Therefore it is suggested to revise the required waiting time for DHW while consider reduction of DHW temperature and DH supply temperature.

- The waiting time for DHW can be further slightly reduced by installation of continuous bypass, operating in comparison to the traditional thermostatic bypass without deadband and thus keeping the inlet to the substation always on 35°C

- Developed numerical model can be further used for optimisation of whole low-temperature DH concept and the DH substations as well.

3.3 Cost-efficient Use of Bypassed DH Water in Bathroom Floor Heating

The results from the numerical simulations of a low-temperature DH substation with instantaneous principle of DHW heating (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 thus increasing the 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 and possibly improving efficiency of the heat sources while giving the sensation of a warm floor to the users. This concept is referred to as the "**comfort bathroom**" (CB).

This chapter gives summary of the second ISI paper "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" [29] where all details can be found.

3.3.1 Methods

3.3.1.1 Modelling of external bypass

To investigate the realistic performance of a bypass flow redirected to the bathroom floor heating, it was necessary to model the bypass solution first.

The bypass flow needed to keep the inlet to substation on desired temperature (in our case 35°C) is changing based on the location of the customer in the DH network. The closer the customer is to the heat source, the higher the temperature of DH water on the beginning of the service pipe, because the water travelled shorter distance and thus cooled down less. To reflect this phenomenon we modelled performance of bypass for three different locations in the DH network. The locations were chosen as the locations with DH water temperature at the beginning of the service pipe 50°C, 40°C and 37.5°C during the bypass period and thus representing the customers located close, middle and far distance from the heat source. The example of such customers is shown in Figure 3.17, as result of Termis network simulations (described later in the text).

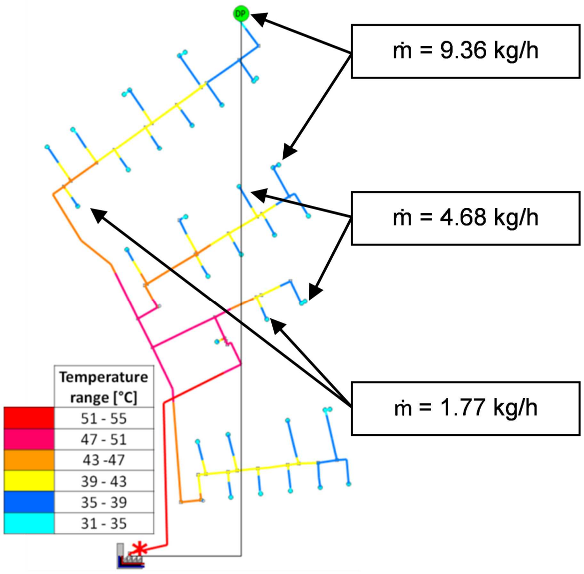


Figure 3.17 – Example of continuous bypass flow needed to keep inlet to substation on 35°C for the customer in close, middle and far location from the heating plant; network represents low-temperature district heating project in Lystrup, Denmark [7]

We used again numerical model developed by Dalla Rosa [40] to model traditional external thermostatic bypass valve FJVR sending the bypass flow in **pulses**, but we modelled also **continuous** bypass flow (see section 3.2.1. equation (5)), which can be in real realised by a needle valve, i.e. a valve with precisely adjustable flow. The continuous flow provided by a needle valve was based on the previous results expected to reduce volume of bypass water (and thus also the heat loss) needed to keep the inlet to substation on 35°C by 30%. We assumed 10 m long service pipe Aluflex 20/20/110, ground temperature 8°C and bypass set-point temperature 35°C.

The results of the numerical simulations showed, that the continuous flow needed to keep the inlet to substation on 35°C is for the customers located in close, middle and far distance from the heat plant 1.77 kg/s, 4.68 kg/s and 9.36 kg/s respectively. Pulse bypass was modelled with bypass flow 0.5 kg/min and dead band of 3°C, but only for customers located in close and in middle distance to the heat plant. The results show that the pulse (intermittent) bypass operates in both locations every 15 minutes. For the close location is opened for approximately 70 s in each cycle and bypasses 0.65L, which corresponds to 2.6 kg/h and for the middle location is the bypass flow opened for approximately 213 s and bypasses in each cycle roughly 1.8 kg, i.e. 7.1 kg/h. The results are therefore in accordance with the expected results that the volume of bypassed water is for thermostatic bypass with pulse flow 50% higher than for continuous bypass flow controlled by the needle valve without the deadband.

3.3.1.2 Technical solutions for Comfort Bathroom

Various technical proposals for redirecting and controlling the bypass flow for bathroom floor heating have been developed (see Figure 3.18). Some of the solutions were implemented with a traditional thermostatic bypass valve, type FJVR [47] supplying a floor heating loop with the intermittent water "pulses" and others with a needle valve providing "continuous flow". 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.

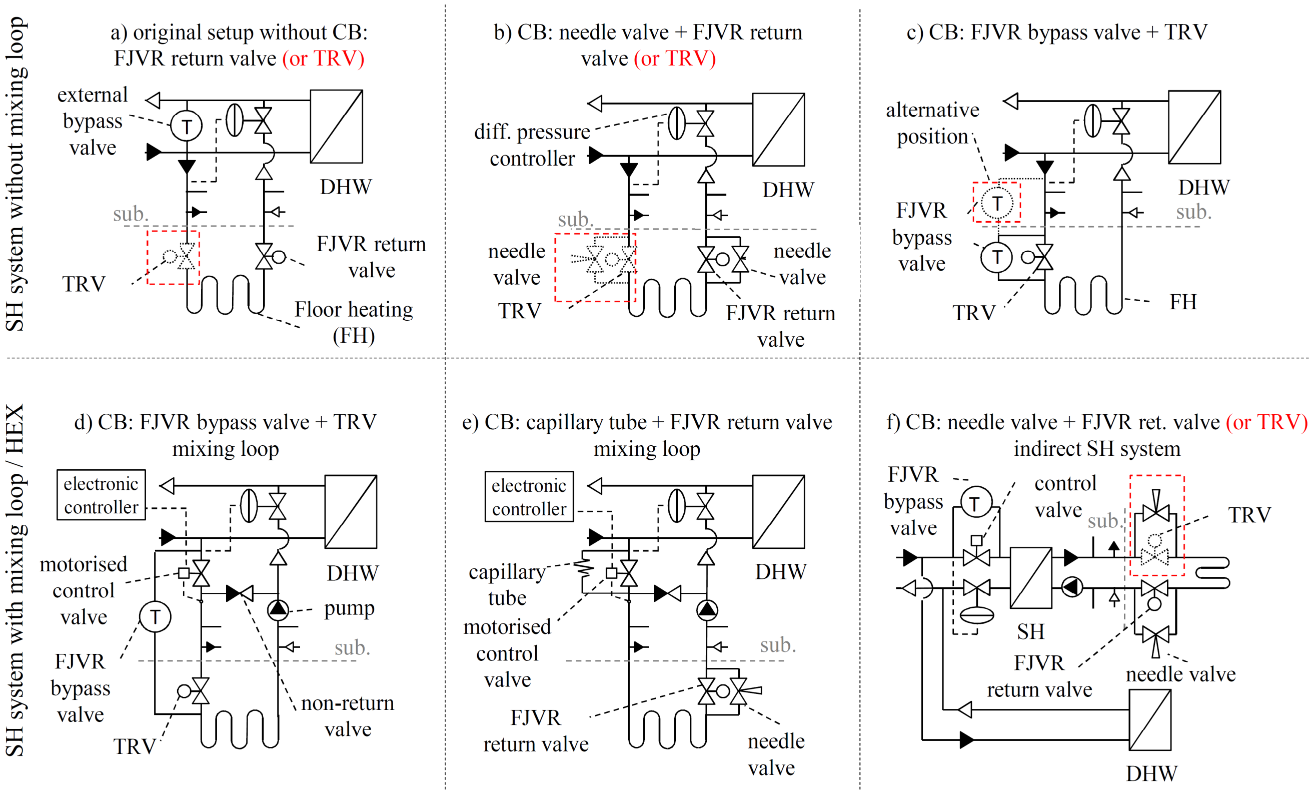


Figure 3.18 – Technical solution for CB implementation. Direct SH system without mixing loop: a) reference case without CB, with traditional external bypass; b) CB realised with a needle valve (installed in parallel to TRV valve on supply pipe or in parallel to FJVR valve on the return pipe of FH loop); c) CB realised with a FJVR bypass valve. Direct SH system with mixing loop: d) CB realised with a FJVR bypass valve; e) CB realised with a needle valve and capillary tube. Indirect SH system: f) CB realised with FJVR bypass valve.

Reference case without CB

Figure 3.18a show original connection scheme of direct space heating system with floor heating (FH) in the bathroom, controlled either on supply pipe by traditional thermostatic valve (TRV) [52], with the thermostatic head sensing operative temperature in the room or by the **thermostatic return valve** [47] installed in the technical room on the return pipe of the FH loop, usually operated with set-point temperature 25°C. The bypass solution is provided by traditional **thermostatic bypass valve** FJVR [47] (same as the thermostatic return valve, just in another position), redirecting the bypassed water back to the DH network.

CB controlled with needle valve

This solution is expected to reduce the volume of bypassed water by 30% in comparison to the traditional thermostatic bypass valve with deadband. CB concept is realised by installing the needle valve in parallel to the thermostatic return valve and thus keeping the constant bypass flow through the FH loop (see Figure 3.18b). For the winter periods when the bypass flow through the needle is not enough, the additional flow can pass through the thermostatic return valve. Similar solution can be applied also in the floor heating loop controlled by TRV, just on the supply pipe of the FH. The traditional external bypass valve is from the DH substation removed. The advantage of this solution is very fast and cheap installation for the solution with thermostatic return valve if installed in the technical room, but the drawback is lack of "automatic stop" of the bypass flow for the periods when the temperature of bypassed water exceed 35°C. This happens with every tapping of DHW, filling the service pipe with 50°C warm DH water. After this happens the bathroom FH is supplied with water with unnecessarily high temperature for approximately 45 minutes before the temperature in the service pipe drops back to 35°C (as can be seen in Figure 3.19).

<u>CB controlled with thermostatic bypass valve</u>

CB concept in FH loops controlled by TRV (see Figure 3.18c) can be easily realised by moving the thermostatic bypass valve from its original position into the new position, parallel to TRV valve. In this setup, the thermostatic bypass valve performs as in original solution, but the bypass water is cooled down in the bathroom FH. The thermostatic bypass valve can be installed either in parallel to the TRV valve in the bathroom, or in the technical room. The second solution is more precise regarding setting of required bypass set-point temperature thanks to its position in the substation, but on the other hand it requires installation of additional pipe between the DH substation and TRV controller. It should be mentioned that by using the thermostatic bypass valve to control the bypass flow, the advantage of 30% reduction of bypass flow is lost.

<u>SH systems with mixing loop (direct SH) or heat exchanger (indirect SH)</u>

Figure 3.18d-f shows installation of the CB concept to the substations with possibility of controlling the supply temperature to the SH loop. In Figure 3.18d is presented solution for directly connected SH system with mixing loop and bathroom heating controlled by TRV. In this case the CB flow is continuously bypassed through the thermostatic bypass valve, installed in parallel to the main control valve and TRV controlling the FH loop. Using of bypass flow in the FH controlled by the thermostatic return valve can be made similar (see Figure 3.18e), by bypassing the main control valve and installation of needle valve in parallel to the thermostatic return valve. However for bypassing the main control valve is not used thermostatic bypass valve, but capillary tube. Realization in the SH system with indirect connection is show in Figure 3.18f, again by installation of thermostatic bypass valve bypassing the main control valve and subsequent installation the needle valve in parallel to the FH controller.

3.3.1.3 Modelling of CB

Some of the solutions presented in Figure 3.18 were investigated on an example of 157m^2 single-family house built in accordance with Danish building requirements class 2015 [53], meaning that the annual energy demand, accounting for space heating, DHW heating, and operation of HVAC systems should be after accounting for primary energy factors below 37 kWh/(m^2.a). The house has ten rooms and two bathrooms (8.3 and 4.3 m^2) and the CB concept was installed in both of them (see Figure 4.4). The ventilation rate for the house was based on the BR10 requirements dimensioned to 216 m^3/h and the heat recovery in the ventilation system has 85% efficiency. The windows in the house were shaded with the external blinds (with g value of 0.14) and drawn when the solar irradiation exceeded 300 W/m^2. Moreover all windows were shaded with 0.5 m deep roof overhang. The venting of the house by opening the windows started when the indoor temperature exceeded 24°C and stopped when the temperature drop below 22°C. However the windows couldn't be open when the occupants were not at home, i.e. during working days between 8 a.m. and 3 p.m. The overall internal heat gains (people + equipment) in the whole house were modelled with value of 5 W/m^2, but the heat gains from the bathrooms were transferred to the living room and kitchen, because based on Molin et. al. [54] the internal heat gains in the bathroom are negligible.

Performance of developed technical solutions implementing the CB concept was evaluated by advanced level of IDA-ICE, version 4.22 [55]. The advanced level is an object-based interface (similar to MATLAB-Simulink) which allows the use of detailed component models and development of your own models, e.g. a mixing loop.

3.3.1.4 Detailed modelling of bypass flow in the CB and influence of DHW tapping

As already mentioned, bypass flow needed in case with the needle valve to keep inlet to substation on 35°C is 30% lower than the bypass flow needed by the thermostatic bypass valve with 3°C deadband, but on the other hand the needle valve is missing the possibility to stop the bypass flow when the temperature increases over the desired bypass temperature, i.e. in our case 35°C. Such increase happens during and after every DHW tapping, because the supply service pipe is full of 50°C DH water, and it takes around 45 minutes until the temperature in the service pipe cools down back to 35°C, as it can be seen in Figure 3.19. The black curve indicates DHW tapping, occurring every third hour for the period of five minutes and the dotted red curve the decrease of the temperature of bypass flow from 50°C back to 35°C, which takes approximately 45 minutes.

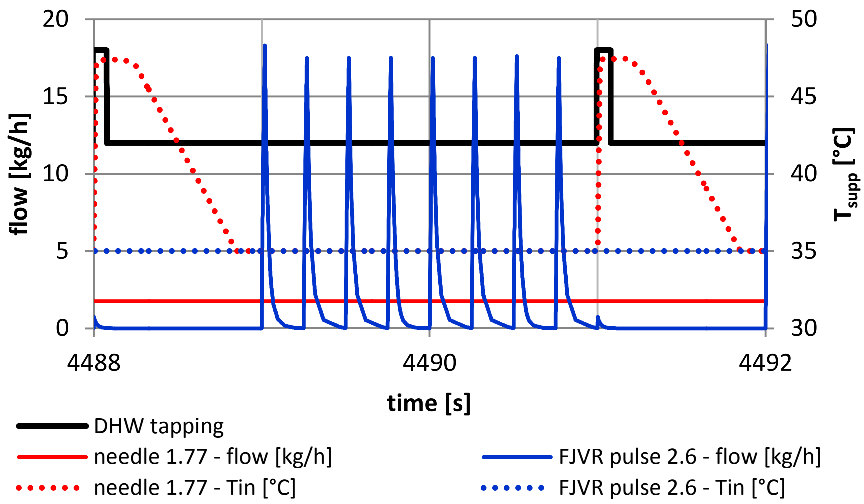


Figure 3.19 – Influence of DHW tapping on the temperature and flow rate of DH bypass water on the inlet to substation, when controlled by needle valve and thermostatic bypass valve

The figure also shows the water mass flow for both bypass solutions. It can be seen that the bypass water is in case of thermostatic bypass valve supplied to the floor heating in pulses every 15 minutes, but the bypass flow is stopped during and after each DHW tapping. Therefore it is expected, that advantage of the 30% lower flow rate in case of needle valve will change when considering the DHW tapping. Furthermore it is expected that the pulse and continuous delivery of the bypass flow have influence on heat transferred to the FH.

Normally the FH is in IDA-ICE modelled as one FH element, simulating performance based on the logarithmic mean average temperature and not giving the possibility to see distribution of floor surface temperature. Therefore we divided bathroom floor to 81 FH elements, to make possible model also spatial distribution of the floor surface temperature and the temperature drop along the heating pipe. However, the hydronic connection of each element should be made manually with the preceding and succeeding element and furthermore each of the elements should be manually connected with four neighbouring elements to account also for the horizontal conduction. Considering the number of manual connections, this solution is rather time consuming and very vulnerable for mistakes and even we spent considerable amount of time, we unfortunately haven’t succeed to build the model properly. Therefore we decided to model the FH just as a one element.

To find out influence of DHW tapping on total bypassed volume for needle valve and thermostatic bypass valve and the difference in pulse and continuous delivery of bypass flow to the FH, we make comparison of several cases. To catch all the dynamic of DHW tapping and pulse bypass flow, the simulations were performed with maximal time step of 0.001 s. Due to the long computational time the simulations were performed only for very short time periods, in range of days.

We modelled four different cases and all of them for two locations in the DH network (close and middle distance from the heating plant) differing in the nominal bypass flow.

- The first modelled case was case of needle valve (for close located customer with flow without DHW influence of 1.77 kg/h) providing continuous bypass flow (see Figure 3.20, triangle markers)
- The second case was the thermostatic bypass valve, operating with pulses and therefore with 50% higher volume of bypassed water (i.e. for the customer located close to the heat plant the bypass flow without the DHW influence of 2.6 kg/h), (diamond markers)

To investigate also influence of pulse/continuous operation of bypass flow on performance of FH, we modelled thermostatic bypass valve with continuous flow instead of pulse, giving on average the same amount of bypassed DH water. This simulation was made however for nominal bypass flow of 1.77 kg/h and 4.68 kg/h, to investigate how it will look if the thermostatic valve retain “stop” function, while remove the 50% higher bypassed volume as the result of removing 3°C dead band, which can be in fact realised by electronically controlled bypass valve. Therefore the:

- Third case is modelled as thermostatic bypass valve but with **continuous flow** instead of pulse, giving on average nominal flow of 1.77 kg/h (circle markers)
- Forth case as thermostatic bypass valve, modelled again with **pulse flow**, but also for the nominal flow of 1.77 kg/h (square markers)

All the cases included influence of DHW tapping, performed every three house, daily between 6 a.m. and 12 p.m.

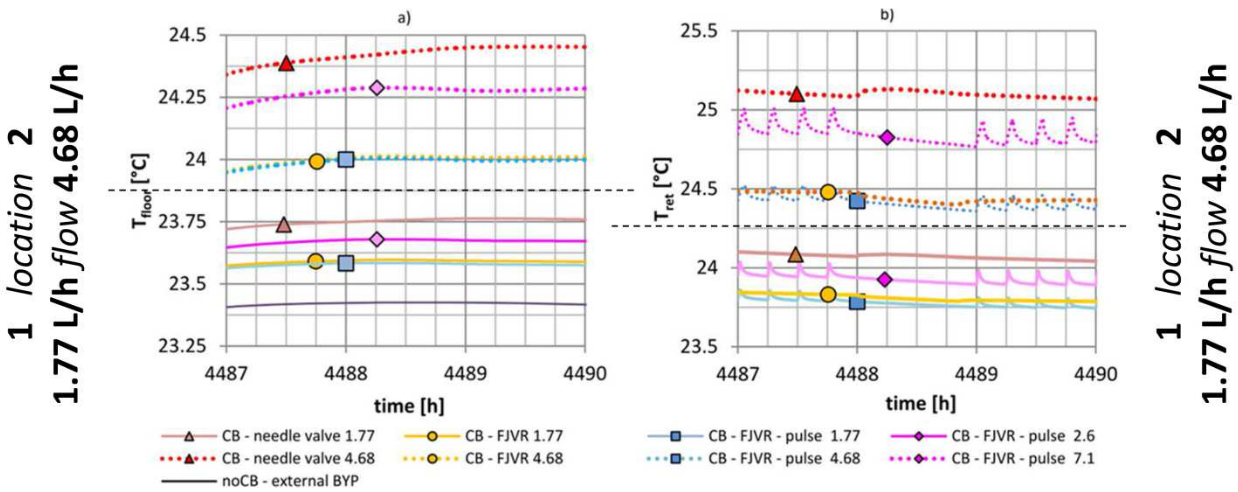


Figure 3.20 - Comparison of: a) T_{floor} and b) T_{ret} in CB for two locations in the DH network modelled with four different control strategies for the bypass flow, with the output for results in steps of 3.6 s.

Figure 3.20 compares the floor surface temperature (T_{floor}) in the bathroom and the return temperature of bypassed water (T_{ret}) for two locations in the DH network (close

– lower part of the figure and middle distance from the heat plant - upper part of the figure) for three modelled variants of the thermostatic bypass valve and 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 (time h=4488 h), because it does not stop the flow of bypassed water. Second, although the thermostatic bypass valve (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 average supply temperature and thus in lower floor surface temperature and lower return temperature of bypass water than with the needle valve, even the volume of bypassed water is still 7% higher. And finally, there is a negligible difference in the floor surface temperature and the return temperature of the bypass water between the thermostatic bypass 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 thermostatic bypass valve 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.

3.3.1.5 Specification of modelled cases

We investigated CB solution applied to three locations in the DH network, resulting in different bypass flows and for each of the location we applied between four and six different solutions. The first solution (case 1) was the reference case, with traditional external bypass, bypassing the DH water back to the DH network. The second case (case 2) was solution also with traditional external bypass and without CB, but the FH in the bathroom was controlled by thermostatic return valve, installed on the return pipe of the FH loop with set-point temperature 25°C. This case was investigated in order to compare if the traditionally used thermostatic return valve can be used instead of CB. Both mentioned cases are depicted in Figure 3.18a. The CB solution realised by needle valve is represented by case 3 (Figure 3.18b) and the CB solution realised by thermostatic bypass valve (case 4) is shown in Figure 3.18c. The case number 5 can be described by the same figure, but it is hypothetical solution of electronically controlled thermostatic bypass valve, combining advantages of needle valve and thermostatic bypass valve by removing the deadband of self-acting controller. Finally, the last investigated case (case 6) was installation of CB concept into the directly connected space heating system with mixing loop (see Figure 3.18d). Overview of all simulated cases can be found in Table 3-2, including definition of nominal bypass flow rates.

Table 3-2 - Matrix of simulated cases

case #	abbreviation	T_{sup} to CB >35°C	flow [kg/h]		
			1.77	4.68	9.36
1	noCB - FJVR	yes	✓	✓	×
2	noCB - external bypass	no	✓	✓	✓
3	CB - needle valve	yes	✓	✓	✓
4	CB – FJVR	*no*	*2.6*	*7.1*	*14.0*
5	CB - electr. step valve	no	✓	✓	✓
6	*CB - FJVR + mixing loop*	*no*	×	*7.1*	×

✓ - simulated × - not simulated

3.3.2 Results and Discussion

3.3.2.1 Floor heating without CB

Figure 3.21 shows floor surface temperature (T_{floor}) and operative temperature (T_{op}) *(comparable control strategies have the same markers)* in bathrooms located in close and middle distance from the DH heating plant during a two-day period for five investigated cases. It is the reference case without FH (case 2 – black diamond markers), the case with FH controlled in traditional manner by thermostatic return valve with set-point temperature 25°C (case 1 - grey diamond markers) and case with CB controlled by thermostatic bypass valve, modelled for two locations in the DH network (case 4 - square markers for the customer located close and circle markers for the customer located in the middle distance from the heating plant).

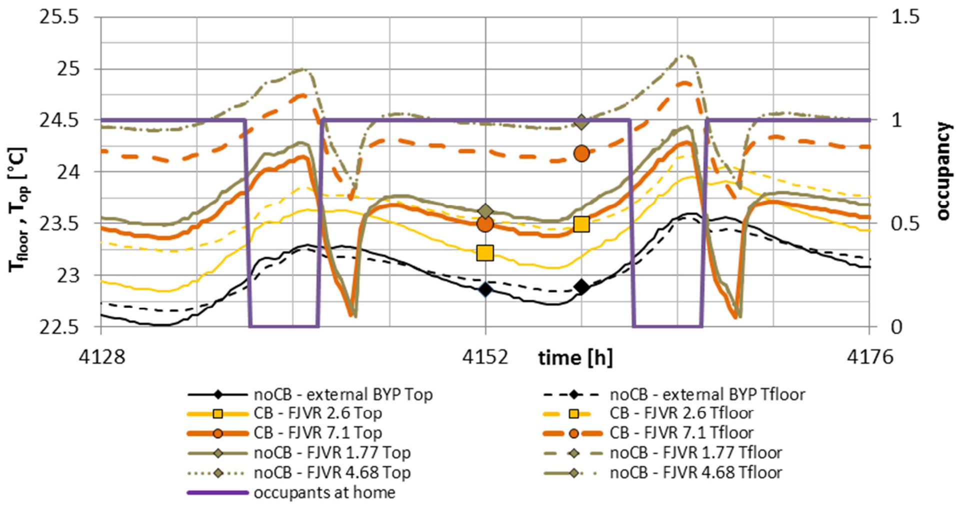


Figure 3.21 – T_{op} and T_{floor} in bathroom during non-heating period, considering various control of floor heating system

First, we can see that for the reference case without FH in the bathroom, T_{op} and T_{floor} are almost identical and lowest from the presented cases. The floor temperature is 23°C, meaning that there is still possible to increase the floor temperature to improve the thermal sensation of bare feet occupants. On the other hand T_{floor} and T_{op} for the

FH loop controlled in traditional matter by the thermostatic return valve is the highest from investigated cases, having the same performance no matter of the location of the user, i.e. the curve are on top of each other. Furthermore, the T_{op} is in the bathroom for some periods over 24°C and it triggers opening of the window if the occupants are at home (plain violet curve). The window is closed again when the T_{op} drops below 22°C or the occupants leave the house. The flow through the FH loop varies to meet the requirement of 25°C set-point temperature of the water leaving the FH loop and for some period during the non-heating season it mean drop below the bypass flow required to keep the inlet to substation on 35°C. Namely, for the customer located close to the heating plant the flow through the FH drops below 1.77 kg/h for 3% and for the customer in middle distance below flow of 4.68 L/h for 17% of non-heating period. The results therefore show that FH controlled by thermostatic return valve with set-point temperature 25°C cannot be used as a full replacement of traditional thermostatic bypass to keep the substation ready on 35°C. T_{op} and T_{floor} for the CB concept realised with the thermostatic bypass valve depends on the bypass flow, defined by position of the customers in DH network and lies between the reference case without FH heating and the case with traditionally controlled FH.

3.3.2.2 Floor *heating* with CB

Figure 3.22 *(note: comparable control strategies have the same markers)* compares T_{floor} in the bathroom of low-energy single-family houses situated at short (square markers), medium (circle markers) and long distances (triangle markers) from the heating plant for a two-day period during the non-heating season. It shows CB solutions realised with a needle valve (case 3) and thermostatic bypass valve (FJVR) with a 3°C deadband (case 4) together with case of traditional redirection of bypass flow directly back to DH network (case 1) and case with FH in the bathroom operated during non-heating period and controlled by thermostatic return valve (case 2).

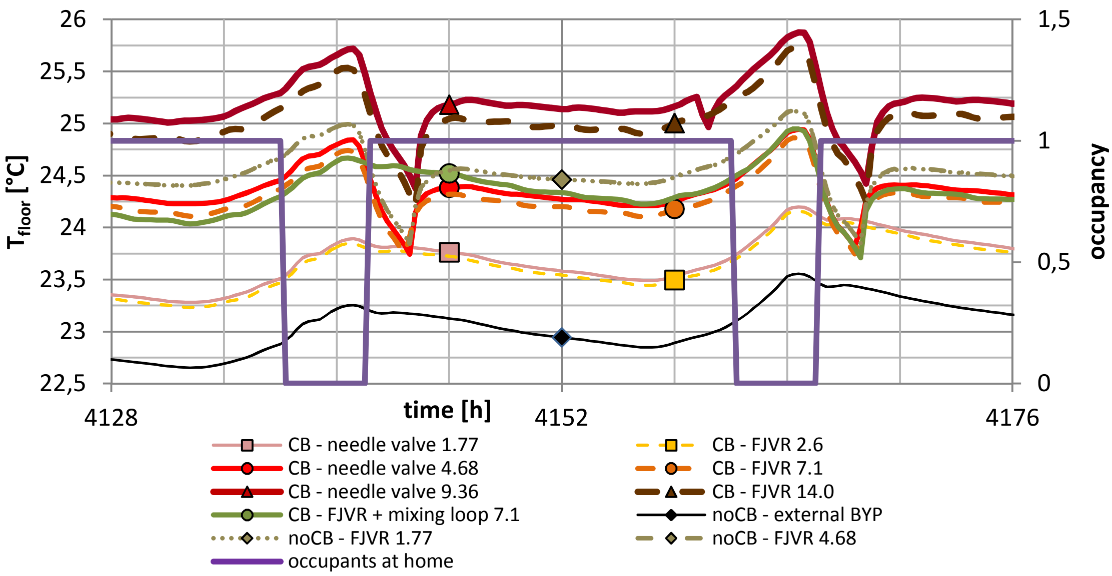


Figure 3.22 – T_{floor} during a non-heating period in a bathroom with CB realised with a needle valve or an FJVR valve

Looking on the reference case without FH (black plain curve), the surface floor temperature in case of installation of CB solution 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, and show negligible difference between the bypass flow controlled by needle valve (solid lines - shades of red) or thermostatic bypass valve (dashed lines - shades of yellow). It can be said, that the further from the heat plant is the user situated, the higher bypass flow needed to keep 35°C receives and thus has also higher temperature in the bathroom. Installation of CB to the system with mixing loop has practically no influence on the performance compared to the comparable case without the mixing loop (as can be also seen in Table 3-3).

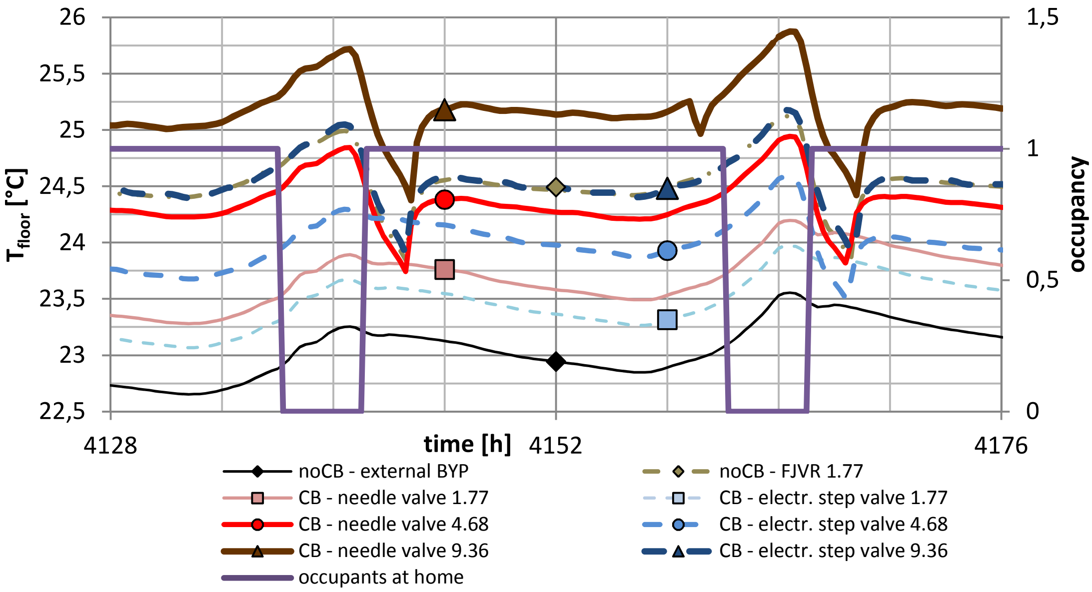


Figure 3.23 - T_{floor} during non-heating period in a bathroom with CB realised with a needle valve or electronic step valve

Figure 3.23 shows also T_{floor} for CB located in three different places in the DH network, but compares solutions realised with a needle valve with the solution realised by hypothetical electronic step valve, i.e. electronically controlled thermostatic valve with reduced deadband, resulting in the same nominal flow rate a needle valve but providing “stop function” for the water with temperature above desired set-point. It can be seen that reducing the nominal flow by 50% while keeping the stop function result in additional reduction of T_{floor} by 0.1°C, 0.25°C and 0.25°C in comparison to CB realised with the needle valve (or an thermostatic bypass valve, as shown in Figure 3.22).

3.3.2.3 Comparison of all investigated cases

Table 3-3 reports the average values for whole non-heating period for all simulated cases, i.e. from 15 April to 15 November. Comparison of operative, average return and average floor surface temperature, together with bypassed volume and energy used in the FH confirms that the performance of the CB realised with the needle valve (case 3) is very similar to the solution realised with thermostatic bypass valve with 3°C deadband (case 4). However, CB realised by thermostatic bypass solution should be preferred because of simple adjustment, no need for re-adjustment when the DH supply temperature changes during the year and automatic shutdown of the CB during the heating period.

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.

	case #	nominal bypass flow	T_{op} avg.	T_{floor} avg.	T_{ret} avg.	bypassed volume	average heat output from FH	energy delivered by SP	energy used in FH	heat demand incl. FH	increase of heat demand	CB cost for customer	bypass cost for DH company
		[kg/h]	[°C]	[°C]	[°C]	[m³]	[W]	[kWh]	[kWh]	[kWh]	[%]	[DKK]	[DKK]
location 1	1	1.77	23.5	24.4	25.2	24.3	97	1212	500	2945	17%	325	-49
	2	2.6	22.4	22.4	35.0	9.7	0	435	0	2352	0%	0	110
	3	**1.77**	23.0	23.0	23.3	9.1	32	430	165	2518	7%	107	-3
	4	**2.6**	22.7	23.0	23.2	9.7	29	435	150	2502	6%	98	12
	5	**1.77**	22.9	22.9	22.9	6.9	22	322	116	2467	5%	75	3
location 2	1	4.68	23.5	24.4	25.2	32.6	96	1454	496	2943	17%	322	-199
	2	7.1	22.4	22.4	35.0	25.7	0	1089	0	2348	0%	0	97
	3	**4.68**	23.2	23.8	24.4	24.1	72	1069	373	2721	14%	242	-151
	4	**7.1**	23.2	23.7	24.3	25.7	64	1089	333	2681	12%	216	-119
	5	**4.68**	23.1	23.3	23.8	16.8	46	727	236	2586	9%	153	-90
	6	**7.1**	23.2	23.7	24.4	25.7	63	1089	323	2697	12%	210	-119
location 3	2	14.0	22.4	22.4	35.0	50.9	0	2126	0	2348	0%	0	89
	3	**9.36**	23.5	24.8	25.8	48.3	127	2110	655	2996	22%	425	-341
	4	**14.0**	23.4	24.6	25.6	50.9	111	2126	571	2919	20%	371	-283
	5	9.36	23.3	24.1	24.8	34.0	81	1428	418	2766	15%	272	-213

The CB concept controlled by the thermostatic bypass valve (case 4) results for the three locations in the DH network in an increase of average floor surface temperature from 22.4°C (reference case) 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 without the CB concept to 23.2°C, 24.3, 25.6°C, respectively, meaning additional cooling between 9.4°C to 11.8°C. From the economic perspective, the customer gained 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 heating demand. We are assuming heat price of 650 DKK/MWh. However, we suggest bill all customers for the same price, without considering their location in the DH network, because there should be no difference. Furthermore the price should reflect the fact that the heat used in the CB would be lost in the DH network anyway if it was not extracted for in the bathroom floor heating.

The cost of bypass operation for the DH utility for individual solutions is calculated for 10 m long service pipe as the heat lost in the service pipe (mass flow of bypassed DH water * thermal capacity of the water * temperature difference at the service pipe, i.e. 50°C, 40°C or 37.5°C – 35°C) minus payment for the heat from the customer. The results show that the heat sold for the operation of CB covers in most of the cases running cost of the bypass in the service pipes, while in the case without CB, the DH utility will pay for the running of the traditional bypass solution roughly 100 DKK during non-heating season.

Further saving potential, mainly for the location in middle and far distance from the heating plant can be seen in application of electronic bypass valve (case 5) reducing the volume of bypassed water roughly by 30% while keeping the inlet to substation in 35°C, but slightly reducing the heat transferred to the bathroom floor.

The CB solution can be substituted by the FH with thermostatically controlled return valve with set-point 25°C, however without changing the set-point temperature the 35°C at the inlet to substation it is not guaranteed and the traditional bypass valve will be activated, resulting in traditional redirecting of bypass water back to the DH network without additional cooling. The floor surface temperature will be for the customers located in close and middle distance from the heating plant higher than in case of CB, but as well the bill for heating the bathroom during the non-heating period.

3.3.2.4 Effect of CB on heat production and distribution

Table 3-3 reported results only from the perspective of individual buildings. The evaluation of the performance of CB from perspective of DH network was tested on the example of the low-temperature DH network supplying 40 low-energy houses in Lystrup, Denmark [15], see Figure 3.17. It should be stressed that we adopted only the layout of the DH network, while we kept the original house described previously in the text. The network was modelled in software Termis® [56], as a steady state simulation. The bypass temperature required in all buildings was set to 35±1.5°C, providing continuous bypass flow representing the conditions for the CB operated with a needle valve. However, since the bypassed volume is only 6% higher than for the CB controlled by thermostatic bypass valve, we assume that the results are valid for both cases. The heat demand in every bathroom was set to 30W, chosen as an average value, on the conservative side considering that people are not opening windows in the bathrooms when the temperature rise above 24°C and thus the heat transfer to the bathroom floor is reduced.

As a consequence of applying CB in the buildings, the DH water bypassed through the FH was further cooled by 7.5°C on average (max.: 8.0°C, min.: 4°C). The average return temperature at heating plant drop from 27.7°C to 23.8°C, as can be seen in Table 3-4 under case B, representing the simulations with applied CB concept. Case A represent reference case without the CB, just with continuous bypass.

Table 3-4 – Results from the network simulation with the application of only the "continuous bypass" (A) or "CB concept" in the summer season (B).

	case		difference
	A	B	
Heating Power [kW] (plant)	3.8	4.5	18%
T_{supply} [°C] (from the plant)	55	55	-
T_{return} [°C] (to the plant)	27.7	23.8	-3.9
Heating Load [kW]	-	1.2	-
Total Heat Loss [kW]	3.8	3.3	-13%
Heat Loss Supply [kW]	2.25	2.3	2%
Heat Loss Return [kW]	1.55	1	-35%
Heat loss/production [%]	100	72.4	28%

The table furthermore show that the heat loss from the return pipes was reduced by 35%, thanks to the reduced return temperature. However since the DH network is built from the twin pipes and supply and return pipe are in the same casing and affects each other, there is a slight increase of the heat loss from service pipe. The final impact of the CB concept on the DH network is 13% reduction of heat loss. Nevertheless the lower return temperature coming to the heating plant requires additional energy input from the heat source, in our case 0.7 kW, which at the same time compensated by heat additionally sold 1.2 kW for heating of CB. However the additionally needed 0.7 kW represents only 60% of the total heat needed in CB, meaning that the DH utility should bill the customers only for 60% and remaining 40% should be free. Applying this discount on the case of CB realised with the thermostatic bypass valve, the price for the individual customers will be between 60 and 220 DKK, depending on their location in the DH network, but as it was discussed before the price should be recalculated to be same for all customers.

Furthermore if the DH utility s from the reduced return temperature to the heating plant also in terms of energy efficiency (condensation boiler, combined heat and power) or the lower return temperature doesn't represent additional cost (geothermal heat plants) the price for the heat used in CB should be additionally reduced.

3.3.3 Conclusions

- Heat loss caused during the non-heating period by using the traditional external bypass is reduced by application of "comfort bathroom" solution, redirecting the bypass flow to the bathroom floor heating where is additionally cooled while heating the bathroom floor.

- The bypass flow rate varies for different locations in the DH network. The longer the distance from the heating plant, the lower the temperature of DH water at the beginning of the service pipe and the higher the bypass flow needed to keep 35°C at the inlet to substation, resulting in more heat for the floor heating in the bathroom.

- By application of comfort bathroom concept, the bathroom floor surface temperature in the reference house increased during the non-heating season between 0.6 and 2.2°C on average, while the bypass water was cooled from 35°C in case of traditional bypass solution with thermostatic valve to between 23.2°C and 25.6°C, depending on the location of the customer in the DH network.

- Cooled bypass water returning to the heating plant reduces heat losses from the DH network by 13%, covering 40% of the heat transferred additionally to the bathroom floor heating. Therefore the customers should pay only remaining 60% of the heat used in the comfort bathroom during the non-heating season. Applied on the reference case of low-temperature DH network with 40 low-energy houses it represents annual cost between 60-220 DKK per customer. However it is suggested to bill all customers with the same, averaged price disregarding their location.

- In case the DH heating sources making profit from reduced DH return temperature e.g. by improved condensation of flue gases, increasing power generation in combined heat and power plant or having no additional cost with reduced DH return temperature (geothermal heat plant), this improvement in should be also regarded in the price for operation of comfort bathroom.

- Modelling of bathroom floor heating with pulse and continuous flow show that the thermal mass of the bathroom floor reducing for the investigated cases differences in type of flow and therefore the heat transferred to the bathroom floor depends mainly on the flow rate of the bypass water.

- Continuous bypass flow provided with the needle valve instead of traditional thermostatic bypass valve with deadband reduces for case of constant DH supply temperature bypassed volume by 30%. However considering changes of DH supply temperature caused by tapping of DHW water, in fact occurring many times per day, results negligible difference in bypassed volumes caused by the fact that thermostatic bypass valve closes the flow when the flow temperature is above the set-point temperature.

- For implementation of comfort bathroom concept is therefore suggested use traditional thermostatic valve because of simple settings requiring only set-point temperature while the solution with needle valve requires calculation of required flow, changing with the location of the customer in DH network and changes with changing of DH supply and soil temperatures.

- Considering influence of DHW tapping and changes in temperature of the bypassed DH water related to the temperature changes in the DH network the continuous bypass realised by the needle valve is not recommended. The optimal solution could be electronically controlled thermostatic bypass with reduced dead band, combining advantage of lower bypass flow while still keeping the thermostatic function.

- CB concept realised in the typical medium temperature DH network built from the single pipes without state-of-the art insulation properties will result compared to the low-temperature DH network in increased bypass flow needed to keep inlet to the substation at 35°C, but at the same time in possibility to save more heat from the return pipes due to the worse insulation properties of the DH network. For the customers it mean higher bypass flow and thus more heat available in the bathroom FH and higher comfort for discounted price.

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 [57]. The research related to low-energy buildings is reported in conference paper [58].

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 [59]. 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 [53] with an energy framework of 63 kWh/(m^2.a), existing building from the 1970s have considerably greater energy demand, resulting in a typical energy use of about 200 kWh/(m^2.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 [60]. 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.

However the renovation of existing buildings should be seen from the perspective of the integration of renewable sources of heat which needs to be built before 2035 because it is cheaper to refurbish the existing buildings as fast as possible to reduce their peak capacity and thus also the investment costs for low-temperature DH. Subsidies for the building refurbishment could be therefore from the long-time perspective advantageous [61].

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 (i.e. dependency of the temperature supplied to the space heating system on the outdoor temperature) 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 157m^2 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], [62]. 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 [63] 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 [64] 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 (4.2 W/m^2 – constantly) [58] and ran the model with the Danish design reference year weather file. By step-by-step lowering of the supply temperature for various outdoor temperatures, we defined a supply temperature curve for the SH system. To maintain the same hydraulic conditions in the DH network we limited the maximum flow rate from the DH network to the value defined originally for the design conditions of radiators 70/40/20.

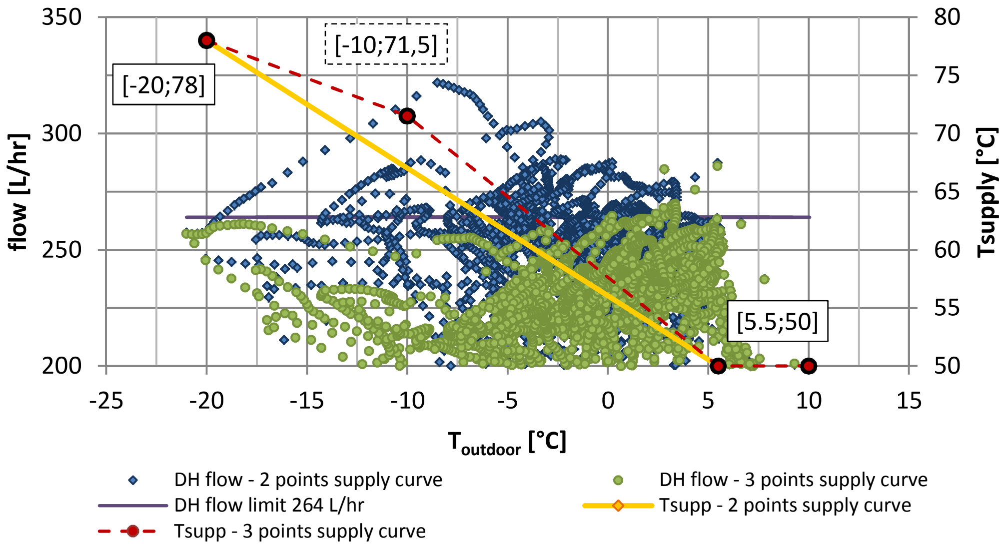


Figure 4.1 - Construction of weather-compensated supply temperature curve for non-renovated house, set-point temperature 22 °C.

Figure 4.1 shows flow of heating water needed in the SH system for linear and non-linear supply temperature curve. It can be seen that considering the supply temperature curve as linear results in non-uniform use of DH capacity (blue diamonds) and since the flow is many times above the DH flow limit (based on the design conditions 264 L/h), it will be needed to further increase the supply temperatures to reduce the maximal flow below the limit. However, defining the supply temperature curve with at least one additional point results in equalized use of flow capacity of the DH network and thus reducing the supply temperature to the minimal possible values. The additional points on the supply temperature curve were found by continuous adjustment of the supply temperature curve for various outdoor temperatures until the actual flow in the SH system approached close to the hydraulic limit of the DH network.

Non-linearity of the supply temperature curve is caused by the thermal capacity of the building. Even the outdoor temperature rises and thus gives signal to reduce the supply temperature to the SH system, the building still keeps the “cold” accumulated from the previous period and it takes some time to heat up this mass to the new thermal condition. An alternative solution to define the supply temperature curve by more than two points is therefore “delay” in reduction of heating supply temperature for the periods when the outdoor temperature increases. We chose time delay of 6 hours and this condition is in further text called T_{out}SHIFT.

We also investigated case with supply temperature limited to 70°C, resulting in maximal flow rate increased from designed 264 L/h to 432 L/h. This condition is denoted “HIGHFLOW” and represent condition when the DH network has enough reserve in capacity to increase the flow. In reality this condition is very relevant because the DH networks are usually built with capacity reserve up to 30% [65]. Possibility to increase the maximal flow in the SH system depends on the design conditions for the system, but for the case of investigated house it doesn’t represent problem [57].

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 (designed for conditions 70/40/20) with low-temperature radiators (designed with condition 50/25/20), with the same projected area, but deeper (increased number of convection plates). Finally, we also investigated influence of milder outdoor temperatures then defined in Danish design reference year but considering the supply temperature curve defined for Danish design reference year. The outdoor temperature used is the outdoor temperature measured in 2009 during Realea project [12].

4.1.1 Results and Discussion

Figure 4.2 shows the supply temperature (T_{supply}) 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 (LT). The maximum flow in the DH network and the SH system is exceeded only in the case of “HIGHFLOW”.

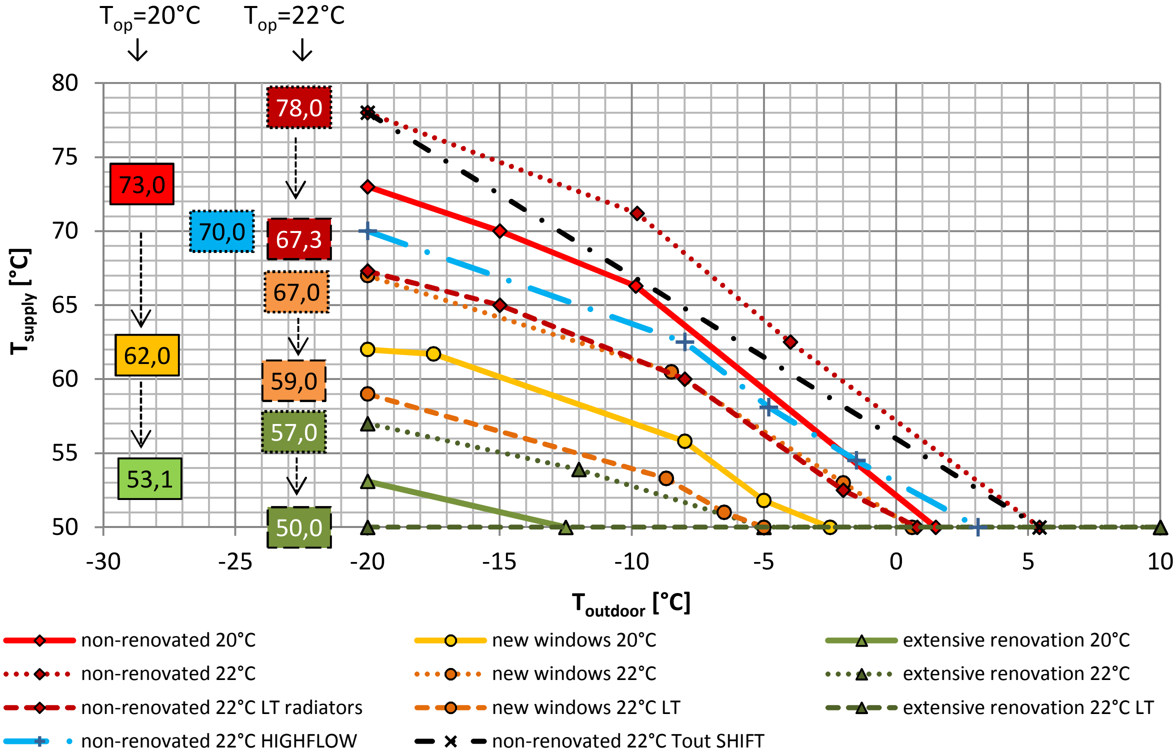
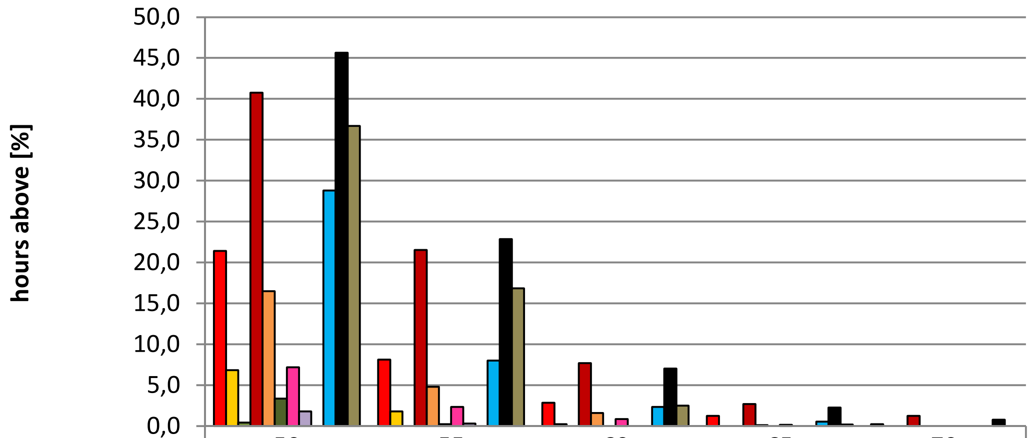


Figure 4.2 – Supply temperature curves for all the cases investigated. LT – low temperature radiators, HIGHFLOW – hydraulic limit of SH system and DH network increased to 400 L/h, T_{out}SHIFT – 6 hours time delay when the T_{out} increases

Figure 4.2 shows that reducing the desired set-point of operative temperature T_{op} from 22°C to 20°C reduces the maximum supply temperature needed by about 5°C for non-renovated and house with new windows and by 4°C for extensively renovated house. Installation of low-temperature radiators with the same projection area, just with more heat transfer plates (numbers in dashed rectangles), makes possible to keep 22°C T_{op} while compared to 20°C 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 by 3°C, i.e. down to 50°C for the extensively renovated house.

Increase of flow limit to 432 L/h while keeping the original radiators in case of non-renovated house means reduction of maximal supply temperature from 78°C to 70°C and reaching the value of 50°C supply temperature already for outdoor temperature 3°C instead of 5°C.

Figure 4.3 reports the duration of the period (in percentage of hours during the year) when the supply temperature needed to be increased over 50°C for all the cases investigated. The house with new windows can be supplied with low-temperature DH at 50°C and maintain an operative temperature of 22°C for approx. 83.5% (see Figure 4.3; 100 – 16.5%) of the year and needs a maximum supply temperature of 67°C (see Figure 4.2).



	50	55	60	65	70
non-renovated 20°C	21,4	8,1	2,9	1,3	0,2
new windows 20°C	6,8	1,8	0,2	0,0	0,0
extensive renovation 20°C	0,4	0,0	0,0	0,0	0,0
non-renovated 22°C	40,8	21,5	7,7	2,7	1,2
new windows 22°C	16,5	4,8	1,6	0,1	0,0
extensive renovation 22°C	3,4	0,2	0,0	0,0	0,0
non-renovated 22°C LT radiators	7,2	2,3	0,9	0,2	0,0
new windows 22°C LT	1,8	0,3	0,0	0,0	0,0
extensive renovation 22°C LT	0,0	0,0	0,0	0,0	0,0
non-renovated 22°C HIGHFLOW	28,8	8,0	2,4	0,5	0,0
non-renovated 22°C Tout SHIFT	45,6	22,9	7,0	2,3	0,8
non-renovated 22 °C ODUM	36,7	16,9	2,5	0,2	0,0

T_{supply} [°C]

Figure 4.3 – Percentage of hours during a year with supply temperature higher than 50°C; case non-renovated ODUM represents conditions with real weather data input

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 (0.2%) 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 desired 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 (i.e. extensive renovation) 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.

Keeping the supply temperature curve linear but apply six hours delay for the calculation of new supply temperature when the outdoor temperature increases lead in reduction of maximal water flow in SH system from original 432 L/h (flow in SH system for T_{op} set-point 22°C) to 280 L/h, which is fairly comparable with the design limit of 264 L/h, but the period with supply water temperature of 50°C-60°C is increased by 12% (from 41% of the year to 46%).

The limitation of supply temperature to 70°C and allowance of maximal flow 432 L/h resulted in decrease of period with supply temperature over 50°C by 30% (from 41% of the year to 29%). Applying the real weather data input resulted in reduction of hours with supply temperature over 50°C by 10% (from 41% to 37% of the year).

Table 4-1 compares the annual heating demand, the maximum heat power (P_{max}) needed for SH system (equal to the heating power delivered by DH system), the maximum supply temperature needed, and the average return temperature (TRW) from the SH system for all the investigated cases. It can be seen that the light (replacement of windows with standard ones) and extensive (low-energy windows and ceiling insulation) renovations reduce in comparison to the non-renovated building the maximum heat power needed for the SH system by about 21% and 45% respectively while the annual heating demand by 25% and 50% respectively. The percentage reduction of maximal heat power and annual heating demand are not the same and therefore the reduction in annual heating 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_{max}) and weighted average return temperature (TRW) for all the cases investigated.

$T_{operative}$ [°C]	internal heat gains [W/m^2]	T_{supmax} [°C]	RAD[a]	P_{max} [kW]	TRW [°C]	heating demand [MWh/a]	P_{max} reduction [%]	heating demand reduction [%]
non-renovated house - basic								
20[b]	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								
22[c]	4.18	70.0	O	10.5	35	24.5	0%	0%
22[d]	4.18	78.0	O	10.5	33.1	24.6	0%	0%
22[e]	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 on the basis of air temperature

[c]: maximum flow limit increased to 432 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 2009 for the non-renovated house reduced the maximum heat power needed for SH by 25% and the annual heating 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 connection heat power for SH and would lead to people feeling thermal discomfort and asking the DH utility to increase the DH supply temperature.

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.

4.1.2 Conclusions

- Single-family house built in 1970s, representing the typical example of Danish building stock, can be heated by DH to indoor temperature of 22°C during whole year without compromising thermal comfort or exceeding the design flow rate in the DH network and without any renovation measure if the DH supply temperature is raised above 60°C for roughly 8% of year (700 hours). This result shows that even under these unfavourable conditions it is possible to decrease the DH supply temperature for considerable periods during the year.

- In reality, most houses from the 1970s 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% of hours in the year when the temperature is above 60°C.

- By installing low-temperature radiators (with the same projected area as the original ones), the maximum supply temperature can be reduced further 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.

- 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. Therefore DH utilities should start require 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 needs to be defined by more than two points ensuring optimal use of flow capacity of the DH network and thus minimal DH supply temperature. Operation of DH network on lower than maximal flow capacity will result in higher heat losses and poor cost-effectiveness. The alternative solution to non-linear supply temperature curve is linear supply temperature curve with time delay in increase of supply temperature when the outdoor temperature increases.

- 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.

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. As result of this mistake, the set-point temperature activating the bypassing of the heat recovery unit in the ventilation system was reduced to 16°C to prevent overheating of the building. Overheating was reduced but the heat demand increased by 25% compared to case 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

As shown in chapter 2.3, low-temperature DH is cost-efficient for low-energy houses thanks to the reduced heat losses from DH network. For this purpose the heat loss from DH network can be defined as difference between the heat sent to the DH network from the DH heating plant and the heat consumed by the customers, i.e. their <u>heat demand</u>. If is the heat demand of the buildings reduced, as it is in case of low-energy buildings, the share of the heat losses from DH network is increased and thus increases the price of the heat for the customers and reduces cost-efficiency of the DH network. In addition to that, the cost-efficiency of DH network is related also to the <u>connection heat power</u>, i.e. the maximal heat output required by the customer, defining dimensions of the DH network and thus beside the heat loses from the DH network also the investment cost for the pipes and size of the heat sources.

Estimated heat demand and connection heat power depends on set of input parameters, being mainly represented by:

- internal heat gains - from the occupants and equipment
- desired indoor temperature
- space heating system and principle of DHW heating – only for connection heat power

<u>Expected heat demand of the buildings</u>

Heating 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 known to have considerable influence on the heating demand of the buildings and this importance is even higher when considering the low-energy buildings.

In Denmark all new buildings are divided into the energy classes based on the expected annual energy demand calculated with the national tool Be10 [66]. The annual energy demand for the residential buildings covers the energy needed for heating, cooling, running of HVAC systems (fans, pumps, etc.) and heating of DHW.

The values suggested for calculation of the energy class are internal heat gains 5 W/m^2, desired operative temperature of 20°C, and energy use for DHW of 13.1 kWh/m^2.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 heating demand of the building. Nevertheless, many engineers take these values as an input for the calculation of the expected energy use of buildings.

So it is not surprising that the real energy demand is higher than the one originally calculated. The reason is simply that the people use the house in different way than it was assumed for the energy calculation, or to be more precise, the energy calculation used input values different from the realistic ones. The buildings are built for the occupants and therefor the focus should be put on their needs. Report from Lystrup project [7] documented that the occupants tend to maintain an operative temperature of 22°C instead of 20°C, according to [67] the suggested value for DHW use is 20 kWh/(m^2.a)[1] instead of 13.1 kWh/m^2, and the improved energy efficiency of currently used equipment reduces the internal heat gains from 5 W/m^2 to 4 W/m^2 [68].

However, the problem is not just the disillusion of building owners on receiving higher energy bills than promised; underestimated heating demand negatively affect cost-efficiency study on the suitability of using the DH in investigated area.

<u>Connection heat power</u>

Connection heat power of the house is given by contribution of maximal heat power needed for space heating and DHW heating, being with consideration of individual simultaneity factors the base for dimensioning of the DH network [8]. The following text focuses only the connection heat power for the space heating, without considering addition from DHW preparation.

The needed heat output for SH system is in Denmark based on the dimensioned heat loss of the building, calculated in accordance with standard DS 418 [63], for an outdoor temperature of -12°C, an indoor operative temperature of 20°C, and excluding internal and external heat gains. This simple approach makes the calculation very simple but since the approach doesn’t reflect internal heat gains, changes in outdoor temperature and building mass it is expected to result in over dimension of the space heating system and thus increasing connection heat power needed for connection of the building.

Wrong input values can result in incorrect results of cost-efficiency study and thus stop DH project which will be in reality economically viable and thus their proper estimation is very important when analysing economic feasibility of any DH network. We therefore investigated how much the annual heating demand in the low-energy house changes when considering more realistic input parameters for internal heat

[1] Based on expected DHW demand 800 kWh/a.person, 4 people in 157m^2 house

gains and desired indoor temperature and how much of the connection heat power of the space heating system based on the traditional dimensioning approach is used in reality.

4.2.2 Methods

The influence of internal heat gains and desired indoor operative temperature on the maximum heat output needed from the SH system and the annual heating demand was investigated in IDA-ICE software [55] on example of single-family house with 159m^2 [29] build in accordance with low-energy class 2015 [53]. The house was built as multi-zone model (12 rooms = 12 zones) and the ground plan is shown in Figure 4.4. We chose to implement space heating system with radiators and dimensioned them for temperature levels 55/25/20 (supply, return, operative temperature) for the conditions in accordance with the standard DS 418 [63]. This case is in the text called “design case”, denoted later in the text as “design”.

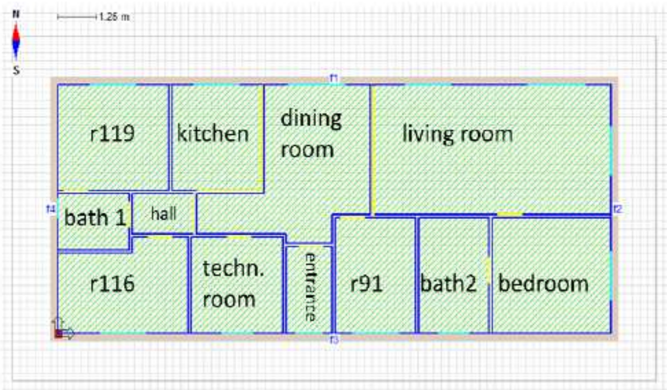


Figure 4.4 – Ground plan of 159m^2 single-family house built in IDA-ICE

The ventilation system was designed on the basis of BR10 requirements [53], resulting in a total air flow 60L/s for the supply and 60L/s for the exhaust. The performance of the heat recovery unit in the ventilation system was based on the long experience of a building company [69] set to 76%, meeting the requirement of minimal heat recovery efficiency of 75% defined in BR10 [53]. 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 to prevent the risk of cold draught for the occupants. For periods when the supply temperature of the air after the heat recovery dropped below 16°C or 18°C a heating coil in the ventilation system 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.

To investigate the influence of various modelling of internal heat gains and increase in desired indoor set-point temperature we performed annual simulation with Danish Design Reference Year (DRY) weather file and defined four cases, which are all listed in Table 4-2. The internal heat gains were first modelled as a constant value of 5 W/m^2, i.e. in accordance with the values suggested in Be10; represented by case “A”. Then we defined our own schedule of internal heat gains, reflecting reduced heat output of home appliances, resulting in average constant value of 4.18 W/m^2, represented by case “B”. The next modification was to investigate influence of modelling the internal heat gains not as averaged continuous value, but as real intermittent activities, changing over the time. Therefore we make case “C” giving on weekly average the same value of internal heat gains as case B, but changing intermittently due to the hourly schedule. The schedules for the activities are shown in Appendix and in [58]. The last case “D”, investigates change in desired indoor temperature from 20°C to 22°C.

Then we compared the maximum heat output and annual return temperature from the SH system, heating demand and length of space heating period.

4.2.1 Results and Discussion

4.2.1.1 Connection heat power

Table 4-2 show influence of various modelling of internal heat gains and desired indoor temperature on connection heat power needed from the DH system for the SH system for all investigated cases. For easier orientation, colours express magnitude of the value or the difference compared to the design case, lowest (green) and highest (red), in each column.

It can be seen that the SH system based on traditional dimensioning due to DS 418 (design case), needs 5.2 kW heat connection power (including heating coil for air). The radiators were chosen very close to the required dimensioned heat output, which can be seen from the return temperature of 25°C.

Table 4-2 – Connection heat power for space heating system (radiators, heating coil and total) and annual return temperature from the radiators for all simulated cases

case	Internal Heat Gains		T_{op}	Connection Heat Power for SH						T_{return} annual from radiators [°C]
				radiators		heating coil		Total		
	[W/m^2]	type		max. heat output [kW]	[%]	max. heat output [kW]	[%]	max. heat output [kW]	[%]	
design	0	-	20	2.9	0%	2.2	0%	5.2	0%	25
A	5	constant	20	2.2	-26%	1.2	-44%	3.4	-33%	20.5
B	4.18	constant	20	2.3	-21%	1.2	-44%	3.6	-31%	20.6
C	4.18	scheduled	20	2.8	-5%	1.3	-43%	4.0	-22%	21.2
D	4.18	scheduled	22	2.9	-1%	1.3	-43%	4.2	-19%	23.7

Running the simulation for whole year with DRY weather file and continuous internal heat gains 5 W/m^2 representing the values from Be10 (case “A”), the maximal needed heat output from the heating system drops by 33% to 3.4 kW. The radiators have reserve in the heat output of 26% and the heating coil 44%. The over-dimensioning is result of the fact that the radiators were designed with constant outdoor temperature of -12°C and the heating coil for air was dimensioned for outdoor temperature -15°C without consideration of heat recovery and both excluding internal heat gains. The 44% over-dimension of the air heating coil is the same for all investigated cases.

Reducing the internal heat gains from 5 W/m^2 (case A) to 4.18 W/m^2 (case B) results in only 5% increase of the maximal heat output for the radiators, while modelling the internal heat gains as intermittent (case C) results in roughly 16 and 21% increase of the heat output needed for the radiators compared to the cases modelled with internal heat gains as constant values of 5 W/m^2 and 4.18 W/m^2. The increase of desired indoor temperature from 20°C to 22°C has in comparison with the case C almost negligible importance (1%).

Modelling the internal heat gains with reduced value and intermittent profile increases the annual return temperature from the radiators maximally by 0.7°C, meaning no practical influence. Setting the desired indoor temperature from 20°C to 22°C increases the annual return temperature in comparison to case A by 3.2°C, but the value is still below the design value of 25°C. 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, meaning that even the indoor temperature is increased and the internal heat gains reduced the space heating system designed in accordance with DS 418 is still able to deliver requested comfort.

Comparing the heat connection power to the DH calculated for overall SH system based on the standard dimensioning approach with the results of more realistic cases "C" and "D", the original value 5.2 kW decreased by 1.2 and 1.0 kW, which is a decrease of approx. 20%. However the 20% reduction is caused only by over-dimension of the air heating coil in the ventilation system. The fact that in the traditional DH networks the connection power for the SH system is usually increased by another 20-30% results in total 50-60% over-dimensioning of the SH connection power than really needed. On the one hand, over-dimensioning ensures free capacity to provide customers with needed heat and makes possible connect more customers, but on the other hand it results in unnecessarily large DH pipes with higher heat loss, which is contrary to the philosophy of the low-temperature DH concept. Over-dimensioning the connection heat power leads to need of higher heat capacity of the heating plants and thus further increasing the investment.

4.2.1.2 Annual heating demand

Contrary to the reduction of connection heat power (or maximal heat output of the space heating system), lower internal heat gains and higher desired indoor temperature increases the annual heating demand and length of the heating period (see Table 4-3).

Table 4-3 – Heating demand, length of heating period and annual return temperature from space heating for all simulated cases

case	Internal Heat Gains		T_{op}	Heating Demand				heating period [month]
	[W/m^2]	Type		radiators [kWh/a]	heating coil [kWh/a]	total [kWh/a]	increase [%]	
01	0	-	20	-			-	-
A	5	constant	20	2810	138	**2948**	0%	5.9
B	4.18	constant	20	3386	139	3525	20%	6.2
C	4.18	scheduled	20	3519	138	3657	24%	6.3
D	4.18	scheduled	22	4461	186	4647	58%	7.3

Reducing the constant internal heat gains from 5 W/m2 to 4.18 W/m2 results in considerable increase of the annual heating demand by 20% and extension of the heating period by 0.3 month, i.e. around 8 days. Modelling the internal heat gains intermittently based on the detailed schedule, results in an additional 4% increase in the heating demand and 5 days extension of the heating period, meaning some but not really important increase. But increasing the desired indoor temperature from 20°C to 22°C results in an additional heating demand of 27%, which is in accordance with the results of Tommerup [68] and Olsen [70]. Comparing this result with the reference case based on Be10 values, the heating demand increased by 58% and heating season by 42 days (1.4 months). Table 4-4 – influence of DHW on total heating demand

Case	Annual heating demand			
	DHW		SH & DHW	
	[kWh/(m^2.a)]	[kWh/a]	[kWh/a]	[%]
A	13.1	2083	5031	0%
B	20.1	3200	6725	34%
C	20.1	3200	6857	36%
D	20.1	3200	7847	56%

Adding also more realistic demand for DHW heating (see Table 4-4) than the values defined in Be10 calculations will not additionally change the percentage increase of annual heating demand for case "D", but it will further increase the absolute amount of heat sold to the customer, which is a considerable difference from the perspective of both customers and the DH company which should be reflected in the feasibility studies for DH networks.

4.2.1.3 Heating demand load

The heating demand load on the DH network is smooth for all the cases simulated, without large oscillations (see Figure 4.5). The reason is the control of each radiator by individual thermostatic valve with the thermostatic head (in IDA-ICE modelled as proportional controller), which reacts immediately on changes of heating demand in the room. However, this is not the case for floor heating, which is usually controlled by on/off valves resulting in rapid oscillations in heating demand from the DH network [58], not shown in the figure.

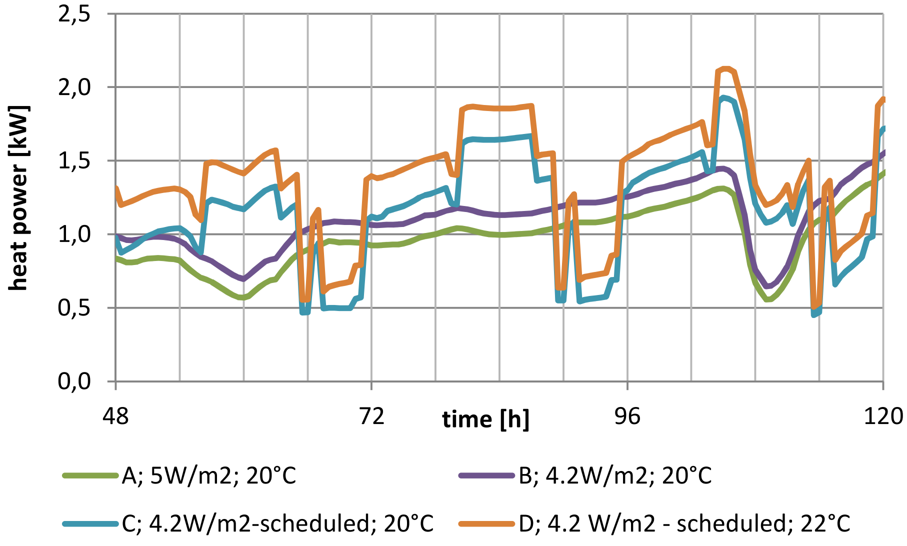


Figure 4.5 – Heat load on DH network from SH system with radiators (excluding heating coil)

Figure 4.5 shows the difference in heating demand needed for the space heating system modelled with continuous internal heat gains (cases A and B) and scheduled (cases C and D). It can be seen that compared to the averaged continuous value the heat output needed for scheduled internal heat gains oscillates ± 0,8 kW, but from the perspective of the DH network these oscillations are negligible.

4.2.1.4 Air heating coil

The heating coil covers only a small part of the heating demand, only around 4%, but still needs to be installed. 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 to 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 -8°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 the heat recovery, e.g. supplied by return water from the radiator system, but this system was not investigated.

Regarding limited heating demand of the heating coil but its high connection heating power, it is suggested to use electric heating coil rather than hydronic heating coil, because this would reduce the heat connection power by 2.2 kW, 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 as for DH, meaning possible problems with fulfilling energy frame of BR10 regulations [53]. This solution should be preferable mainly for houses with DHW heated in DH water storage tank, because it considerably reduces the needed connection power. But the benefit will be in the network seen also for houses with substation with instantaneous heat exchanger.

4.2.2 Conclusions

- Internal heat gains reduced from 5 W/m^2 to 4.2 W/m^2 due to higher efficiency of the house equipment results in approximately 20% increase in annual heating demand. Indoor temperature set to 22°C instead of 20°C representing value preferred by the occupants results in additional 27% increase of annual heating demand. Together it is approximately 60% increase of expected annual heating demand compared to the values suggested by Be10.

- Modelling internal heat gains as intermittent instead of continuous averaged values resulted in negligible increase of annual heating demand but in 16% increase of maximal heating output needed from the space heating system.

- Space heating system traditionally designed based on DS 418 for 20°C desired temperature fits very well required heating output needed for the most realistic case of daily living, i.e. 22°C desired temperature and reduced internal heat gains. The annual return temperature increases by 3.2°C, but it is still below designed return temperature 25°C.

- Documented higher value of expected annual heating demand will improve economy of DH feasibility studies and thus make DH more competitive to the alternative individual heating solutions.

- Space heating system with low-temperature radiators supplied by low-temperature DH provides required thermal comfort for occupants while keeping the heating load on the district heating network smooth, without high oscillations. However modelling the internal heat gains as with intermittent profile based on the hourly schedule varies the annual heating load ±0.8 kW.

- Considering defrosting of the heat recovery unit during very could outdoor temperatures requires to equip the ventilation system with air heating coil to ensure that air supplied to the rooms will be in worse case maximally 4°C lower than required indoor temperature and thus comply with high requirements for comfort.

5 GENERAL CONCLUSIONS

The following paragraph gives answers to the hypothesis defined in chapter 1.3. More detailed conclusions are at the end of each dedicated chapter.

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. The DHW systems designed for low-temperature operation ensure delivery of DHW with desired temperature, in time comparable to traditional DH and without increased risk of Legionella. However, to ensure the required thermal comfort for the 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.

DHW heated by Low-temperature DH

Based on the literature study is concluded that the DHW temperature can be below 60°C without increased risk of Legionella if the overall volume of DHW is below 3L. This finding sets the rule for designing the DHW systems supplied by low-temperature DH.

The laboratory tests confirmed that a low-temperature DH substation supplied with 50°C warm water can produce required 47°C warm DHW, while cooling the DH water to 20°C. However, developed numerical model accounting for the influence of the service pipes showed that the delivery time of DHW is highly dependent on the stand-by operation of the service pipes. Leaving the supply service pipe cool down in the periods without the heating demand leads up to 15 s waiting time for DHW with 40°C, which is by 5 s longer than 10 s suggested by the DHW standard. By using the external bypass, keeping the supply service pipe warm and inlet to the substation on 35°C the waiting time is reduced to 11 s, i.e. slightly above the suggested value.

The results therefore documents that low-temperature DH can be used for delivery of DHW, with required comfort temperature of 45°C and without increased risk of Legionella if the volume of DHW system is below 3L. Use of external bypass solution is needed to keep the waiting time for DHW short as possible.

Energy efficient use of bypass flow

Numerical simulations applied on the example of low-temperature DH network showed that the proposed solution of redirecting the bypassed water during the non-heating period to the floor heating in the bathroom reduces the heat loss from the DH network by 13% while increasing the floor temperature by 2.2°C on average in comparison to traditional bypass solution with thermostatic bypass valve.

The 13% reduction of heat loss from the DH network corresponds to 40% of the additional heat delivered to the bathroom and therefore it seems reasonable to bill the customers only for 60% of heat used in the bathroom floor heating during the non-heating period. In DH networks with DH heat sources profiting from reduced DH return temperature as e.g. condensation boilers, combined heat and power plants or solar-thermal plants should be added also benefit of reduced return temperature and result in further discount on heat used in bathroom floor heating during the non-heating period.

The concept is advantageous for DH customers by improvement of the thermal comfort for the discounted price, for the DH utility to monetize some of the heat losses and for the environment by increasing of energy efficiency of the DH heat sources.

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 55°C for almost 95% of the Danish 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 replacement of the currently used substations with the low-temperature DH substations to ensure that the DHW is produced 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 heating demand on the DH network and cooling the heating water to a low return temperature.

Be10 input values traditionally used for calculation of heating demand in the buildings, but in fact not meant for this purpose, leads in comparison with more realistic data reported in the literature and reflecting reduced heat gains thanks to the higher efficiency of home appliances and increased indoor temperature set-point to 22°C instead of 20°C in as much as 58% higher heating demand and in up to 20% over dimensioned heat connection power for the space heating system. Lower expected annual heating demand and over-dimensioned heat connection power negatively influences cost-effectiveness analyses of planned DH systems and therefore careful consideration of the correct input values is very important for their future successful expansion.

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 utilities 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° 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 in regards to Legionella proliferation in the HEX, kept consciously warm.

- 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 heating 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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8 LIST OF FIGURES

9 LIST OF TABLES:

APPENDICES

Appendix I

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

Appendix 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

Appendix III

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

Appendix IV

Schedule for Internal Heat Gains

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

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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 fulfil the comfort requirements for DHW instantaneously prepared.

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 CO_2, 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

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Table 1
Primary energy factors and energy frames for residential building in Denmark in accordance with BR08 and BR10.

Energy frame calculation [kWh/(m^2.a)]		Prim. energy factors	
		DH	Electricity
BR08			
Class BR08	70 + 2200/A[b]		
Low-energy class 1	35 + 1100/A	1	2.5
Low-energy class 2	50 + 1600/A		
BR10			
Class BR10	52.5 + 1650/A	1	2.5
Class 2015	30 + 1000/A	0.8	2.5/2.2[a]
Class 2020	20	0.6	1.8

[a] The value will be based on share of renewable sources, not decided yet.
[b] Gross heated area [m^2].

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 exergy 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 aluminium 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 grows 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 DH for DHW 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, while 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 W551 [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 fulfil 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 fulfil 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
Transportation delay for nominal flows for individual fixtures due to DS439, in DHW system in typical house in Lystrup, for pipes with inner diameter 10 mm.

DHW fixture	Nominal flow [L/min]	Length to fixture [m]	Volume in pipes [L]	Velocity [m/s]	Transportation delay [s] for	
					Nominal flow	Flow 0.2 L/s
Shower	8.4	2.2	0.17	1.8	1.2	0.9
Basin	3.4	4.1	0.32	0.7	5.8	1.6
Kitchen	6	6.3	0.49	1.3	4.9	2.5

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.

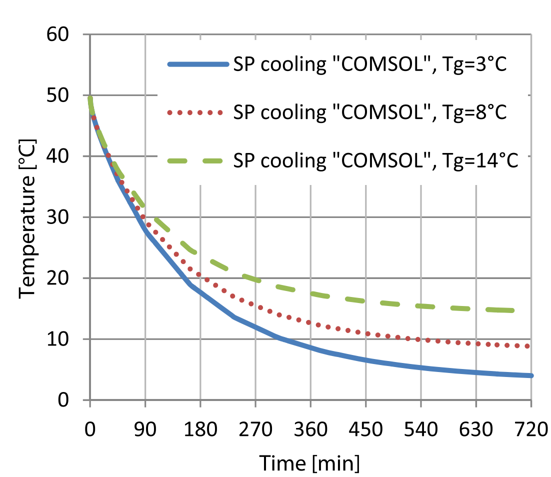

Fig. 1. Cooling down of DH water standing in an AluFlex 20/20/110 service pipe during idling. The initial temperature of water in pipe is 50 °C, and the initial temperature of insulation is 15 °C.

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_v-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.

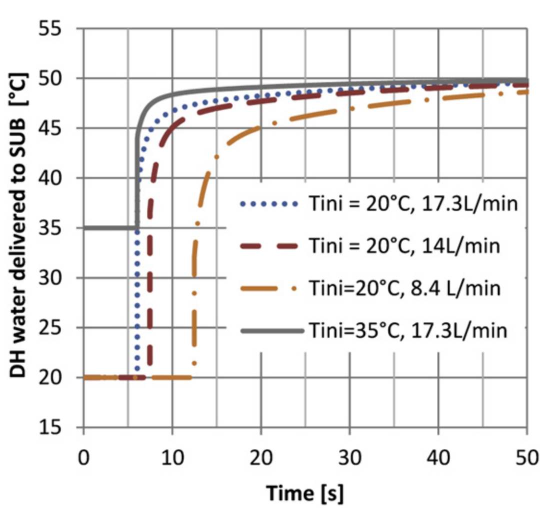

Fig. 2. Effect of the thermal capacity of an AluFlex 20/20/110 service pipe on the temperature of water delivered after substation idling.

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 (XB37H-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

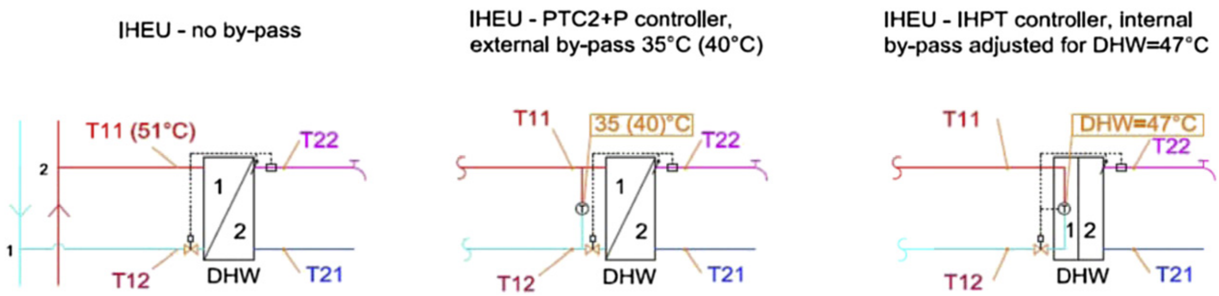

Fig. 3. Various IHEU bypass strategies: left – no bypass; middle – external bypass (cold HEX); right – internal bypass (warm HEX).

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. 21.6 °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 α [W/(m^2 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.

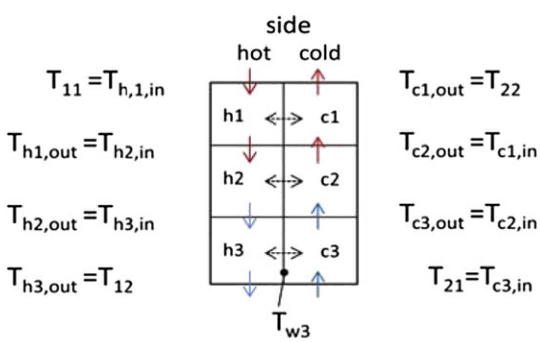

Fig. 4. Description of IHEUs numerical model with three sections.

Energy balance equation for cold medium

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

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

$$\frac{d}{d\tau}(T_{c1,\text{out}}) = \frac{1}{m_{c1} \cdot c_{p,c}} \cdot \left[\dot{m}_c \cdot c_{p,c} \cdot (T_{c1,\text{in}} - T_{c1,\text{out}}) - \alpha_{c/w} \cdot A_c \cdot \left(\frac{T_{c1,\text{in}} - T_{c1,\text{out}}}{2} - T_{w1}\right)\right] \quad (5)$$

Energy balance equation for hot medium

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

Energy balance equation for HEX plates

$$\frac{d}{d\tau}(m_w \cdot c_{p,w} \cdot T_w) = \alpha_{h/w} \cdot A_h \cdot \left(\frac{T_{h,\text{in}} - T_{h,\text{out}}}{2} - T_w\right) - \alpha_{c/w} \cdot A_c \cdot \left(\frac{T_{c,\text{in}} - T_{c,\text{out}}}{2} - T_w\right) \quad (3)$$

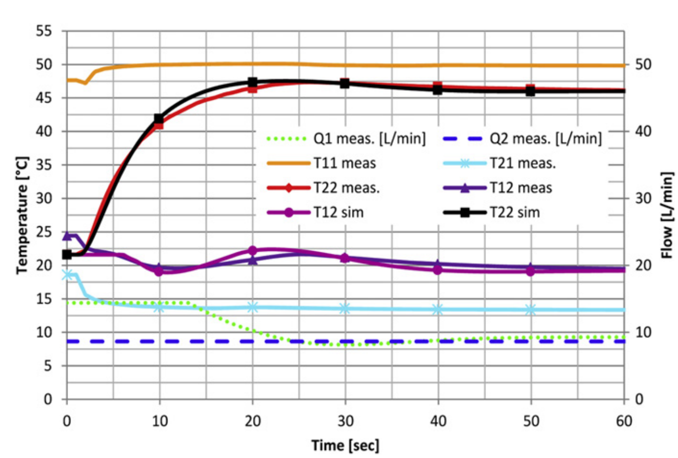

Fig. 5. Comparison of experimentally measured and numerically simulated temperatures T_{22} and T_{12} for an IHEU equipped with a PTC2+P controller for the same input data.

Convective heat transfer coefficient α

$$\alpha = 0.2 \cdot \lambda_w / (\delta_p \cdot 2) \cdot Re^{0.67} \cdot Pr^{0.4} \quad (4)$$

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_v 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_{22} and T_{12}. The numerical model (values are denoted "sim") can be in the dynamic build-up phase seen to be slightly faster than real measurements, but 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]. The code simulating the water cooling is used to calculate the temperature profile in the SP during idling. The result is the time period (see Fig. 1) needed to cool the last control volume of water in the SP down to temperature of 32.5 °C (opening temperature for the bypass). Then we fed the obtained temperature profile along the SP to the code modelling the dynamic heat-up of the SP, we set the flow to 3 L/min and got the time dependent temperature at the SP outlet during bypass operation (see Fig. 2). The flow of 3 L/min was kept until the temperature at the end of SP reached 37.5 °C (closing temperature for bypass). The result is the new temperature profile in the SP. Next, we re-applied the code modelling the cooling of the water standing in the SP and repeated the whole procedure recursively until we got the temperature profile in the SP after the desired number of bypasses (see Fig. 6).

Table 3
Nomenclature for energy balance equations – cold side.

A_c	Total plate area on the cold side in one section	[m^2]
$c_{p,c}$	Specific heat capacity of water	[J/(kg K)]
m_c	Mass of water in the section	[kg]
$\dot{m}_c$	Mass flow of water through the section	[kg/s]
m_w	Total mass of plates in the section	[kg]
Pr	Prandtl number	[–]
Re	Reynolds number	[–]
$T_{c,in}$	Temperature of water coming into the section	[°C]
$T_{c,out}$	Temperature of water leaving the section	[°C]
T_w	Average wall temperature in the section	[°C]
$\alpha_{c/w}$	Convective heat transfer coefficient	[W/(m^2 K)]
δ_p	Distance between individual HEX's plates	[m]
λ_w	Thermal conductivity of water	[W/(m K)]

Table 4
Recovery times for IHEU for all described cases, case 3–7 includes also influence of service pipe.

Case	Description	Recovery time of IHEU [s] to produce DHW				
		30 °C	35 °C	40 °C	42 °C	45 °C
Without SP						
M	Experimental measurement	4.1	6.0	9.1	10.8	15.8
0	Verification of num. model	4.6	6.3	8.6	10.0	13.1
1	Pure recovery time, $T = 50$ °C, HEX20	3.7	4.9	6.5	7.3	9.4
2	Pure recovery time, $T = 50$ °C, HEX40	4.1	5.5	7.4	8.55	10.9
With SP						
3	Continual bypass 50 °C–35 °C	5.7	7.7	10	11.3	14.1
4	Ext. bypass before 2nd bypass flow	5.9	8.7	11.0	12.4	15.5
5	Ext. bypass before 3rd bypass flow	7.2	9.5	11.9	13.2	16.4
6	Without bypass – 35 °C	7.2	9.8	12.3	13.6	16.9
7	Without bypass – 20 °C	11.0	12.7	15.2	16.7	20.9

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].

$$T_d = \frac{M}{K} + T_u - \frac{M}{K}\exp^{\left(-K \cdot \frac{L}{V}\right)} \quad (6)$$

where T_d is the downstream fluid temperature [K], T_u is the upstream temperature [K], L is the pipe

$$M = \frac{1}{\rho \cdot c_p}\left[[1 - \rho \cdot c_T] \cdot V \cdot \frac{\partial P}{\partial x} + \rho \cdot \frac{2f}{D} \cdot |V| \cdot V^2 + C_h \cdot \frac{T_a}{A}\right]$$

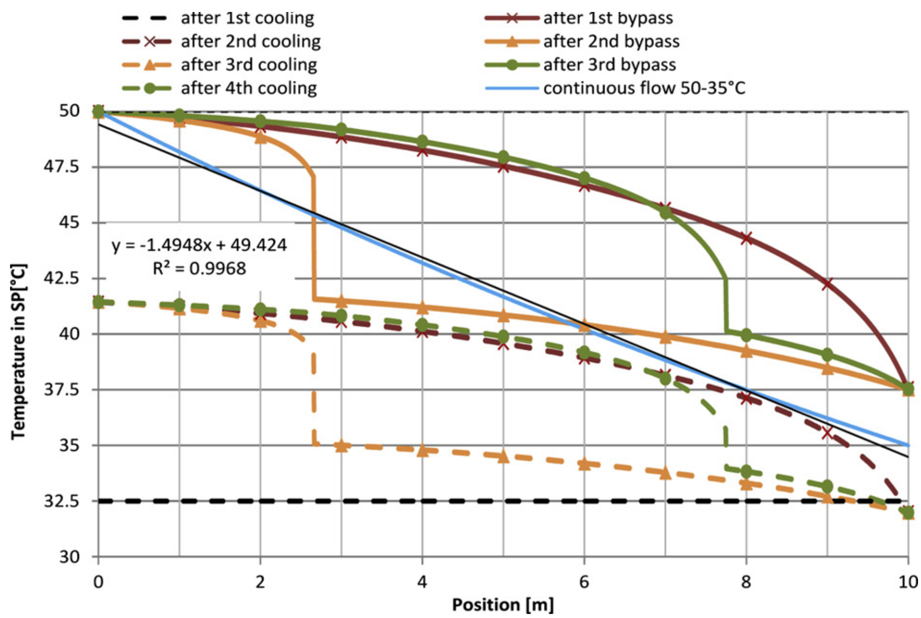

Fig. 6. Temperature profile along 10 m long SP AluFlex 20/20/110 for IHEU equipped with external bypass in traditional and continual operation modes. Traditional bypass: set-point 35 °C, dead band ±2.5 °C, bypass flow rate 3 L/min, $T_{ground} = 14$ °C. Continual bypass: temperature drop 50–35 °C, flow rate 0.024 L/min, $T_{ground} = 14$ °C.

$$K = \frac{1}{\rho \cdot c_p} \cdot \frac{c_h}{A}$$

length [m], T_a is the ambient temperature [K], C_h is the overall heat transfer coefficient (W/m/°C), C_p is the specific heat capacity at constant pressure of the fluid [J/kg/K], C_T is the specific heat capacity at constant temperature of the fluid [J/kg/K], A is the cross sectional area of pipe [m^2]. V is the velocity [m/s], $\partial P/\partial x$ is the pressure gradient [N/m^2/m], D is the hydraulic diameter [m], f is the friction factor [−], and ρ is the density of fluid [kg/m^3].

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 $\tau = 0$. 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). The bypass flow starts to move cooled water in the SP towards the substation until the temperature at thermostatic valve reaches 37.5 °C (i.e. 35 °C + 2.5). 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 thermostatic 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 the 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.

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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 (domestic hot water) comfort in bathrooms in low-energy buildings supplied by low-temperature district heating

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A B S T R A C T

Using a bypass to redirect a small flow through the in-house DH (district heating) 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 FH (floor heating) to cool down further and thus reduce the heat lost from bypass operation while tempering the bathroom floor and guaranteeing fast provision of DHW (domestic hot water). 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 FH during the non-heating period resulted in comparison to the reference case on average in a 0.6 °C–2.2 °C increase of the floor surface temperature and additional cooling of bypass water by 3.9 °C, reducing the heat loss from the DH network by 13% and covering 40% of the heat used in the bathroom FH. The use of the bypass flow in bathroom FH is a cost-effective solution exploiting the heat that would otherwise be lost in the DH network to improve comfort for customers at discounted price.

1. Introduction

The heat demand in buildings drops outside the heating season because the only heating requirement users usually have is in connection with DHW (domestic hot water). 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 IHE (instantaneous heat exchanger) 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 Fig. 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 DH (district heating) 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 (district heating) pilot project in Lystrup [3], the set-point temperature for the external bypass is 35 °C.

This operation, although necessary, results in increased heat losses 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

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List of abbreviations and symbols

$\dot{m}$	mass flow, [kg/h]
T_{bypass}	bypass set-point temperature, [°C]
T_{floor}	floor surface temperature, [°C]
T_{op}	operative temperature, [°C]
T_{ret}	return temperature, [°C]
T_{soil}	temperature of soil, [°C]
ΔT_{DB}	deadband, [°C]
CB	comfort bathroom
DH	district heating
DHW	domestic hot water
FH	floor heating
FJVR	TRV controlled by the fluid temperature
HEX	heat exchanger
IHE substation	instantaneous heat exchanger substation
SH	space heating
SUB	DH in-house substation
TRV	thermostatic regulation valve

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 FH (floor heating) 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, $T_{bypass,set}$ and the amplitude of the "deadband", ΔT_{DB}, which also defines a "top temperature", $T_{bypass,top} = T_{bypass,set} + \Delta T_{DB}/2$, and a bottom temperature, $T_{bypass,bottom} = T_{bypass,set} - \Delta T_{DB}/2$. The bypass control ensures that the temperature is kept within the range of operation set by the deadband.

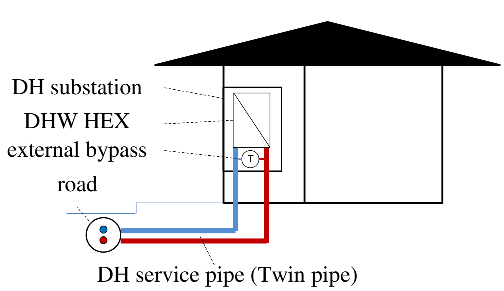

Fig. 1. Principle of bypass valve installed at in-house DH substation.

Our investigations compared the energy performance of two bypass operations. The first case is the theoretical situation with an "ideal" thermostatic valve without a deadband ($\Delta T_{DB} = 0$) 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 realistic "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 [2]. The Matlab® code modelling the DH service pipes developed in Refs. [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 Fig. 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 Fig. 3. The model of the DH pipes in the heat transfer simulations was built in accordance with the methodology explained in Ref. [9].

We modelled both ideal and pulse bypass operation with code developed in Ref. [8]. 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 of pulse bypass at a low-temperature DH substation connected to the DH network at the Danish Technological Institute, but even using DH energy metres 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

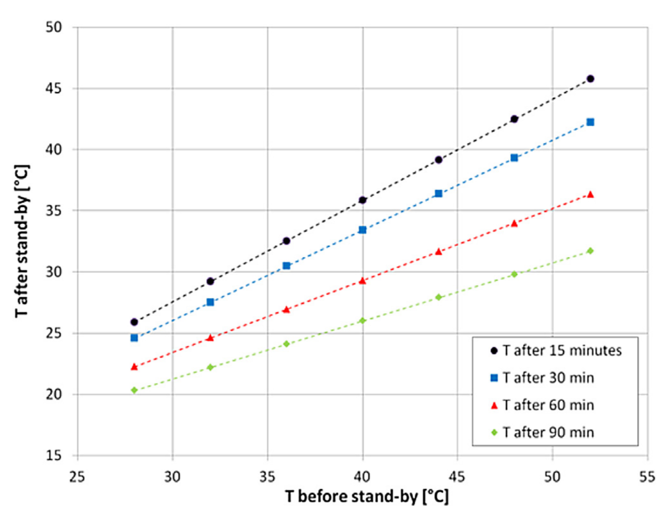

Fig. 2. Examples of cooling-off curves derived from 2D transient heat transfer simulations. $T_{soil} = 8$ °C. Service pipe: Aluflex 20-20/110.

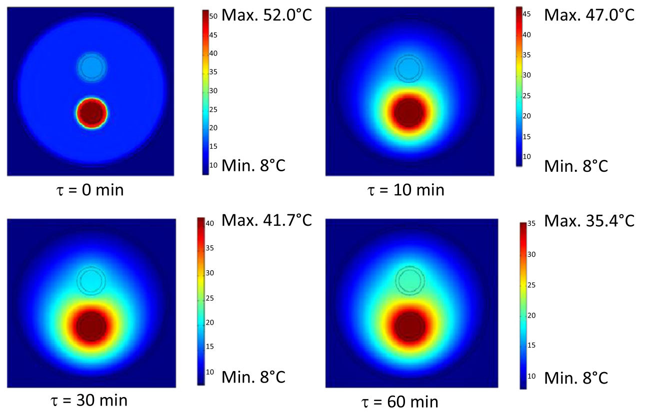

Fig. 3. Cooling of the supply media pipe in the service pipe during the stand-by period. At $\tau = 0$ s: $T_{\text{soil}} = 8$ °C, $T_{\text{PUR}} = 15$ °C, $T_{\text{return}} = 20$ °C. Service pipe: Aluflex 20-20/110 – the numbers referring to the diameters of the supply pipe, the return pipe, and the casing of the twin pipe.

could not be used and the pulse operation of the thermostatic bypass was not documented.

The DH supply temperature expected at the beginning of the service pipe in the non-heating season during periods without DHW tapping will differ based on the location in the DH network. The further down the DH network the house is situated, the lower the temperature at the inlet to service pipe. To reflect this phenomena we modelled both types of bypass operation at three locations in the DH network (see Fig. 10), specified by the inlet temperature to the service pipe 50 °C, 40 °C and 37.3 °C. The individual locations can be seen as the users being situated in close, middle and far distances from the DH heat source. 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 pulse (intermittent) bypass in both locations operates repeatedly approximately every 15 min. 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 approximately 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, e.g. bypass realised by using a small needle valve. However, this consideration does not take DHW tapping into account, which is very important as explained later in Section 3.2.2.

To improve the energy efficiency of the DH network, it might therefore 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 CB (comfort bathroom) 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 surface temperature in the bathroom during non-heating period is only 22.4 °C. Based on Ref. [11], the floor surface temperature preferred in bathrooms is 28 ± 0.3 °C, and this justifies the option of increasing floor surface temperature comfort in the bathroom.

It is true that a similar effect can be achieved by an FH loop controlled with a traditional TRV (thermostatic regulation valve) [12] (thermostatic valve controlled by indoor operative temperature) modified for use in FH loop or an FJVR (TRV controlled by the fluid temperature) valve [13] (thermostatic valve controlled by temperature of fluid), but in comparison CB utilises 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 during non-heating season as FH controlled by TRV or FJVR, but to utilise part of the heat from bypass operation being otherwise lost in the DH network to heat up the floor in bathrooms at discounted price.

3.1. Technical solutions for CB

FH in the bathroom can be controlled either by an FJVR valve, or by a TRV with self-acting thermostatic sensor (e.g. Refs. [12,13]) or by an electronic actuator (e.g. Ref. [14]) with wired/wireless remote temperature sensor (not shown in Fig. 4). The first two solutions are 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.

Fig. 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. Fig. 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 CB solution designed for this setup could have high potential for realisation.

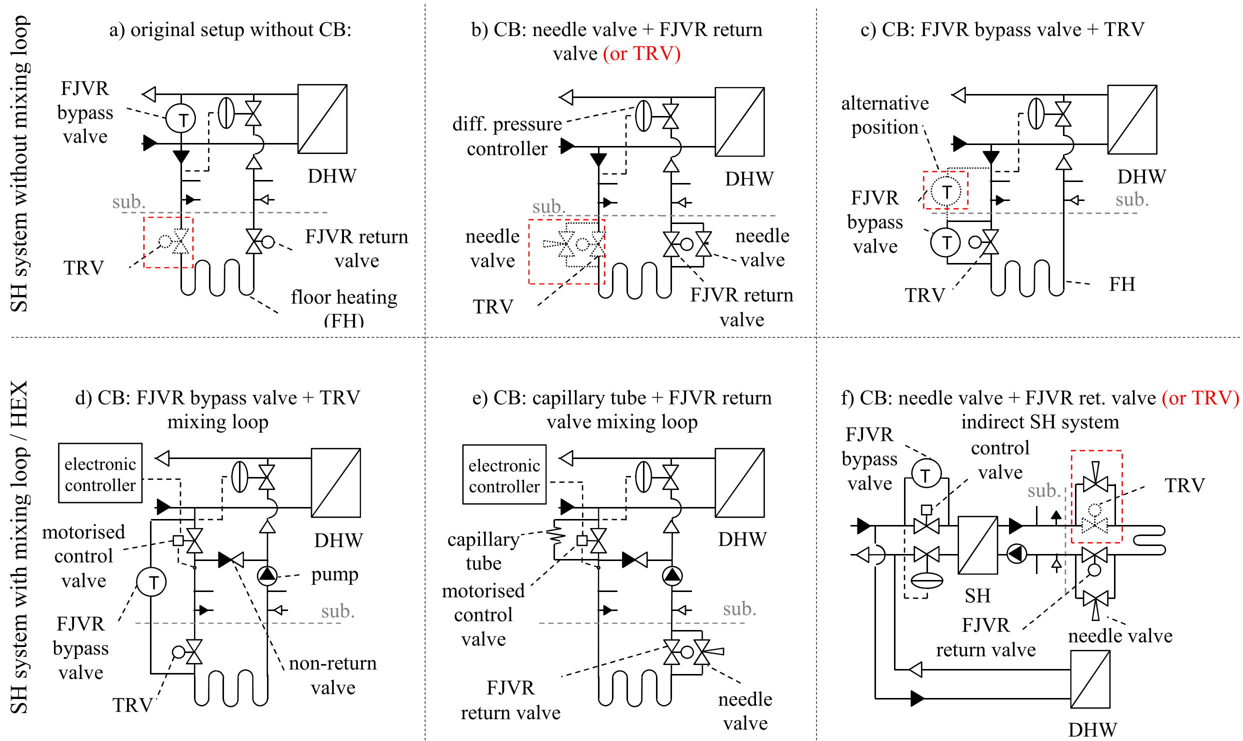

Fig. 4. Technical solution for CB implementation. **Direct SH system without mixing loop**: a) reference case without CB, with traditional external bypass; b) CB realised with a needle valve (installed in parallel to TRV valve on supply pipe or in parallel to FJVR valve on the return pipe of FH loop); c) CB realised with a FJVR bypass valve. **Direct SH system with mixing loop**: d) CB realised with a FJVR bypass valve; e) CB realised with a needle valve and capillary tube. **Indirect SH system**: f) CB realised with FJVR bypass valve and needle valve.

3.1.1. SH systems without mixing loop

3.1.1.1. 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 Fig. 4b) keeping the constant bypass flow through the bathroom FH or by installing a needle valve on the supply pipe in parallel with the existing TRV valve. 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 or TRV valve. 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), happening during non-heating period after every DHW tapping and filling the service pipe with 50 °C warm DH water. This means that the CB will be supplied with water with a temperature above 35 °C for about 45 min after DHW tapping, resulting in higher energy use possibly leading to overheating and also higher return temperature, which is undesirable.

3.1.1.2. 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 installed in parallel with the TRV valve (see Fig. 4c). The FJVR as an external bypass valve redirects a small DH flow to the bathroom FH, but when the temperature of this water increases above the bypass set-point, the valve closes. This situation occurs during the non-heating period after each DHW tapping, but it also means that the CB flow through the bypass valve will automatically shut off during a heating period. However, CB realised using an FJVR valve is applicable only in FH loops controlled by a TRV valve, because the FJVR valve in function of the bypass flow controller must be installed before the FH loop.

The location of the FJVR valve controlling the bypass flow depends on the location of the TRV valve. For bathroom FH loops with an electronic actuator controlled remotely by the bathroom operative temperature and located in the substation (not shown in Fig. 4c), the FJVR valve is installed in the substation in parallel with the TRV valve. However the TRV valve should be relocated from its usual position at the return pipe of the FH loop (collecting manifold) to the supply pipe of FH loop (just after the splitting manifold).

For the FH loop controlled by a traditional TRV valve located in the bathroom, the FJVR valve can be installed either in the bathroom or in the substation. Installing the FJVR valve in the substation needs additional supply pipe between the substation and bathroom. The pipe should be properly insulated (e.g. Twin pipe - two media pipes embedded in one casing pipe — as used for the service pipe connections) to limit the heat loss from the pipes as much as possible. In this case, the set-point temperature of the FJVR is very close to the desired standby temperature because the FJVR valve is installed directly in the substation. The alternative solution (without needing additional pipe) is to place the FJVR valve directly in the bathroom, but then the temperature drop of the bypassed water before reaching the FJVR valve is influenced by conditions in the house, resulting in non-optimal performance.

It should be mentioned that the use of a traditional FJVR valve with a deadband (and thus pulse operation) instead of a needle valve would prevent the supply of extra heat after DHW tapping, but on the other hand it will eliminate the advantages of reduced flow/heat loss achieved by a needle valve as discussed in Section 2.1.

3.1.2. SH systems with mixing loop (direct SH) or heat exchanger (indirect SH)

Fig. 4d–f shows the implementation of CB in the substations which can control the supply temperature SH system. CB for directly connected systems with a mixing loop is realised by "bypassing" the main SH control valve. In the case of FH controlled by a TRV valve (see Fig. 4d), the bypass flow passes through an FJVR valve and is returned to the original FH pipe just after the TRV valve. In the case of an FH loop controlled by an FJVR valve (see Fig. 4e) the needle valve is installed in parallel and the main SH control valve is bypassed with a capillary tube. An alternative solution could be to use a main SH control valve (electronic step valve) as a bypass valve to keep the substation on the standby temperature, but the question is: Can the valve and the circulation pump control such a small flow and is the valve fast enough to close in the case of DHW tapping to prevent the supply of 50 °C DH water into the FH loop?

Fig. 4f shows the implementation of CB for indirect SH systems realised by bypassing the main SH control valve with an FJVR valve, however the main SH control valve should be located before the SH HEX (heat exchanger). The main SH control valve is in fact usually situated at the return pipe of the HEX, i.e. in the position defined historically by the need to protect the control valves from high supply temperatures, which is not necessary from the perspective of low-temperature DH. The change needed on the secondary side (the in-house SH system) is the additional installation of a needle valve in parallel with the FJVR or TRV valve. In this case, the SH HEX is kept continuously at bypass standby temperature, but this is not seen as a problem, because the HEX is expected to be insulated.

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 BR10 (Building Regulation 2010) [16]. This means that the energy needed for SH, DHW heating and the operation of HVAC systems should be below 37 kWh/(m^2.a) after accounting for primary energy factors. The house is described in more detail in Ref. [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 DRY (design reference year) 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 h a year.

3.2.2. Needle valve and FJVR bypass valve

As mentioned above, from the on-site measurements we could not confirm whether the FJVR valve operates in continuous or pulse mode because of the low resolution of flow metres. So we modelled FJVR valve in both pulse and continuous operation 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 min 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, the FJVR valve was still considered as an ideal controller with no deadband, but this time with operation in pulse mode. Finally we considered a realistic FJVR valve with a 3 °C deadband, resulting in pulse operation and an increase of bypassed volume of roughly 50%, i.e. 2.6 kg/h. The same cases were modelled also for a second location in the network, 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.

Fig. 5 compares the floor surface temperature T_{floor} and the temperature of water returning from CB T_{ret} for bigger bathroom with installed CB solution for all four modelled cases, in both investigated locations in the DH network. For the sake of comparison, the floor surface temperature for the bigger bathroom, but without CB concept, is also shown. The values in Fig. 5 are reported in time steps of 3.6 s. The cases can be divided into three groups. First, the reference case without CB (no markers), second the case defined as continuous flow of 1.77 kg/h (lower half of the figure), and third the case with continuous flow of 4.68 kg/h (upper half of the figure).

It can be seen that for the case of 1.77 kg/h the maximal difference in T_{floor} (see Fig. 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 and the maximum differences in T_{ret} (see Fig. 5b) are 0.3 °C and 0.7 °C, respectively. Moreover, the curve of T_{ret} also documents difference between the FJVR valve with the ability to stop the bypass flow when the temperature of bypassed water increases above the desired set-point and the needle valve missing this feature. While the T_{ret} for CB realised with needle valve increases slightly after each DHW tapping (see Fig. 5b – triangle markers, DHW tapping performed at time $h = 4488$), the T_{ret} for all cases realised with FJVR valves decreases because the flow to the bathroom is stopped during and for 45 min after the DHW tapping. Comparing pulse and continuous flow of bypassed water, the pulse operation of the FJVR valve (caused by the deadband) can be seen as an increase of T_{ret} every 15 min (square markers). Nevertheless the T_{floor} and T_{ret} are comparable with the case of ideal FJVR valve (circle markers) modelled without the deadband as the continuous flow. Increase of the bypass flow by 50% (diamond markers) follows the same pattern and increases the T_{floor} and T_{ret} accordingly.

Based on Fig. 5, it can be concluded that the difference in performance of 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 (missing in the case of needle valve), is more important. Pulse operation of the FJVR valve makes no noticeable changes in T_{floor}, because the heat delivered in pulses is averaged by the thermal mass of FH. The difference in weighted averaged T_{ret} is also negligible.

This finding allows us to model the FJVR valve with continuous flow, no matter if the bypass flow is in reality pulse or continuous. Modelling the valve with continuous flow furthermore saves computational time because the simulation of pulse bypass needs time steps small enough to catch the nature of intermittent bypass (around 30 s) while in some situations time steps for other heat

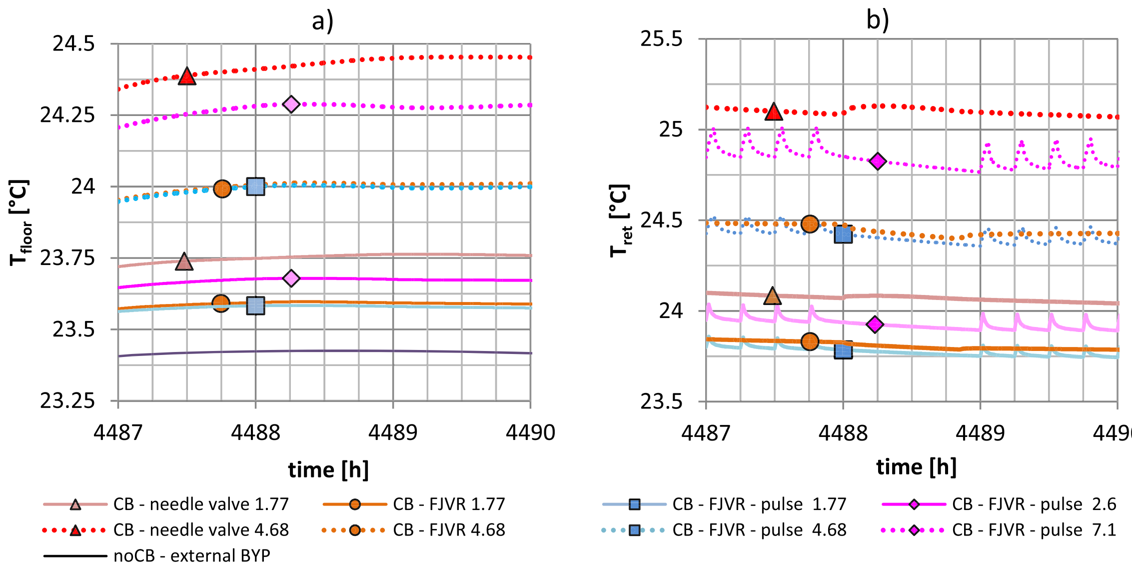

Fig. 5. Comparison of: a) T_{floor} and b) T_{ret} in CB for two locations in the DH network modelled with four different control strategies of bypass flow, output for results in steps of 3.6 s.

transfer phenomena in buildings can be increased to steps ranged in hours. Moreover, output data reported every 30 s 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 min 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-min period of DHW tapping every 3 h (daily between 6:00-24:00), which means that the temperature in the service pipe is 50 °C for 5 min during DHW tapping and then drops linearly for 45 min 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 controlled by an FJVR valve (installed on the return pipe of the FH loop), the FH loop is supplied by bypass water linearly decreasing from 50 °C to 35 °C for a period of 45 min.

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.

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. 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 temperature to the substation at 35 °C, the traditional external bypass opens to keep the inlet temperature to the substation at 35 °C. 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 Ref. [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 bypass valve installed in

Table 1
Matrix of simulated cases.

Case	Abbreviation	T_{sup} to CB > 35 °C	Flow [kg/h] 1.77	4.68	9.36
1	noCB – FJVR	Yes	✓	✓	✗
2	noCB – external BYP	No	✓	✓	✓
3	CB – needle valve	Yes	✓	✓	✓
4	CB – FJVR	No	2.6	7.1	14.0
5	CB – electr. step valve	No	✓	✓	✓
6	CB – FJVR + mixing loop	No	✗	7.1	✗

✓ – Simulated, ✗ – not simulated.

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 min (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 resulting in 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 ideal 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_{op} and floor surface temperature T_{floor} reported with time steps of 0.3 h and, if not specified, only for the bigger bathroom.

4.1.1. Heating period

Fig. 6 compares the performance of FH in the bigger bathroom for one week during a heating period. This is directly connected FH without a mixing loop controlled by a TRV valve with a set-point $T_{op} = 20$ °C (square markers), an FJVR valve with a set-point temperature of 25 °C (sensing temperature of water returning from FH loop, triangle markers), and FH with a mixing loop controlled with TRV valve (circle markers). It can be seen that for the FH loop controlled traditionally by an FJVR valve, T_{op} is constantly around 21.5 °C and T_{floor} around 23.5 °C, while for the TRV valve the values in both cases are 20.5 °C and 21.5 °C, respectively. The average T_{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_{ret}.

The heat demand for both bathrooms is 90% higher for the reference case with FJVR return valve during heating season than for the case with TRV, as a result of T_{op} being higher by 1.1 °C on average. But most of the additional heat is recovered by the heat recovery of the ventilation system, so the overall additional heat demand of the house with FH controlled by FJVR return valve is during the heating period 93 kWh, i.e. 4%.

4.1.2. Floor heating without CB

Fig. 7 shows results for a two-day period during the non-heating season for bathrooms in two locations in the DH network. The figure compares T_{floor} and T_{op} for the reference case without FH heating (Case 2 – black diamond markers), the case with FH traditionally controlled by an FJVR (Case 1 – grey diamond markers), and for the sake of comparison also with the CB concept realised with an FJVR valve (Case 4 – circle and square markers). 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 return valve in the traditional manner results in the highest T_{op} and T_{floor}. As can be seen, the curves for both locations are nearly identical and fall on top of each other, 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 FJVR valve 25 °C 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 (curve without markers). In this case the window is opened for 415 h, i.e. 8% of the non-heating period, despite the building's

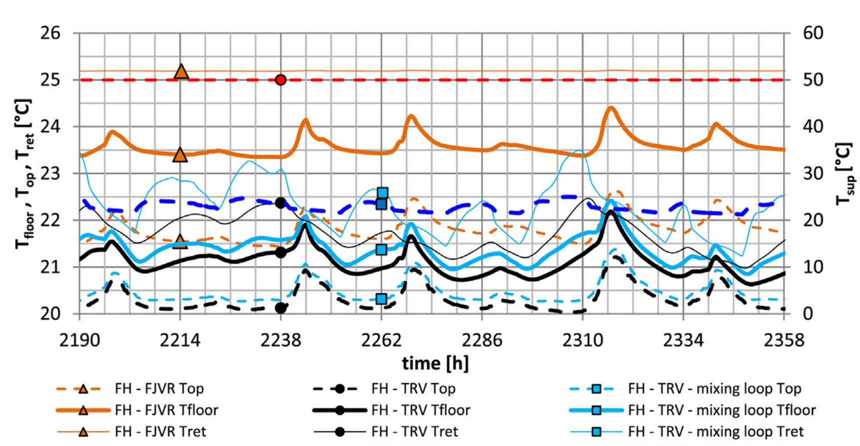

Fig. 6. Performance of FH in the bathroom during heating season.

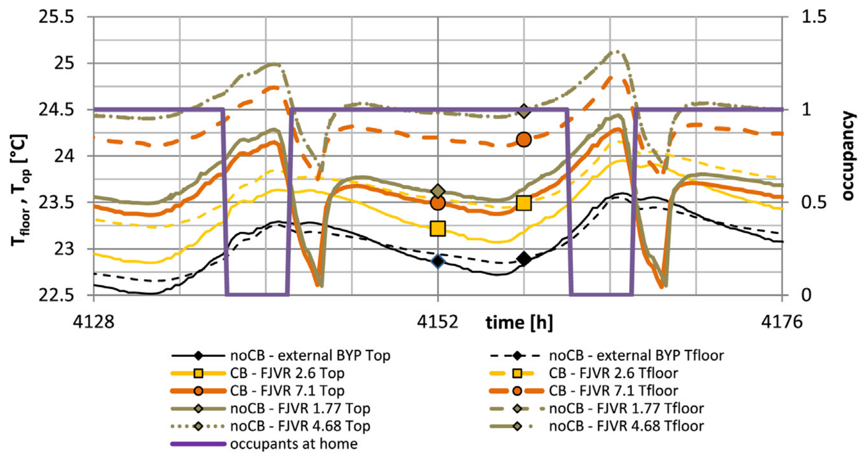

Fig. 7. T_{op} and T_{floor} in bathroom during non-heating period, considering various control of floor heating system.

location in the DH network. However the flow of DH water required by the FJVR valve with the set-point temperature of 25 °C is not high enough to keep the inlet of the substation at 35 °C for the whole non-heating period. In case of customer located close to the heating plant the flow is below required value of 1.77 L/h for 3% during non-heating period and for the customer located in the middle of DH network the flow is below 4.68 L/h for 17% of non-heating period. The results documents that FH controlled with FJVR return valve installed on the return pipe 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} in the bathrooms with implemented CB solution located in close and middle distances from the DH heating plant 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 to improve thermal sensation of the occupants. Results for all cases are summarised in Table 2.

4.1.3. Floor heating with CB

Fig. 8 shows T_{floor} for a two-day period during a non-heating season for the CB solution in three locations in DH network (close, middle and far distances from DH heat source) realised with a needle valve and an FJVR bypass 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 Fig. 8) depend 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_{floor} for CB realised with a needle valve and an FJVR with a deadband in the same locations in DH network, T_{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% from original 50% (see Table 2). Similar can be concluded also for CB realised in the system with a mixing loop.

Fig. 9 also shows T_{floor} 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 bypass valve with a reduced deadband resulting for the same locations in nominal 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

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

	Case#	Nominal bypass flow [kg/h]	T_{op} avg. [°C]	T_{floor} avg. [°C]	T_{ret} avg. [°C]	Bypassed volume [m^3]	Average heat output from FH [W]	Energy used in FH [kWh]	Heat demand incl. FH [kWh]	Increase of heat demand [%]	CB cost for customer [DKK]	Bypass cost for DH company [DKK]
Location 1	1	1.77	23.5	24.4	25.2	24.3	97	500	2945	17%	325	−49
	2	2.6	22.4	22.4	35.0	9.7	0	0	2352	0%	0	110
	3	1.77	23.0	23.0	23.3	9.1	32	165	2518	7%	107	−3
	4	2.6	22.7	23.0	23.2	9.7	29	150	2502	6%	98	12
	5	1.77	22.9	22.9	22.9	6.9	22	116	2467	5%	75	3
Location 2	1	4.68	23.5	24.4	25.2	32.6	96	496	2943	17%	322	−199
	2	7.1	22.4	22.4	35.0	25.7	0	0	2348	0%	0	97
	3	4.68	23.2	23.8	24.4	24.1	72	373	2721	14%	242	−151
	4	7.1	23.2	23.7	24.3	25.7	64	333	2681	12%	216	−119
	5	4.68	23.1	23.3	23.8	16.8	46	236	2586	9%	153	−90
	6	7.1	23.2	23.7	24.4	25.7	63	323	2697	12%	210	−119
Location 3	2	14.0	22.4	22.4	35.0	50.9	0	0	2348	0%	0	89
	3	9.36	23.5	24.8	25.8	48.3	127	655	2996	22%	425	−341
	4	14.0	23.4	24.6	25.6	50.9	111	571	2919	20%	371	−283
	5	9.36	23.3	24.1	24.8	34.0	81	418	2766	15%	272	−213

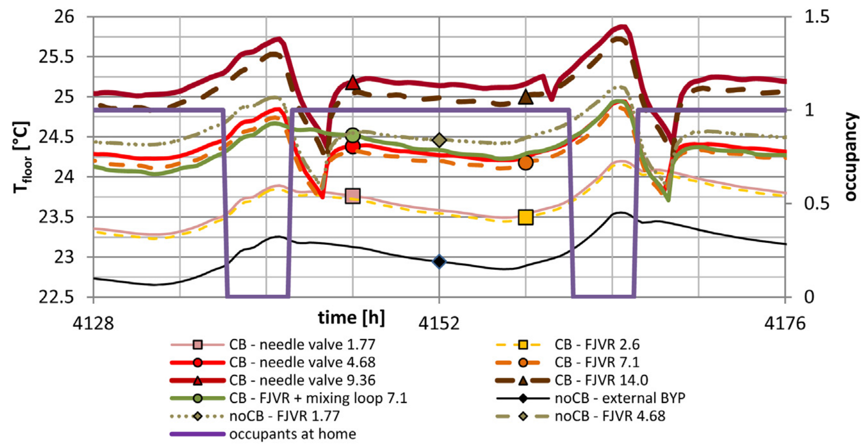

Fig. 8. T_{floor} during a non-heating period in a bathroom with CB realised with a needle valve or an FJVR valve.

blocking function when the bypass water is above 35 °C, T_{floor} is reduced by 0.1 °C, 0.25 °C and 0.25 °C in comparison to CB realised with a needle valve (or a FJVR bypass valve with a deadband as shown in Fig. 8).

4.1.4. Comparison of all cases investigated

Table 2 summarises 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 into account the volume of the DH water cooled during the DHW tapping. Taking the DHW production corresponding to 3200 kWh/(person.a) into account with designed 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 min).

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_{op}, 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 utility for bypass operation in the 10 m long service pipe during the non-heating season. The cost of the bypass operation is calculated as the cost of heat lost in the service pipe during bypass operation (heat loss = mass of bypassed water * water thermal capacity * temperature difference at the beginning and the end of the service pipe) minus the heat used in CB, which is paid for by the customer. We are assuming a heat price of 650 DKK/MWh (currency exchange rate 1 DKK = 0.13 EUR). The cost calculation takes into account the different temperatures at the beginning of the service pipe for different locations in the DH network (namely 50 °C, 40 °C and 37.3 °C) and 35 °C at the end. The results show that the heat sold for operation of CB during the non-heating period covers, in most of the cases, the running cost of bypass, while in the case of traditional bypass operation it is a DH utility who pays for the bypass operation.

Although the CB realised with an FJVR valve (case 4) results in a slightly higher volume of bypassed water than with a needle valve (case 3), 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 CB realised with 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

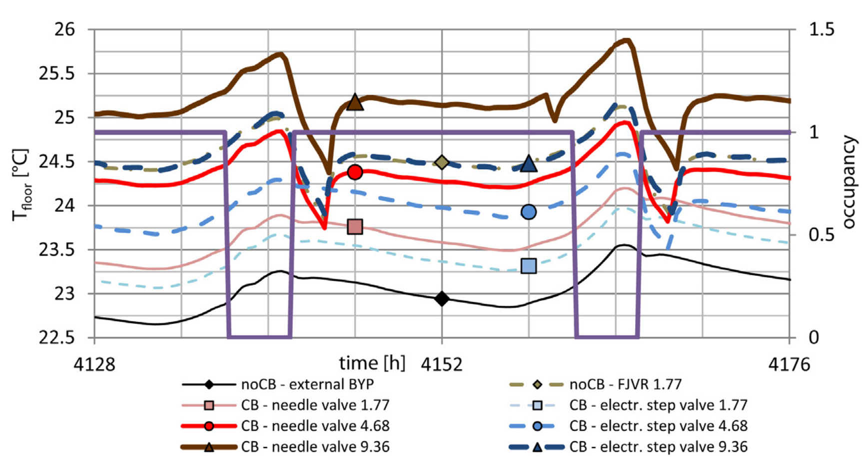

Fig. 9. T_{floor} during a non-heating period in a bathroom with CB realised with a needle valve or electronic step valve.

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 (case 2) to the CB results in same bypassed volume, but in the average return temperature 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.

In the case of FH traditionally controlled by an FJVR return valve (case 1), having the whole year adjusted to a set-point of 25 °C will sometimes result in higher flow than needed to keep the service pipe ready for use in the non-heating period. On the other hand, for some periods the flow rate required by FH in both bathrooms will be not high enough to keep the inlet of the substation at the desired temperature, and it will call for the traditional external bypass anyway. This solution means the customer will get higher T_{op} and T_{floor} but will also pay for a 17% increase in heat demand compared to the 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 (case 5) with a reduced dead-band. 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 depends on their location in the DH network and for the case realised with an FJVR valve (case 2) 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 and 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 carried by the bypass flow 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 Ref. [3] was chosen for a case study and the results are reported in Fig. 10 and Table 3. It is important to stress that although the DH network layout was adopted from Refs. [3], the investigation was made on the reference house described in Section 3.2.1 and not on the houses built in Ref. [3].

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_{op} and reduced cooling of bypassed water. Results for this simulation are

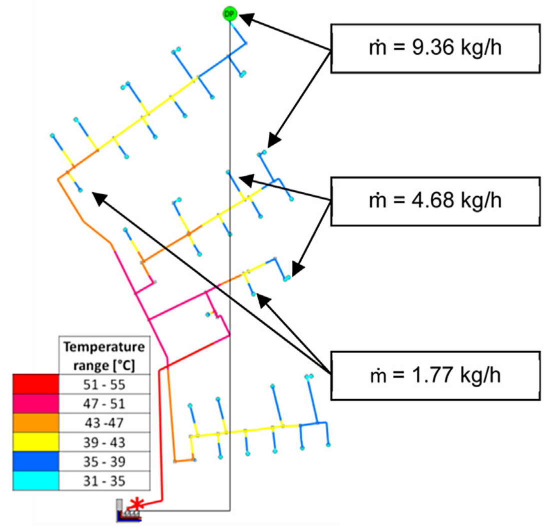

Fig. 10. Plot of the supply temperature in the network with continuous bypass flow and the "CB concept" during the non-heating season. The $\Delta T_{supply\text{-}return}$ in the FH was set to 8 °C and flows needed to keep a constant 35 °C at the bypass.

therefore on the conservative side. As a consequence of installing CB, 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) and the average return temperature at the DH heating plant was reduced from 27.7 °C to 23.8 °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% (see Table 3). However the table also 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%. Nevertheless lower return temperature requires additional energy in the DH heating plant to heat up the DH water back to 55 °C, i.e. in our example 0.7 kW. However at the same time, the customers use in total 1.2 kW of heat in their CB. This additional heat demand is from 40% covered by reduced heat losses in the DH network (0.5 kW) and therefore the DH utility should bill the customers only for the remaining 60%. Furthermore in case of the DH heat sources profiting from the reduced return temperature as e.g. combined heat and power production or heat plants with condensation of flue gases the price of the heat used in CB during non-heating period could be further reduced. The same is also valid e.g. for solar-

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

	Case		Difference
	A	B	
Heating power [kW] (plant)	3.8	4.5	18%
T_{supply} [°C] (from the plant)	55	55	–
T_{return} [°C] (to the plant)	27.7	23.8	−3.9 °C
Additional heating power [kW]	–	0.7	–
Total heat loss DH network [kW]	3.8	3.3	−13%
Heat loss supply [kW]	2.25	2.3	2%
Heat loss return [kW]	1.55	1	−35%
Total heating power in all CB [kW]	–	1.2	–
Heat in CB Covered by Customers [kW]	–	0.7	58%

thermal plants, where the lower return temperature mean increased thermal efficiency. To evaluate economic value of reduced return temperature for the heating plants is however not in scope of this article.

4.3. Cost-efficiency

Considering the fact that the customers pay only 60% of heat used in the CB the example of a CB controlled with an FJVR bypass valve with a 3 °C deadband and operated during the non-heating period costs the customer in the house located at the outskirts of DH network 220 DKK, while close to the main line it will only cost 60 DKK. It is true that T_{floor} for the customer located at the outskirts of the network is on average 1.6 °C higher than for the customer located close to the plant, meaning greater comfort, 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 possible further savings in the heat production based on reduced DH return temperature.

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

In Denmark, DH 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.

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 in the CB solution. 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, corresponding to 40% of the SH demand in the bathrooms during the summer. Considering this the cost of CB during non-heating season would be 60–220 DKK, depending on the location of the customer in the DH network, but we suggest an equal charge for all customers whatever their location. 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.

Application of the CB concept in the typical medium temperature DH network built from the single pipes without state-of-the art insulation properties will result compared to the low-temperature DH network in increased bypass flow needed to keep inlet to the substation at 35 °C, but at the same time in a possibility to save more heat from the return pipes due to the worse insulation properties of the DH network. From the perspective of the customers higher bypass flow mean more heat available in the bathroom FH and thus higher comfort for discounted price.

Use of the CB concept is expected to be more beneficial in traditional buildings then in low-energy buildings, as the floor surface temperature will be lower and cooling of bypassed water 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.

Acknowledgements

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ISI paper III

Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment

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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 DH (district heating) 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 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 (domestic hot water) 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.

1. Introduction

To reduce CO_2 (carbon dioxide) 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]. DH (district heating) 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 DHW (domestic hot water) 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 have 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

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houses are equipped with radiators designed for 50/25 °C supply/return temperatures. The 50 °C supply temperature to the SH (space heating) 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 CO_2 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 focussing 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 an 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 an 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 to 45 °C, the substation provides cooling on the primary side from 50 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 30–50 °C. Most of the national DHW standards therefore require 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 3 L" (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 3 L 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 3 L, the piping can be changed from 1/2″ 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 3 L requirement can be fulfilled by replacement of the original substation with a 120 L storage tank for DH water [16]. The principle is shown in Fig. 4. The storage tank acts as a 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.

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 3 L. 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

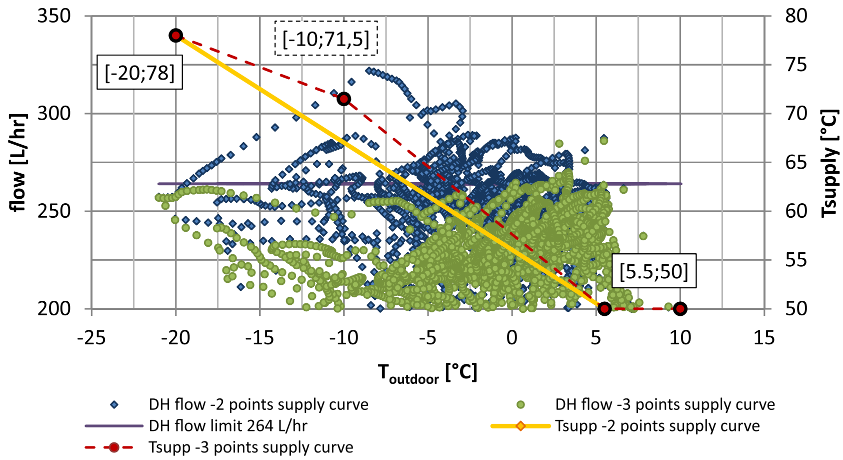
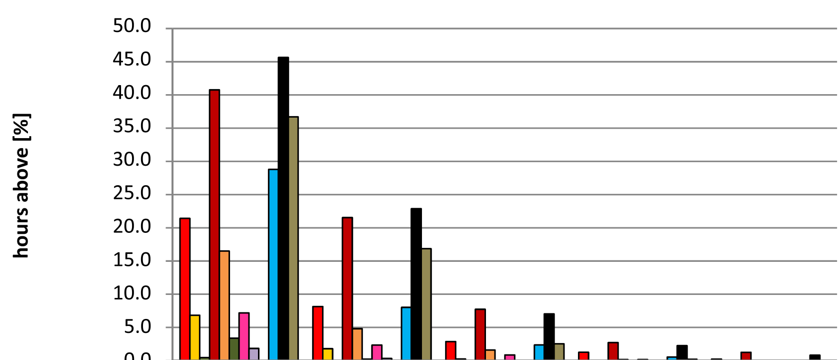


Fig. 1. Construction of weather-compensated supply temperature curve for non-renovated house, set-point temperature 22 °C.

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 AOT (Advanced Oxidation Technologies), where UV (ultraviolet) 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 Ref. [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.



	50	55	60	65	70
non-renovated 20°C	21.4	8.1	2.9	1.3	0.2
new windows 20°C	6.8	1.8	0.2	0.0	0.0
extensive renovation 20°C	0.4	0.0	0.0	0.0	0.0
non-renovated 22°C	40.8	21.5	7.7	2.7	1.2
new windows 22°C	16.5	4.8	1.6	0.1	0.0
extensive renovation 22°C	3.4	0.2	0.0	0.0	0.0
non-renovated 22°C LT radiators	7.2	2.3	0.9	0.2	0.0
new windows 22°C LT	1.8	0.3	0.0	0.0	0.0
extensive renovation 22°C LT	0.0	0.0	0.0	0.0	0.0
non-renovated 22°C HIGHFLOW	28.8	8.0	2.4	0.5	0.0
non-renovated 22°C Tout SHIFT	45.6	22.9	7.0	2.3	0.8
non-renovated 22 °C ODUM	36.7	16.9	2.5	0.2	0.0

T_{supply} [°C]

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

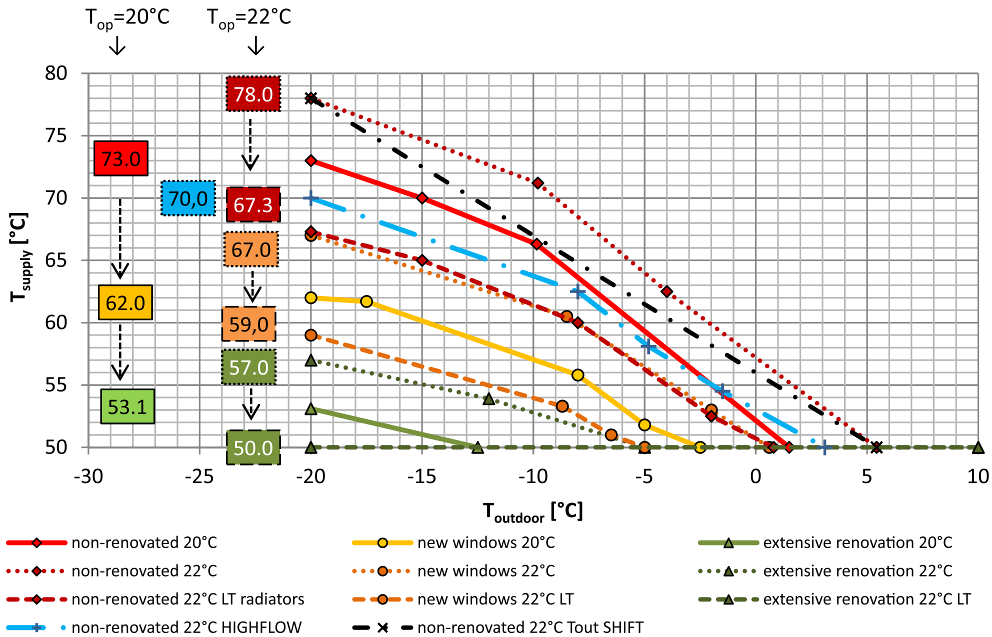


Fig. 3. Supply temperature curves for all the investigated cases.

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 157 m^2. 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, DRY (Design Reference Year) for Denmark.

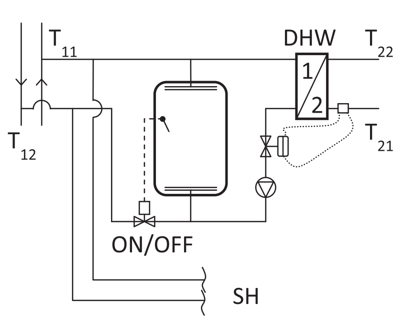


Fig. 4. Principle of low-temperature DH substation with buffer tank for DH water [16].

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 temperatures), 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/m^2 [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 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 9.2 kW. 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). 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 9.4 kW and the maximum water flow rate to 280 L/h (see Table 1). An increase of 0.2 kW 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 (4.18 W/m^2) 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.

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

$T_{operative}$ [°C]	Internal heat gains [W/m^2]	T_{supmax} [°C]	RAD[a]	P_{max} [kW]	T_{RW} [°C]	Heating demand [MWh/a]	P_{max} reduction [%]	Heating demand reduction [%]
Non-renovated house – basic								
20[b]	0	70.0	O	9.2	37.6	–	–	–
20	0	70.0	O	9.4	40.2	–	–	–
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								
22[c]	4.18	70.0	O	10.5	35	24.5	0	0
22[d]	4.18	78.0	O	10.5	33.1	24.6	0	0
22[e]	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	×	–	–
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	×		
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”.

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 264 L/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 Fig. 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 Fig. 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.

Fig. 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 (Fig. 1 – diamond markers). This situation occurs roughly for 170 h 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. Fig. 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 h), which can be negligible considering the thermal inertia of the building.

Fig. 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.

Fig. 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 h) 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 6 h in the periods when the outdoor temperature was increasing. This case in figures denoted “non-renovated 22 °C T_{out} 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 Section 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 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.

3.1.3.1. Hydronics of the SH system. The standard approach to design 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 0.42 m/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 0.25 m/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 mH_2O); in ½″ pipes, this means a maximum velocity of 0.42 m/s and a total pressure drop of 13.7 kPa (1.37 mH_2O). 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 pressured 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.

Fig. 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 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%–7% of hours in the year (see Fig. 2) and by 94%–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 to 27.6 °C.

3.1.5. One-pipe system

The simulations do not consider one-pipe heating system accounting for ten percent 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 a 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/(m^2 K)) were replaced with “energy-glazing” windows with a U-value of 1.2 W/(m^2 K), resulting in an overall U_w value of 1.5 W/(m^2 K). The new windows also reduce air infiltration by 15% to 0.41 h^{-1} (0.278 L/(m^2 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 Fig. 2) decreases by 60%–16.5% of hours over the year and by 95%–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 mH$_2$O), 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 mH$_2$O), 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 ($\lambda_{ins} = 0.56$ W/(m^2 K) including the effect of wooden beams), and by insulating the wooden beams that bear the roof construction with 125 mm of insulation ($\lambda_{ins} = 0.039$ W/(m^2 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 shows the impact of the individual renovation measures on the annual heating demand.

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 °C and from 24.6 MWh to 12.4 MWh for an operative temperature of 22 °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 to 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 °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 °C is needed only for 3% (295 h) of a year and for just 20 h 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 79 m^3/a of 45 °C DHW (corresponding to 800 kWh/(a person)) [27], which is only 8% of the

Table 2
Impact of individual refurbishment measures on annual heating demand; measured weather data input, set-point temperature $t_{air} = 20$ °C.

	Heating demand [MWh/a]	Heating demand reduction	
		[MWh/a]	[%]
Non-renovated house	18.8	–	–
Improved thermal bridges on windows	18.3	0.4	2
Additional 300 mm insulation to the ceiling	14.4	3.9	21
Reduced air infiltration with new windows from 0.327 to >0.27795 L/m^2 s	13.6	0.8	4
New low-energy windows on W + N, U_w from 3.2 to >0.9 W/(m^2 K)	10.3	3.3	18
Added 125 mm insulation to wooden bearings for the roof construction	9.4	0.9	5

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 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 h) 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 h). 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 h) 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 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 an 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.

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Schedule for Internal Heat Gains

Overview of internal heat gains

List of Internal HG		
equipment / event	continuous power [W]	energy [kWh/24h or cycle]
fridge	32	0.77
washing machine	1000	1
tumble dryer	1850	1.85
dish washer	1000	1
Cooking	1100	1.1
TV	200	
Computer	100	
breakfast (1+2kW in 5 minutes)	250	0.25
Ironing	0.75	
IHEU	36	

addition to routine events [kWh/week]			
event	energy [kWh/cycle]	day	time
washing	1	Tuesday	18-22
drying	1.85	Tuesday	22-01
washing	1	Thursday	18-22
drying	1.85	Thursday	22-01
vacuuming	1	Sat	15-16
ironing	0.75	Sun	15-16
total	7.45	[kWh/week]	

	EQUIP + LIGHTS … difference my SCH vs. constant 3,5W/m2			people…difference my SCH vs. Constant 1.5W/m2		total internal HG	
	EQUIP + LIGHTS: const. 3.5 W/m2	EQUIP + LIGHTS: model	difference	people: const. 1.5 W/m2	people: model	const 1.5 + 3.5 W/m2	model
week [kWh/week]	93.5	71.39	22.1	40.07	40.96	133.6	112.3
const. power [W]	3.5	2.67	0.8	1.5	1.53	5.00	4.21
year [kWh/year]	4862	3712	1149	2084	2130	6945	5842

Detailed schedule of internal heat gains for working day

								HG people [kWh/wd]
heat emitted from people [W/wd]	2560	320	0	320	320	320	1280	5.12
	week days (wd)							
	sleep	eating	no one at home	working	cooking/resting	eating	resting	
room	23:00 - 7:00	7:00 - 8:00	8:00 - 15:00	15:00-17:00	17:00-18:00	18:00-19:00	19:00 - 23:00	
room 11.6	1			1	1		1	
room 11.9	1			1	1		1	
tech. room 9.1								
entrance 4.1								
room 9.1								
bathroom 1								
bedroom 15.0	2							
kitchen 9.4					1			
bathroom 2								
corridor								
living room 32.0					1		2	
dining room 20.1		4				4		total
time for the activity [h]	8	1	7	2	1	1	4	24

heat emitted [W]	week days (wd)						
fridge + IHEU	68						
washing machine							
tumble dryer							
dish washer							250
cooking					1100		
TV					200		200
computer				200	200		200
eating (kettle, toaster)		250					
lights				25	250	150	150

total EQ + LIGHT in the house [W]	545	318	477	586	1818	218	3472	7.4	kWh/wd
					constant value 3.5W/m2			13.4	kWh/wd
					difference			5.9	kWh/wd
					total EQUIP and LIGHT			**29.6**	kWh/5wd

Detailed schedule of internal heat gains for weekend day

										HG people [kWh/we]
heat emitted from people [W/we]	2560	320	1280	320	320	960	320	320	1280	7.68
	weekend (we)									
	sleep	eating	morning	cooking/resting/working	eating	afternoon	cooking/resting/working	eating	resting	
room	23:00 - 7:00	7:00 - 8:00	8:00 - 12:00	12:00-13:00	13:00 - 14:00	14:00-17:00	17:00-18:00	18:00-19:00	19:00 - 23:00	
room 11.6	1		1	1		1	1			
room 11.9	1		1	1		1	1			
tech. room 9.1										
entrance 4.1										
room 9.1										
bathroom 1										
bedroom 15.0	2									
kitchen 9.4				1			1			
bathroom 2										
corridor										
living room 32.0			2	1		2	1		4	
dining room 20.1		4			4			4		total
time for the activity [h]	8	1	4	1	1	3	1	1	4	24

heat emitted [W]	weekend (we)								
fridge + IHEU	68								
washing machine			250						
tumble dryer				1850					
dish washer									250
cooking				1100			1100		
TV			200	200		200	200		200
computer			200	200		200	200		
eating (kettle, toaster)		250							
lights						16.7	250	150	150

total EQ + LIGHT in the house [W]	545	318	2872	3418	68	1454	1818	218	2672	13.4	kWh/we
						constant value 3.5W/m2				13.4	kWh/we
						difference				-0.0	kWh/we
						total EQUIP and LIGHT				-0.1	kWh/2we

This PhD thesis is based on three published scientific papers that describe challenges related to DHW and space heating systems supplied by low-temperature district heating (LTDH) and provide suggestions for their solution. The results indicate that LTDH can deliver the heat required by DHW and space heating systems, but needs specially designed LTDH substations and low-temperature heating systems. In areas with older, non-refurbished buildings with traditional space heating systems, the DH supply temperature will need to be increased briefly during cold winter periods. The thesis concludes that LTDH can be widely implemented in areas with both low-energy and non-refurbished buildings, and thus make an immediate contribution towards the achievement of a fossil-free and cost-efficient heating sector.

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