Enabling space cooling in combined heat distribution circuits by grouping same temperature demands

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Abstract

A 2-pipe collective thermal system is considered suitable only for space heating and Domestic Hot Water (DHW) distribution. However, this study investigates through dynamic simulations the extent to which space cooling (SC) is possible by cleverly using decentralised DHW storage to distribute time-dependent supply temperatures, without local boosters. The results show an 90,8% reduction in indoor temperature discomfort compared to the traditional 2-pipe system (without SC). Compared to a 4-pipe system, cooling and heating production increases by 8,2% and 19,7%, respectively, due to large shares of dissipated heat and cold. SC in 2-pipe systems has been shown to be possible, but heat and cold dissipation needs to be minimised to exploit more potential.

Highlights

- Time-dependent T_{sup} that cleverly uses decentralised DHW storage enable space cooling in 2-pipe collective thermal systems.
- To avoid unintentional heating – thus additional cool loads – in central changeover systems, use a smart valve for the emitter that shuts off when $T_{sup} > T_{zone}$ in cooling mode.
- About 11,4% of cooling and 28,7% of heating must be dissipated to change between SC and DHW mode, which could be stored for later use.

Introduction

Collective thermal systems are important to achieve the Climate Goals by 2030 and 2050 (European Commission, 2019). They facilitate the use of renewable energy sources, such as heat pumps (HP), to decarbonise the heat supply in buildings (Lund et al., 2010). In literature, the benefits of collective thermal systems are mostly discussed for district heating and cooling. However, the use of collective thermal systems at building level offer the same benefits (De Pauw et al., 2018). A Combined Heat Distribution Circuit (CHDC) is such a collective heating system in apartment buildings that supplies the heat for both space heating (SH) and domestic hot water (DHW) with only one supply and one return water pipe (Jacobs et al., 2022).

The study of he European Commission (2022) reports that the relative shares of DHW and space cooling (SC) in the total energy demand of the building stock is rising in Europe and that the cooling market increases by 1 to 2% a year. However, the single supply pipe in CHDCs does usually not allow SC to be delivered to end-users, due to conflicting temperatures for SC and DHW. Since end-users need to be able to use DHW at all times, the supply temperature set point ($T_{sup;\,SP}$) in CHDCs is usually set to 65°C (Rebollar et al., 2017; Jacobs et al., 2023). There are two main options for providing SC in collective systems, namely the use of booster heat pumps (BHP) in CHDCs or 4-pipe systems.

On the one hand, various studies have been carried out on the integration of BHP in the collective heating networks (e.g. Thorsen et al., 2021; Zhu et al., 2021) and on optimising the overall efficiency and recuperating energy (e.g. Jacobs et al., 2021; Hermansen et al., 2022). There are two variants of BHPs. Either they only boost the central temperature to the DHW temperature, or they can additionally convert the central temperature to the temperature for SH and SC in each dwelling.

Despite the possibility of decoupling the central supply temperature (T_{sup}) and available temperatures in dwellings by a BHP, there are three main drawbacks. First, as each dwelling is equipped with a BHP, there is more refrigerant throughout the system, increasing the risk of leaks. Therefore, each BHP requires maintenance, resulting in higher maintenance costs than fully centralised production. Second, BHPs lead to lower energy costs, but the savings must be greater than the initial investment costs, which is currently not always the case (Østergaard and Andersen, 2018). BHPs consist of a heat pump and a storage vessel, which makes them currently more expensive than conventional heat interface units. Finally, the control of BHPs is at the local level. However, more Energy-as-a-Service Companies (ESCOs) are investing in building heating and cooling systems in Europe. They invest in building or renovating the thermal production and distribution systems and provide control strategies. As they continue to control and improve the system, they will realise energy savings compared to the previous situation, which is important for their business model. In this regard, they do not have access to control in dwellings (e.g. from BHPs), which limits their options. Therefore, fully centralised control in heating systems is preferred.

On the other hand, 4-pipe systems consist of two separate distribution circuits, one for SH or SC and one for DHW, where control and production are fully centralised. However, they require more pipes and the DHW

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recirculation loop leads to higher losses. For these reasons, this paper focuses on enabling SC in a 2-pipe CHDC without using decentralised BHPs. Jacobs et al. (2022, 2023) presented a new methodology to control the T_{sup} in a CHDC with decentralised storage. It was shown that grouping high- temperature (i.e. DHW) and low-temperature (i.e. SH) demands reduced the energy consumption in January by up to 36%. Moreover, the T_{sup} should be high (i.e. 65°C) only for 12,5% to 25% of the time with time-based control and with sensor-based control between 20% and 90% of the time, depending on the charging flow rate and the size of DHW storage vessels. These small time frames of high distribution temperatures for DHW suggest that SC could be possible in summer with such control strategies, as the DHW demand is relatively independent of the season. However, distributing space cooling and DHW with only one pipe (i.e. a central changeover system) is more challenging to ensure thermal comfort, as the temperature for DHW is not usable for SC and vice versa, while the temperature for DHW in the previous study is still usable for space heating. Since the state-of-the-art does not provide any control strategies to enable SC in CHDCs that provide DHW supply without booster systems, the contributions of this study are I) to identify the potential of the grouping methodology for heating from Jacobs et al. (2022, 2023) to also be applied to distribute space cooling and II) to determine its energy-related implications.

Focus of research

Case description

Figure 1 gives an overview of the case study used in this research. It involves an apartment building in Belgium with 24 dwellings on 6 floors connected to central production units by the CHDC. On each floor, the dwellings are oriented northeast, southeast, southwest and northwest. The U-value of the walls is 0,24 W/m²K and the windows are double-glazed with a heat loss of 1,2 W/m²K and a g-value of 0,6. Due to different lay-outs, the average solar gains trough the windows vary from 12,7 kWh/day to 15,6 kWh/day. Also, the design heat losses (21°C indoor and −8°C outdoor) range from 2 kW to 3 kW and the floor areas are between 88 m² and 104 m² with a height of 3 m. Each dwelling is equipped with a DHW storage vessel, which stores heat for one shower and two or three other tapping points, and a Fan Coil Unit (FCU) that covers the heating and cooling need in the dwellings. The FCU is designed based on the design heat losses of each dwelling. Recharging of DHW storage vessels has priority over SC, so when a vessel is charging, the SC in that dwelling is switched off. The “profile generator” developed in the TETRA-SWW (VLAIO, 2014) and Install2020 (VLAIO, 2018) projects is used for the occupancy profile (internal heat gains from inhabitants and electric appliances, ($\dot{\boldsymbol{Q}}_{\boldsymbol{int}}$) and DHW demand profiles. Each dwelling is inhabited by a different family, consisting of 1 to 3 inhabitants (De Schutter et al., 2018). The average indoor heat gains range from 3,3 kWh/day to 11,1 kWh/day and the average DHW demand at 58°C is 0,8677 MWh/year/person. In summer, the indoor temperature set point varies with the outdoor temperature. A temperature difference of 5°C between inside and outside is considered comfortable, with a minimum set point of 23°C and a maximum of 27°C. The idealised boiler room can instantaneously deliver the demanded T_{sup} and mass flow rate ($\dot{\boldsymbol{m}}_{\boldsymbol{CHDC}}$). This approach allows to focus on evaluating the control strategy, without the effects of transient behaviour in production units or incorrectly set PID controllers. The central changeover between DHW and SC distribution temperatures is facilitated by a bypass valve between the supply and return pipe.

Demand-based control strategy: 2SC

In this research, the new two-sensor control strategy (2SC) of Jacobs et al. (2023) is applied to investigate the extent to which grouped charging of DHW storage can provide time for SC distribution. This strategy was selected because it is not case-specific (unlike the schedule-based one) and it aims at DHW comfort due to grouping the DHW recharging. For 2SC, two temperature sensors are placed in each DHW vessel, with the upper sensor dedicated to high-priority charging and the lower to low-priority charging. Thus, the upper one initiates a high central supply temperature set point ($T_{sup;\,SP}$) when its temperature falls below the DHW set point and the lower one only charges the DHW vessel when $T_{sup;\,SP}$ is already high. When all storage vessels are fully recharged, $T_{sup;\,SP}$ is reduced to the supply temperature for space cooling in this research. Figure 2 visualises this $T_{sup;SP}$ control strategy. Based on the findings of Jacobs et al. (2023), DHW storage volumes are set at 150 L with a charging flow rate ($\dot{m}_{ch}$) of 300 kg/h and a hysteresis of 4°C, as this resulted in a high temperature for only 27% of the day while securing DHW at all times.

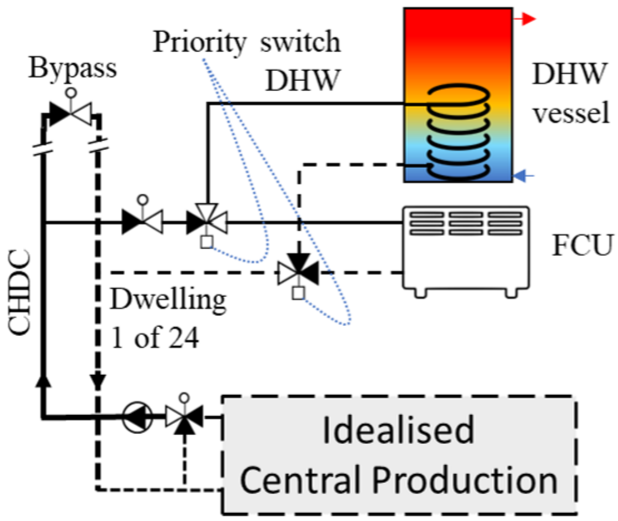

Figure 1: Overview of the case study. All 24 dwellings have a DHW storage vessel and an FCU. At the top of the CHDC’s riser pipe, there is a bypass valve. Recharging of DHW vessels has priority over space cooling.

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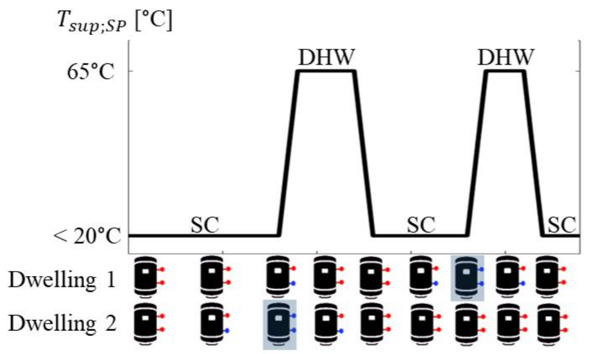

Figure 2: The $T_{sup;SP}$ is 65°C when the upper temperature sensor of one of the DHW vessels drops below (58°C - hysteresis). Then all DHW vessels with too cold lower temperature sensor are recharged. When all vessels are recharged, $T_{sup;SP}$ switches back to space cooling temperatures.

Analysed variants of 2SC

To determine the energy-related implications of distributing SC and DHW in a 2-pipe system, the effects of a smart valve in the FCUs, different control strategies for the bypass valve in the CHDC and different design distribution temperatures for SC are examined.

When high and low temperatures are distributed in the same pipe, it is possible that T_{sup} is high while an FCU intends to cool the dwelling. Therefore, normal FCU valve operation might cause the zone temperature to rise instead of to be cooled. To counter this, the effect of a smart valve for the FCU, which opens only when T_{sup} is lower than the respective zone temperature, is investigated. The normal valve is denoted with a ”0”, and the smart one with ”1”.

With regard to controlling the bypass valve, three types are considered, namely ”A” no bypass, ”B” a constant bypass flow when the $T_{sup;} < T_{sup;\,SP}$ and ”C” is the same as ”B”, but is activated only when one of the dwellings has an actual cooling demand. As the $T_{sup;\,SP}$ is automatically reset to SC temperatures when all DHW vessels have been recharged, control ”C” is intended to minimise the number of changeovers from DHW to SC regimes.

The following regimes are considered for SC design temperatures: 7°C/12°C, 12°C/17°C and 17°C/22°C. Higher SC design temperatures reduce the dissipated heat and cold when $T_{sup;SP}$ changes, but also reduce the cooling capacity of the FCUs, as their sizing does not change and is only based on design heat load. For DHW, $T_{sup;\,SP}$ is set at 65°C.

Evaluation Framework

The two baselines: no SC and 4-pipe

To evaluate the performance of the demand-based control strategy, two baseline concepts are used, namely the two extremes in terms of distribution cooling capacity. First, the conventional 2-pipe CHDC with a fixed $T_{sup;\,SP}$ of 65°C that ensures the supply of DHW and does not allow SC distribution. Second, an advanced 4-pipe collective heating system with two separate distribution circuits, one for SC and one for DHW. Due to separating the demands in two circuits, this concept provides the most optimal thermal comfort to end-users. It is the goal to be as close as possible to the performance of the 4-pipe baseline. Both baselines have decentralised DHW vessels.

Simulation framework

This research uses a simulation environment in Matlab as a testbed for new control strategies intended to enable space cooling in collective heating and, if successful, cooling systems with a single supply pipe. In this way, the SH, SC and DHW demands are the same for different variants, which makes an objective comparison possible. The structure and models of the simulation environment are based on the PhD dissertation of Van Riet (2019) and have been used in previous research (Jacobs et al., 2021, 2022, 2023). The dynamic thermal behaviour is modelled based on ordinary linear and non-homogeneous first-order differential equations, generally written as equation (1). Only the storage vessel models have partial differential equations by temperature and height to simulate stratification inside.

$$\frac{dy(t)}{dt} = -a(t)y(t) + b(t) \qquad (1)$$

With y the calculated temperature of a component and t [s] the time. The explicit solution of $y(t)$ is used with a zero-order hold, meaning $a(t)$ and $b(t)$ are constant during each time step as in De Pauw et al. (2018) and Van Riet (2019). The simulation time step is 10 seconds to include the DHW demand in detail and the simulated period is July and August. For the hydraulic behaviour, it is assumed that the CHDC is perfectly balanced with balancing valves (Tayler and Stein, 2002), thus the nominal mass flow rates are always available. The valve authority is assumed to be 0,5 or higher, which allows a stable control of all flow rates between zero and the nominal flow rate (Hegberg, 2000). All control valves have a time delay of 32 seconds when opening or closing (Van Riet, 2019).

Main models

<u>*Stratified DHW storage vessel model.*</u> A partial differential equation in temperature and height represents the thermal behaviour of the DHW storage vessels, including conduction, advection, heat losses to surroundings ($\dot{Q}_{sto}^{DHW}$) and the heat gains from the internal coil heat exchanger. The model is based on Type 60 of TRNSYS (SELUWM, 2009) and is described in previous works (Van Riet, 2019; Jacobs et al., 2022).

<u>*Emitter model.*</u> The space cooling is emitted by FCUs (100% convection) in each dwelling. To account for the thermal inertia of the emitter and avoid required iterations with the $\varepsilon - NTU$ method, the emitter is divided into three segments of uniform temperature as in Van Riet (2019). For each segment the differential equation is as follows:

$$\frac{C_{em}}{3}\frac{dT_i^{em}}{dt} = c_p\dot{m}_{em}(T_{i-1}^{em} - T_i^{em}) - \dot{Q}_{em,i}^{zone} \qquad (2)$$

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With C_{em} the overall thermal capacity of the emitter [J/K], T_i^{em} the uniform temperature of each segment ($T_0^{em} = T_{in}^{em}$ and $T_3^{em} = T_{out}^{em}$) in [°C], cp = 4187 J/kgK, $\dot{m}_{em}$ the flow rate through the emitter [kg/s], controlled by a thermostatic valve, and $\dot{Q}_{em,i}^{zone}$ the heat flux between the zone and segment i. $\dot{Q}_{em,i}^{zone}$ is calculated for each segment based on an overall heat transfer coefficient $UA_{em}/3$, T_i^{em} and the indoor zone temperature, T_{zone} [°C]. The total heat flux from emitter to zone ($\dot{Q}_{em,tot}$) equals to $\sum_{i=1}^{3} \dot{Q}_{em,i}^{zone}$.

The overall thermal capacity is calculated with data of the *FWV C TV* (Daikin Europe N.V, 2022), namely by multiplying the total mass by 530 J/kgK, i.e. the specific heat capacity of metal plate. Based on the design heat load, normalised to a 50/40/21 temperature regime, maximum two FCUs are selected for each dwelling. Table 1 gives an overview of the data.

Zone model. Each dwelling is represented by a 3R2C model. The two thermal capacities in [J/K] are both related to the indoor air volume (V_z) of the respective dwelling: C_w is according to semi-heavy walls (Vlaams Energie- en Klimaatagentschap, 2022), i.e. V_z × 87000 J/m³K, and C_z equals 1296 J/m³K × V_z × 5. The factor 5 is to take account of furniture in dwellings. The thermal behaviour of indoor zone temperature (T_{zone}) and the wall inside surface temperature (T_{wall}) are represented by:

$$\begin{aligned} C_z \frac{dT_{zone}}{dt} = \; & \dot{Q}_{em,tot} + (1 - 0{,}5 r_{sol}) \cdot \dot{Q}_{sol} \\ & + (1 - 0{,}5 r_{int}) \cdot \left(\dot{Q}_{int} + \dot{Q}_{sto}^{DHW}\right) \\ & - h_{s,i} \cdot (T_{zone} - T_{wall}) \\ & - c_{p,air} \cdot \dot{m}_v \cdot (T_{zone} - T_{ext}) \end{aligned} \quad (3)$$

$$\begin{aligned} C_w \frac{dT_{wall}}{dt} = \; & 0{,}5 r_{sol} \cdot \dot{Q}_{sol} \\ & + 0{,}5 r_{int} \cdot \left(\dot{Q}_{int} + \dot{Q}_{sto}^{DHW}\right) \\ & + h_{s,i} \cdot (T_{zone} - T_{wall}) \\ & - UA_w \cdot (T_{wall} - T_{ext}) \end{aligned} \quad (4)$$

In (3), the solar radiation $\dot{Q}_{sol}$ [W] and the outdoor temperature T_{ext} are both from Belgian weather data. $h_{s,i}$ = 8 W/m²K, $c_{p,air}$ = 1005 J/kgK and $\dot{m}_v$ is the ventilation mass flow [kg/s]. In (4), UA_w represents the heat transfer coefficient of wall-outdoor interface. The r_{sol} and r_{int} are the proportion of radiant heat from solar radiation gains (100%) and internal heat gains (50%), respectively. Of the radiant heat, 50% is transferred to T_{wall} and 50% to T_{zone}, since the zone node also includes the furniture.

Distribution pipe model. The plug-flow principle (Van Riet et al., 2018) to simulate pipelines is applied to simulate the water transport delay and thermal losses to surroundings. Only the thermal capacity of the water volume is considered (pipe walls are neglected) and the water pipes inside dwellings are not taken into account in this research. The 68m long distribution pipes are conventionally sized based on the design cooling loads and SC regime temperatures, a simultaneity for $\dot{m}_{ch}$ and a water velocity of 1 m/s. In case of 2-pipe, both distribution pipes have a water volume of 154 L. In case of 4-pipe system, the 2 pipes for SC each contain 103 L and only 51 L for DHW (due to decentralised DHW storage).

Key Performance Indicators (KPIs)

- *Average DHW temperature lack ($t_{DHW;dc}$):* The relative duration of DHW dicomfort, i.e. DHW contumption temperature below 40°C during tapping in [%], as in Jacobs et al. (2022).
- *Room Temperature Escess (RTE):* The number of Kelvin hours per day that T_{zone} is above it set point, given in [Kh/day].
- *Total Heating Energy demand (E_{tot}^H) and Cooling demand (E_{tot}^C):* The total heating and cooling supplied by the central production units in [MWh]. Since the central production units are not defined, this KPI is calculated from $T_{sup;SP}$, T_{ret} and $\dot{m}_{CHDC}$ for each time step.
- *Dissipated heat (E_{dis}^H) and dissipated cold (E_{dis}^C):* The total amount of heat and cold in the distribution pipes that needs to be dissipated to switch the supply temperature from DHW to SC (E_{dis}^H) and vice versa (E_{dis}^C), in [MWh]. At the start of each changeover, the respective dissipated heat or cold is calculated based on the water volumes and average water temperatures in the distribution pipes.
- *Number of changeovers between DHW and SC (N_{change}):* Only taken into account when $\dot{m}_{CHDC} > 0$, thus a demand exists. The fewer changeovers, the less heat and cold has to be dissipated. This is zero for both baselines.

Table 1: Data of Daikin's FCU 2-pipe floor models (Daikin Europe N.V., 2022).

Type	Cool power [kW]	Heat power [kW]	Total mass [kg]	$\dot{m}_{em}^{nom}$ [l/h]
01	1,20	2,14	19	256
02	1,51	2,57	20	359
03	2,11	3,81	25	504
04	3,15	5,63	30	745
06	3,65	6,36	31	820
08	4,91	7,83	41	1154
10	5,96	10,03	41	1343

Results and discussion

Bypass control variants

Table 2 shows RTE, $t_{DHW;dc}$ and N_{change} for different bypass control strategies as well as for the two baselines. The regime temperatures for SC are 7°C/12°C. The 2-pipe baseline has an average RTE of 78,2 Kh/day per dwelling and no changeovers because no SC is provided to the end-users, while the 4-pipe system (with space cooling always available in a separate circuit) reduces the RTE to only 0,4 Kh/day. For both baselines, $t_{DHW;dc}$ is 0,32% of tap time, meaning that the DHW consumption temperature is

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rarely lower than 40°C. The 2SC resulted in slightly worse DHW comfort than the baselines, but this is due to the need of changeovers leading to slower response times for recharging the DHW vessels. In contrast, the RTE is always between the two baselines and can even reach 7,2 Kh/day. In this respect, applying a smart valve in the FCU (”1” variants) is more beneficial than the different bypass control strategies. As can be noted, N_{change} is similar for each bypass control strategy, which means that a SC demand existed in at least one of the dwellings after each period of DHW vessels recharging. Only control strategy ”B” had fewer changeovers, but this is assumed to be due to initialising the simulation.

Table 2: Bypass control strategies: the RTE, $t_{DHW;dc}$, and N_{change} of bypass control variants "A", "B" and "C" are presented. The number "0" and "1" refers to whether or not a smart valve in the FCU is used.

	RTE [Kh/day]	$t_{DHW;dc}$ **[%]**	N_{change}
Baseline 2-p	78,2	0,32	0
2SC A0	67	0,59	373
2SC B0	66,3	0,54	369
2SC B1	7,2	0,56	371
2SC C1	7,2	0,56	375
Baseline 4-p	0,4	0,32	0

Also Figure 3 suggests that the smart valve is important for reducing the RTE. It shows for the different control variants the total heat and cold distributed by the idealised central production and the proportion of that heat and cold that had to be dissipated due to changeovers. The total heat production of 2-pipe and 4-pipe baselines is only 9,13 MWh and 9,23 MWh, respectively, which is sufficient to cover the DHW demand (6,27 MWh) and losses of distribution and DHW storage. In contrast, the two control variants with a normal valve for FCU, i.e. 2SC A0 and 2SC B0, require an E^H_{tot} of 49,11 MWh and 48,7 MWh, respectively, which is 5,3 times the heating demands of the baselines while the flow control of the FCU is in cooling mode. The reason is that the FCU unintentionally heats the dwelling, by withdrawing a flow rate intended for SC, but at the time T_{sup} is still high. This also explains the large RTE of those variants. Moreover, the unintended heating introduces an extra cooling load, which is also shown by the two times larger blue bars of 2SC A0 and 2SC B0 in Figure 3.

Yet the variants with a smart valve in the FCU also distribute 8,2% more cooling energy through the network than the 4-pipe baseline, without being able to achieve the same RTE (7,2 Kh/day versus 0,4 Kh/day). This is due to the large N_{change}, as each changeover from DHW to SC requires cooling capacity from central production and each changeover from SC to DHW requires dissipation of all existing cooling in the distribution pipes. The same applies for total distributed heating and $t_{DHW;dc}$, where 2SC B1 and C1 produced 19,7% more heat than the 4-pipe system. The E^C_{dis} and E^H_{dis} are 2,535 MWh and 3,174 MWh, respectively, for variant 2SC C1, which is 11,4% of the total cooling demand (E^C_{tot}) and 28,7% of E^H_{tot}. Consequently, 2SC was found to have great potential to enable space cooling in 2-pipe collective thermal systems, but needs further refinement to reduce heat and cold losses due to changeovers that are not currently exploited.

Design temperature of SC

In an attempt to reduce the dissipated heat and cold, the difference between DHW and SC supply temperatures is reduced. In this respect, $T_{sup;SP}$ for DHW cannot be lowered, as it is needed to store 62°C (hysteresis of 4°C) in DHW vessels. Therefore, for “2SC C1”-variant, the design temperatures for SC are increased from 7°C/12°C to 12°C/17°C and 17°C/22°C, as SC is possible at higher temperatures with FCUs.

The results in Table 3 are as expected. On the one hand, the RTE increases (poorer indoor temperature comfort) with increasing SC distribution temperatures for both 2SC and 4-pipe and the difference between delivered thermal comfort of 2SC and 4-pipe is smaller. On the other hand, the E^C_{dis} and E^H_{dis} of 2SC reduce at higher SC temperature regimes.

General discussion

The 2SC has shown great potential to distribute SC in 2-pipe systems, such as CHDCs, at difference regime temperatures while ensuring DHW without using decentralised booster systems.

It can reduce the RTE from 78,2 Kh/day (when no SC is available) to 7,2 Kh/day. In this respect, a smart control valve for each FCU is required, as otherwise the FCU might unintentionally heat the dwellings during times of high T_{sup} for DHW. However, even with the smart valve and an attempt to optimise the bypass control (strategy ”C”), the dissipated heat and cold takes up a large part of the distributed heat and cold, up to 28,7% and 11,4% respectively. The main cause for the high share of e.g. dissipated heat is that each changeover from SC to DHW requires heating capacity from central production and each changeover from DHW to SC requires dissipation of all existing heating in the distribution pipes. Since the central production room is not defined and considered ideal, these E^C_{dis} and E^H_{dis} are not recuperated and also increase the total heat and cold demand.

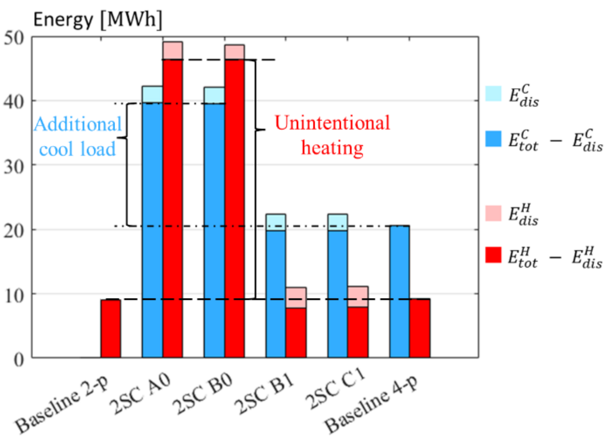

Figure 3: Bypass control strategies: E^C_{tot}, E^H_{tot}, E^C_{dis} and E^H_{dis} for the different control approaches.

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To recover or reduce the E_{dis}^{C} and E_{dis}^{H}, a few possibilities are discussed that need further investigation.

First, an option is to use two central storage vessels: one for storing heat and one for storing cold. Based on the average dissipated heat and cold per changeover, the size of those storage vessels can be determined with equation (5):

$$V_{stor} = \frac{3600\, s/h \cdot E_{dis}^{change}}{1000 \frac{kg}{m^3} \cdot 4{,}187 kJ/kgK \cdot \Delta T} \tag{5}$$

Here, V_{stor} is the volume required to store the dissipated thermal energy (heat or cold, depending on the purpose of the storage vessel) in [m³], E_{dis}^{change} is the average dissipated thermal energy (heat or cold) per changeover in [kWh] and ΔT is the design temperature difference between storage vessel and temperature to be stored. For example, the dissipated heat for 2SC C1 at 12°C/17°C is 2,48 MWh for 373 changeovers. Of these 373 changeovers, 186 were DHW to SC changeovers, resulting in the need for heat dissipation. Thus, on average, 13,3 kWh of heat should be stored per changeover (E_{dis}^{change}). At for example a ΔT of 10°C, the minimum storage volume is 1,15 m³, which is quite large for additional storage. When the ΔT between the distribution temperature to be stored and the current storage temperature is smaller, the volume should be even larger.

Second, both the 2SC and the bypass control should be optimised to reduce the number of changeovers or the dissipated heat and cold per changeover, because only including a SC demand signal in the bypass control (strategy "C") does not seem to affect this.

Third, the sizing of distribution pipes could be optimised taking into account the control strategy applied. Since the day is divided into times of SC and times of DHW, the distribution pipes may need not be that large. When the distribution pipes for 2SC C1 are sized the smallest possible for the simulation time step, thus only based on SC as this requires the largest flow rates, then each pipe volume is only 57 L ($= 24 \times 0{,}237\, kg/s \times 10\, s$). With these very narrow distribution pipes (because pipe length remains 68m), the E_{dis}^{C} and E_{dis}^{H} can be reduced to only 0,80 MWh and 0,98 MWh, respectively, which is 3,14% and 10,8% of E_{tot}^{C} and E_{tot}^{H}. In contrast, the RTE increases to 14,7 Kh/day. However, this small volume indicates that the diameter is only 32,7 mm, so at peak load (when all FCU's in the dwellings are cooling), the water velocity would be 6,8 m/s. This velocity is too high for collective heating networks. Therefore, future research should also focus on optimising the distribution pipe dimensions to make a trade-off between water velocity, delivered thermal comfort and dissipated thermal energy.

Finally, using decentralised storage in the control strategy could be further elaborated by storing the heat and cold dissipation at the end-users level. Regarding cold dissipation, underfloor systems can be used as decentralised cold storage. When $T_{sup;SP}$ is set at 65°C to charge DHW vessels, all underfloor systems can be switched on to store available cold from the supply pipe in dwellings. Due to its high thermal inertia, SC will not immediately affect the indoor comfort and reduce SC demand during DHW vessels charging. Moreover, the smart use of decentralised storage could also focus entirely on the underfloor system, rather than DHW storage.

Another point of attention for future research is further optimising the delivered thermal comfort of 2SC. The RTE is already reduced by 90,7% compared to traditional 2-pipe systems without SC, but the 4-pipe achieves 20,5 times lower RTE (only 0,4 Kh/day). Also $t_{DHW;dc}$ of both baselines is better than for all 2SC variants, but this is always lower than 1% of tap time.

Conclusion

This research identified the potential of grouping same temperature demands by smart use of decentralised storage (Jacobs et al., 2023) to also be applied to distribute SC in a collective 2-pipe thermal system in apartment buildings, namely a CHDC. Moreover, the energy-related impacts of SC supply in this system were determined, with suggestions for future optimisation possibilities.

The evaluation framework was based on simulations of a CHDC consisting of 24 dwellings, each with an FCU and a DHW storage vessel. Two baselines are used, namely the traditional 2-pipe CHDC without SC possibilities and a 4-pipe collective thermal system with both SC and DHW supply.

The results show the need for a smart valve in FCUs that measures the central supply temperature to avoid unintended heating when it is high (for DHW). This in turn leads to unwanted heat production and higher cooling demand. An RTE of only 7,2 Kh/day (compared to 78,2 Kh/day without SC) was reached, but 4-pipe systems achieve 0,4 Kh/day. Moreover, including a SC demand signal in the bypass control did not decrease the number of changeovers. However, for cases with lower cooling requirements, this could help.

Finally, the dissipated heat and cold amounts to about 11,4% and 28,7% of the total heat and cold production at a temperature regime of 7°C/12°C. This dissipation should be minimised in future work, e.g. through centralised storage for the reuse of dumped heat and cold, optimised pipe sizing taking into account the control strategy, optimisation of the control strategy to reduce the number of changeovers and/or the use of underfloor systems in dwellings as decentralised cold storage.

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