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Temperature control strategies for fifth generation district heating and cooling systems: A review and case study

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HIGHLIGHTS

- Comprehensively reviewed temperature control strategies for 5GDHC systems
- Developed detailed virtual testbed for 5GDHC system simulation using Modelica
- Evaluated various strategies' performance across multiple climates and configurations
- Provided practical recommendations for strategy selection based on system analysis

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Fifth Generation District heating and cooling (5GDHC)
Temperature control
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ABSTRACT

Temperature control exerts a significant influence on the performance of Fifth Generation District Heating and Cooling (5GDHC) systems. Ineffective temperature control could lead to not only occupant discomfort and higher operation costs due to increased energy consumption and reduced equipment lifespan, but also a negative environmental impact through higher carbon emissions. Despite its importance, this area remains underexplored, with limited studies conducting in-depth analyses of temperature control strategies applied in 5GDHC systems. To bridge this knowledge gap, this study conducts a comprehensive literature review on temperature control strategies for 5GDHC systems. Three prevalent temperature control strategies have been identified: constant temperature control, multi-stage temperature control, and free-floating temperature control. To facilitate the implementation and relatively fair comparison of these strategies and their various settings, detailed 5GDHC system models have been developed using Modelica. The case study encompasses two network configurations, spans across four distinct climate zones, and evaluates the system performance using five key performance indicators, including total energy consumption, system coefficient of performance (COP), carbon emissions, peak electricity demand and plant capacity. The simulation results reveal that the constant operating temperature control strategy can outperform the free-floating temperature control strategy when appropriate network operating temperature settings are adopted but also exhibit greater sensitivity to changes in these settings. Simple multi-stage temperature control strategies are found to be insufficient to achieve the best system performance. In addition, substation direct cooling could be an effective means to enhance the system performance but necessitates coordination with suitable temperature control strategies to maximize energy savings. In general, the performance of temperature control strategies in 5GDHC systems is strongly influenced by multiple factors such as load profiles, system configurations and specific strategy settings. This study underscores the importance of selecting and designing temperature control strategies based on comprehensive system analyses to achieve high performance 5GDHC systems during operations.

1. Introduction

Amid the rapid societal and economic development, energy crises and climate concerns have risen to the forefront of public consciousness. The building sector stands as the most significant consumer of energy in the U.S. economy, which accounts for around 40% of the nation's total energy usage, 74% of its electricity consumption, and 35% of its overall

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Nomenclature

Abbreviations

4GDH	Fourth-generation district heating
5GDHC	Fifth-generation district heating and cooling
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
COP	Coefficient of performance
DC	District cooling
DER	Distributed energy resource
DH	District heating
DOC	Demand overlapping coefficient
DOE	Department of Energy
HCDR	Heating-cooling demand ratio
HDPE	High-density polyethylene
HVAC	Heating, Ventilation and Air Conditioning
LTDH	Low-temperature district heating
MILP	Mixed-integer linear program
PI	Performance indicator
PV	Photovoltaics
SSS	Sub-keyword Synonym Searching
TOU	Time of use
UBEM	Urban building energy modeling
ULTDHC	Ultra-low temperature district heating and cooling

Variables

E	Electric energy
$\dot{m}$	Mass flowrate
P	Electric power
Δp	Pressure drop
Q	Thermal energy
$\dot{Q}$	Thermal power
T	Temperature
ΔT	Temperature difference
t	Time

Parameters

η	Effectiveness
k	Flow coefficient
A	Amplitude
z	Buried depth
τ	Annual period length
α	Thermal diffusivity of the soil
SF	Safety factor

Subscripts and Superscripts

act	Actual
air	Outdoor air
app	Approach
cap	Capacity
cold	Cold pipe
cond	Condenser
coo	Cooling
dem	Demand
ele	Electricity
ent	Entering
evap	Evaporator
hea	Heating
high	High
hub	Energy hub
lag	Phase lag
lmt	Limit
low	Low
lvg	Leaving
max	Maximum
min	Minimum
ms	Mean annual surface
norm	Normalized
s	Soil/surface
sub	Substation
tot	Total
warm	Warm pipe

carbon emissions [1]. Notably, Heating, V*ent*ilation, and Air Conditioning (HVAC) systems constituted a significant portion of this sector, typically responsible for approximately 40% of a building's total energy usage [2]. Thus, there is an urgent need to curtail energy consumption in HVAC systems to address the growing energy and climate challenges.

District heating (DH) and district cooling (DC) systems have demonstrated their effectiveness in reducing building energy consumption and greenhouse gas emissions, especially in regions characterized by high thermal load density [3–5]. Due to the capability of handling the demand diversity and the use of equipment with higher efficiencies, generating thermal energy in central plants is generally more energy efficient than relying on individual heating and cooling systems for each building; thus resulting in substantial reductions in environmental emissions. In addition, district-scale systems open up opportunities to harness distributed energy sources (DERs) that may not be viable on an individual building scale, such as industrial waste heat or lake water. Furthermore, district energy systems can yield significant economic benefits by capitalizing on economies of scale and higher operational efficiency. These combined advantages make DH and DC systems pivotal in addressing energy efficiency and environmental concerns in urban areas [6].

The evolution of district energy systems have gone through several phases, as depicted in Fig. 1 [7,8]. The first generation of district heating systems relied on steam-based technologies fueled by fossil fuels like coal and oil, operating at supply temperatures of around 200 °C. This approach remained prevalent until the 1930s when pressurized hot water emerged as a safer and more efficient heating medium, which became the major heating medium of the second-generation district heating systems, marking the advent of the second-generation district heating systems but the supply temperature is still above 100 °C. Starting in the 1980s, the third generation of district heating systems began to gain prominence. While still employing the pressurized hot water as the heating medium, this generation significantly enhanced the system efficiency by reducing supply temperatures to below 100 °C. In pursuit of greater efficiency and the transition to a sustainable energy system, the concept of Fourth Generation District Heating (4GDH) was brought up. It introduced lower operational water temperatures, falling below 70 °C, leading some researchers to refer to it as low-temperature district heating (LTDH) systems [9–11]. A widely accepted definition of 4GDH was established by Lund et al. in 2014 [7], emphasizing its ability to be part of a smart energy system by exploiting synergies from cooperation with other elements or sectors of the energy system, such as smart electricity, gas and thermal grids.

In recent years, with increasing concerns about the adverse effects of climate change, electrification has been growing in popularity as a solution to reduce carbon emissions. Concurrently, advances in technology have made electric-based appliances more appealing to consumers [12]. As a result, a new type of district energy network integrated with multiple distributed heat pumps has emerged as a promising technology for achieving building decarbonization, as illustrated in Fig. 2 [13]. The distinguishing feature of these innovative district energy networks is their operation at close-to-ground temperature levels. Since these

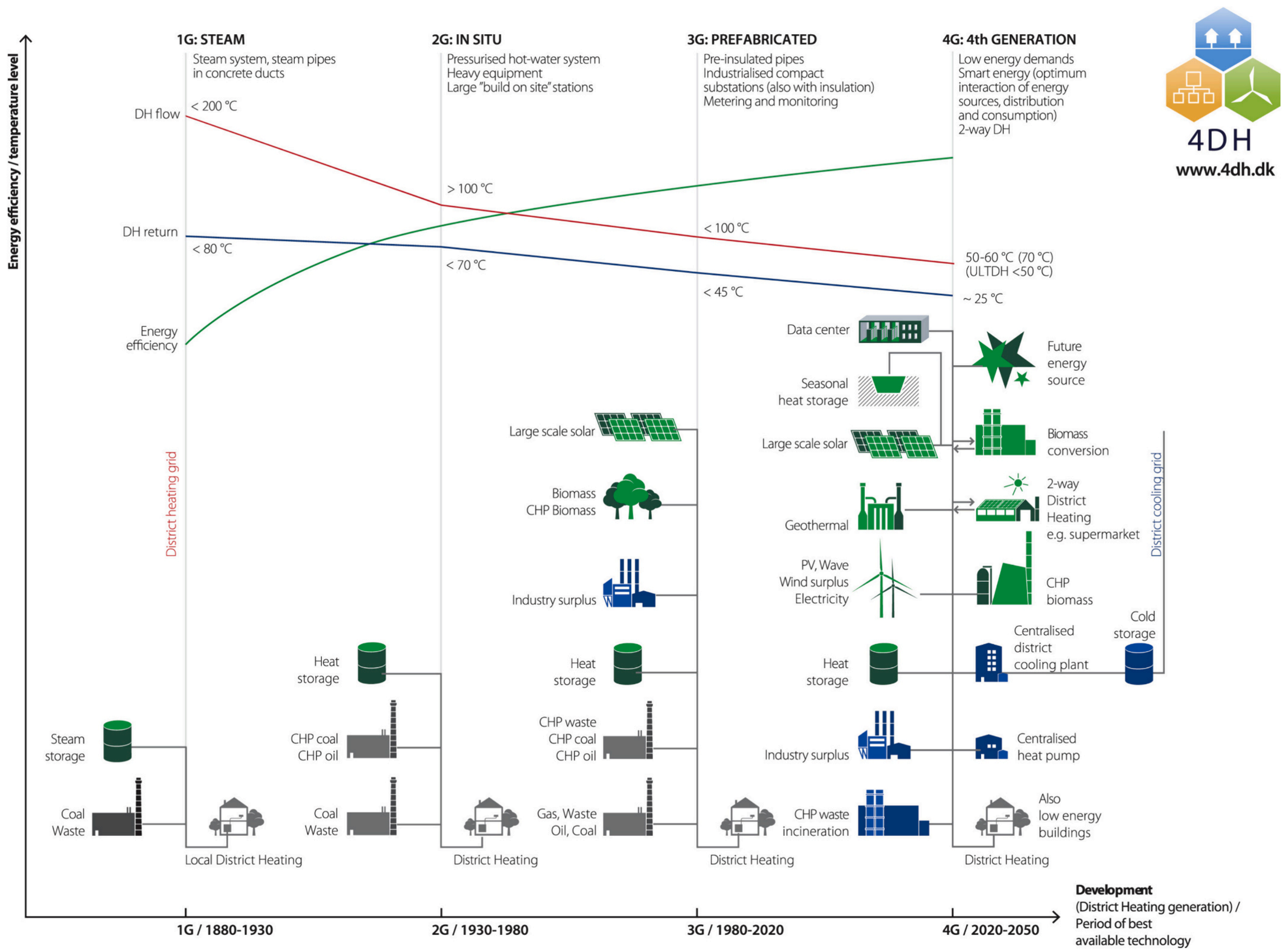


Fig. 1. Development history of district energy systems [8].

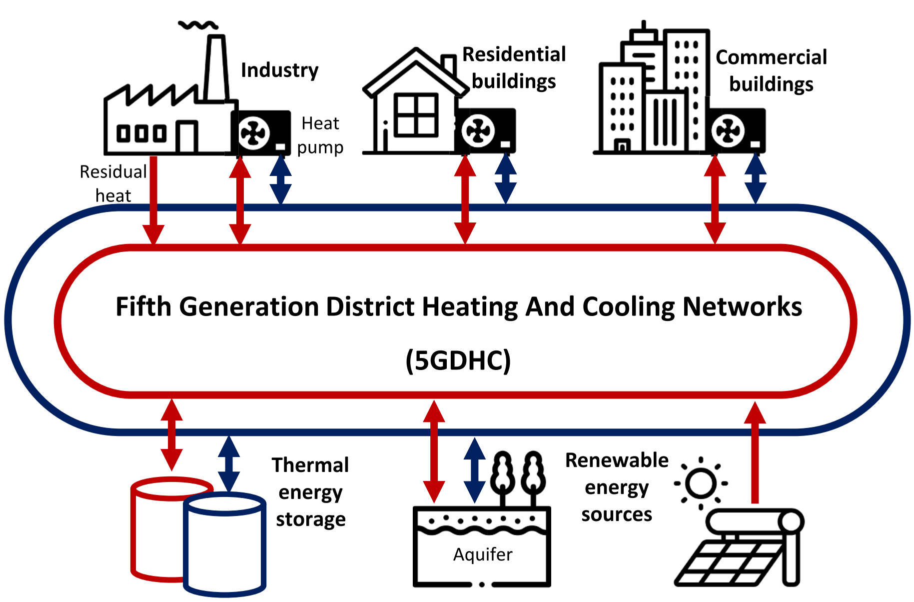


Fig. 2. Conceptual representation of a 5GDHC system (Adapted from [29,30]).

temperatures are insufficient for directly meeting the users' heating or cooling requirements, distributed heat pumps and chillers are equipped at substations to elevate or reduce temperatures to the desired levels, thereby enabling simultaneous heating and cooling. Currently, such systems have gained some adoption in Europe [14]. Various terminologies are being used to refer to this type of system, such as fifth-generation district heating and cooling (5GDHC) systems [15–21], ultra-low temperature district heating and cooling systems [22,23], ambient loops [24], and cold district heating networks [25]. An ongoing debate persists regarding whether this type should be classified as a new generation of district energy systems. While some researchers [26,27] argued for its classification as 5GDHC due to its unique characteristics, others [8,28] contended that the core principles of these so-called 5GDHC systems closely aligned with 4GDH systems and, therefore, this system should not be considered a new generation. Despite this debate, due to its broad adoption in existing studies, this study uses the term "fifth-generation district heating and cooling (5GDHC) systems" to refer to these systems. More comprehensive discussions on 5GDHC system characteristics are provided in Section 2.2.

It is well acknowledged that temperature control plays a pivotal role in the operational performance of district heating and cooling systems [31,32]. First and foremost, it guarantees that desired temperature levels can be consistently attained by all end-users. Conversely, poor temperature control can lead to discomfort and complaints from building occupants. Secondly, the system's operating temperature significantly influences the integration of thermal energy sources. A broader temperature range enables the utilization of a wider spectrum of thermal energy sources [33]. Thirdly, precise temperature control facilitates the efficient operation of heating and cooling devices within the district energy system. Operating at optimal temperature levels reduces energy consumption, leading to cost savings and reduced greenhouse gas emissions. In addition, the operating temperature has a direct impact on heat transfer from the pipelines to the surroundings. For instance, in conventional district heating networks, higher operating temperatures generally result in increased network thermal losses to the surroundings, leading to the wastage of heating energy. District cooling systems are economically favorable in dense urban and industrial areas but less so in low-density regions due to higher cold transportation losses [24]. Lastly, it is worth noting that improper temperature controls can also result in a shortened lifespan of equipment within the network, which in turn leads to higher maintenance costs [34]. Therefore, ensuring high-quality temperature controls is fundamental to the efficiency of district energy systems.

Currently, there are well-defined standards and guidelines for temperature controls in conventional district heating and cooling systems, e.g., ASHRAE District Heating Guide and District Cooling Guide [35,36]. In addition, numerous studies [37–39] are dedicated to optimizing temperature control strategies in these systems. These studies typically share a common objective, which is to either lower the supply temperature (for district heating) or raise it (for district cooling) while increasing the temperature difference (ΔT) between supply and return lines. For instance, in the context of district heating, by providing the same amount of heat, reducing the supply temperature can help increase the overall efficiency of the system while reducing heat losses from the distribution pipelines. Meantime, an increase in ΔT can result in reduced pumping power requirements and decreased heat losses through piping distribution as well. These measures can ultimately lead to cost savings and higher energy efficiency.

While established guidelines and optimization methods have proven effective for conventional district heating or cooling systems, they cannot be directly applicable to 5GDHC networks due to their distinct characteristics. Firstly, the presence of heat pumps and chillers at substations in 5GDHC systems eliminates the need for the network's operating temperature to provide heating or cooling to the substations directly. This flexibility in temperature controls is a fundamental distinction. Secondly, 5GDHC systems, capable of simultaneously providing heating and cooling, lack the clear patterns in the relationship between the system energy consumption and operating temperature found in conventional district heating or cooling systems. Thirdly, as 5GDHC systems operate at temperatures close to the ground level, heat transfer through the pipeline is almost negligible. Under certain circumstances, the surrounding soil can even serve as a thermal energy storage system, enhancing system energy efficiency [40,41]. Lastly, the lower operational temperature of 5GDHC systems facilitates the integration of additional low-grade thermal energy sources and renewable energy sources [42], such as shallow geothermal sources, solar thermal collectors and sewage water [43]. However, these energy sources often exhibit fluctuating and intermittent characteristics, posing significant challenges for temperature controls in 5GDHC systems.

However, there are currently no established guidelines for developing appropriate temperature control strategies tailored to 5GDHC systems. While several reviews have provided basic and advanced control strategies for 5GDHC systems [29,34,44], none have delved into the specifics of temperature controls. Furthermore, there is a lack of studies that comprehensively compare multiple temperature control strategies within the same system. Additionally, the impact of specific settings on the performance of these temperature control strategies remains largely unexplored.

To address these research gaps, the objective of this study is set as follows:

1) To identify existing temperature control strategies applied in 5GDHC systems through a comprehensive literature review.
2) To develop a flexible virtual 5GDHC system testbed (using Modelica) that can simulate different system configurations and to implement multiple temperature control strategies identified from the literature review.
3) To enhance understanding of the impact of various temperature control strategies on 5GDHC system performance through simulations using the developed virtual testbed.

The overall structure of this paper is illustrated in Fig. 3.

It should be noted that this study will exclusively concentrate on temperature control strategies applied to the combined district heating and cooling distribution networks. Consequently, the following aspects are not within the scope of this study: (1) temperature control strategies used in systems that can only provide heating or cooling at a given time, (2) temperature control strategies used in systems that supply heating and cooling through separate networks, and (3) temperature control strategies employed in the in-building distribution systems.

2. Literature review and thematic analysis

2.1. Methodology

To conduct a comprehensive review of temperature control strategies in 5GDHC systems, the Sub-keyword Synonym Searching (SSS) [45] methodology was employed. SSS involves using sub-keywords and synonyms to perform multiple searches, thereby ensuring the thorough collection of the most pertinent papers in the field.

Table 1 presents the parameters employed in the SSS methodology for this study. Given the scope of the study, the search keywords were set to be related to the fourth and fifth generation of district energy systems, while conventional district energy systems (first to the third generation of district energy systems) were not considered. As indicated in the table, the total number of search keywords is $7 \times 5 = 35$ keywords.

The methodology retrieved relevant research from the years 2013 to 2023 for each keyword. After removing duplicate and clearly irrelevant search results, the methodology yielded 372 papers. All these papers were then manually reviewed, narrowing down the selection to 79 papers that align with the scope of the study. These chosen papers provide the essential key information upon which our research is based. Fig. 4

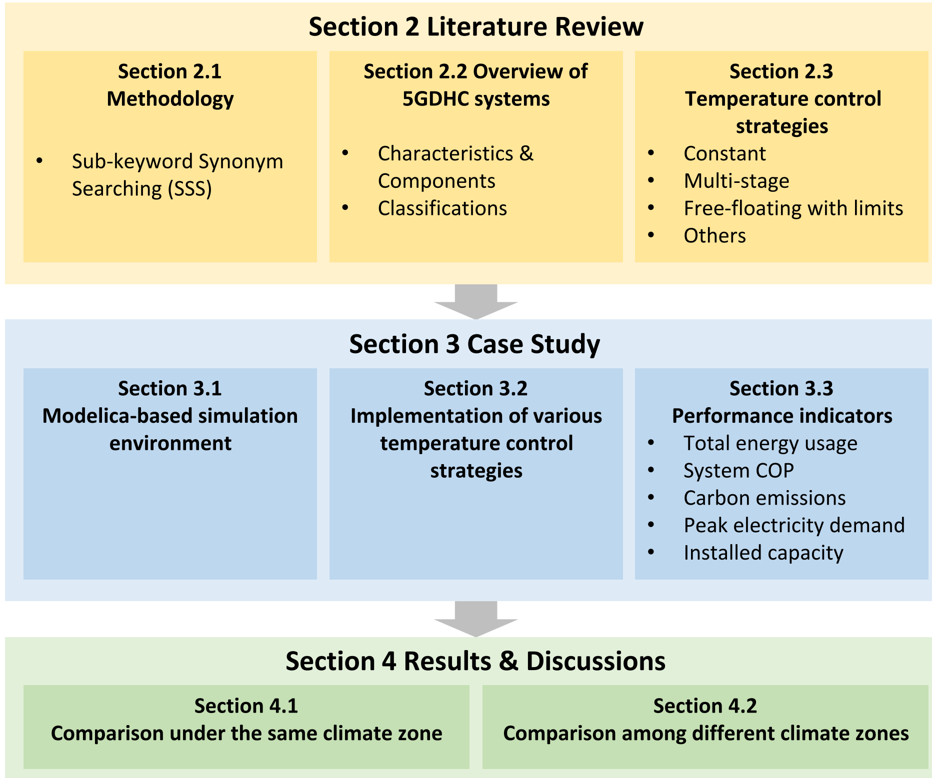


Fig. 3. Content organization chart of this paper.

Table 1
Parameters of Sub-keyword Synonym Searching (SSS).

Parameter	Values
Sub-keywords 1	5th generation, 4th generation, fifth generation, fourth generation, bi-directional, low temperature, ultra-low temperature
Sub-keywords 2	district heating and cooling, district heating, district cooling, district energy, district heat pump, thermal grid
Citation threshold	0
Number of papers per search	20
Year from	2013
Year to	2023

illustrates the number of publications over the years, showing a notable increase in research related to 5GDHC systems, reflecting the emergence and rapid growth of this technology.

Fig. 5 displays a word cloud chart generated from the titles of the retrieved papers [46]. A word cloud is a visual representation of text data where the size of each word indicates its frequency within the dataset. It provides a quick way to grasp the most prominent terms and their relative prominence, helping to identify trends, themes, and patterns in large datasets. Fig. 5 reveals that the literature on 5GDHC systems is comprehensive, covering various aspects such as network infrastructure, thermal energy management, optimization, and integration of renewable energy sources. The focus on analysis, modeling, and simulation underscores the importance of theoretical and computational approaches, while the mention of economic, sustainable, and

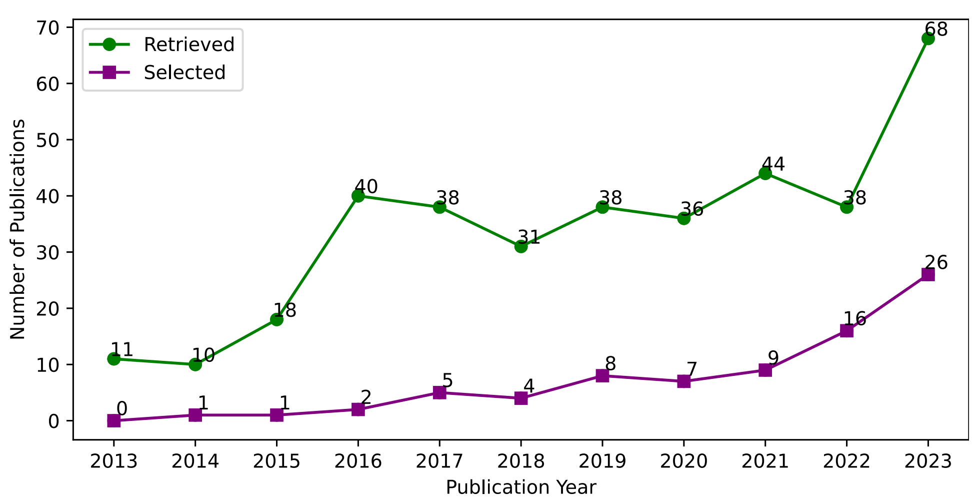


Fig. 4. Publication numbers from 2013 to 2023.

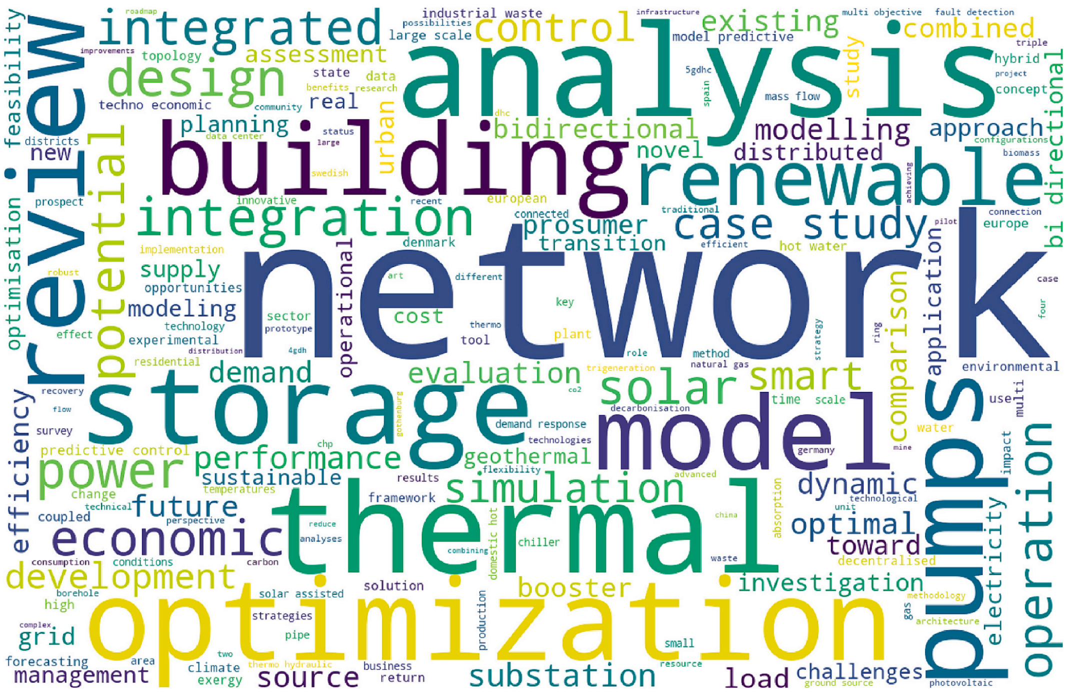


Fig. 5. Word cloud generated based on titles of the retrieved papers.

practical considerations reflects the real-world applicability and interdisciplinary nature of this research field.

2.2. Overview of fifth generation district heating and cooling systems

This section is intended to introduce fundamental concepts related to 5GDHC systems.

2.2.1. Characteristics and components of 5GDHC systems

In line with the definition provided by Buffa et al. [26], a 5GDHC system is a thermal energy grid that utilizes water or brine as a carrier medium and employs hybrid substations equipped with water-source heat pumps. These systems operate at temperature levels close to the ground, typically below 45 °C, and can deliver both heating and cooling to end-users. A typical 5GDHC system comprises three main components: substations, energy hubs, and distribution pipes.

- Substations: Substations are central components responsible for transferring energy from the district network to end-users, ensuring the delivery of heating and cooling as needed. They are often referred to as prosumers because they can act as both energy producers and consumers [47,48]. While typical substations include heat pumps for heating and chillers for cooling, some may also incorporate heat exchangers for direct cooling operations.
- Energy hubs: Energy hubs are alternatively referred to as balancing units or heating and cooling plants in various studies. Their primary role is to manage heat imbalances within the network and maintain network temperature levels by employing auxiliary heating or cooling devices. It is worth noting that energy hubs can draw on multiple energy sources for heating and cooling. Especially given the characteristics of 5GDHC networks, low-grade thermal energy sources (e.g., waste heat from sewage water), and renewable energy sources (e.g., solar) are more readily utilized.
- Distribution pipes: there are negligible thermal losses from the pipeline network to the surroundings due to the small temperature difference between the circulating water and surrounding ground. This allows for the use of uninsulated high-density polyethylene (HDPE) pipes, resulting in a significant reduction in installation time and costs compared to conventional district energy systems.

Table 2
Summary of major advantages and disadvantages of 5GDHC systems [8,26,30,49].

Advantages	Disadvantages
• Exploit the synergies of combined heating and cooling within the grid. • Enable the direct utilization of low-grade or renewable energy sources, such as lake water and industrial waste heat. • Ground-level operating temperatures result in significantly reduced heat losses through the pipeline network. • Enhance the sector coupling of thermal and electrical grids within a decentralized smart energy system through distributed substations.	• The installation of distributed heat pumps and chillers in substations can significantly increase initial investment costs. • Low-temperature differentials (ΔT) may require larger pipe diameters and lead to higher pumping costs due to increased required flow rates. • The design and control approaches used in conventional district energy systems may not be directly applicable and need to be reevaluated. • The transition from conventional district energy systems to 5GDHC systems remains a challenging task.

Table 2 provides a summarized overview of the main advantages and disadvantages of 5GDHC systems.

It is important to note that while most existing studies on 5GDHC systems focus on new designs, integrating 5GDHC systems with pre-existing conventional DHC networks is also feasible [50–52]. However, it requires a comprehensive assessment of current systems, reuse of existing infrastructure, availability of renewable energy sources, and consideration of financial incentives. Addressing these factors can facilitate a more efficient and cost-effective transition to 5GDHC systems [13,26].

2.2.2. 5GDHC classification in terms of different pipe configurations

In previous research [26,28], various classification methods of 5GDHC systems were introduced based on different criteria. For instance, 5GDHC networks can be categorized as open-loop or closed-loop systems, depending on the method for heat extraction or rejection from or towards the thermal source. They can also be classified as one-pipe, two-pipe, three-pipe, or four-pipe systems based on the number of pipelines at different temperature levels. Additionally, 5GDHC networks can be categorized as directional medium flow or non-directional medium flow systems, depending on the flow direction of the pipeline medium.

In addition to the classification methods discussed above, this study proposes a more comprehensive classification method based on both the pipeline configuration and substation connection type of common 5GDHC systems. Through an analysis of the literature, it is proposed that common 5GDHC systems can be categorized into the following four categories, as outlined in Table 3. For a visual representation of these categories, Fig. 6 presents schematic diagrams of different 5GDHC system configurations.

- Category I (1-pipe unidirectional series network): In this network configuration, all components, including substations, plants, or storage units, are connected in series to a main loop. Both centralized and decentralized pumps are required. Decentralized pumps extract the flow from the loop and provide the required flow to their corresponding components, while the centralized pump station maintains the flow rate and limits temperature variation in the main loop. Due to the short distance between the inlet and outlet of each component, there is minimal pressure drop. This configuration allows for the hydraulic decoupling of each component, providing better hydraulic control quality and extensibility. However, the substations are thermally coupled, which can lead to unstable substation inlet temperatures. Therefore, careful design and controls are necessary to ensure the performance of this system [54].
- Category II (2-pipe unidirectional parallel network): In this system, all components are connected in parallel, with predetermined flow directions for each pipe in the system. Pipes are classified as either supply or return pipes. This system typically employs a main centralized circulation pump station, but distributed pumps can also be utilized [57]. Compared to the single-pipe system, the inlet temperature of substations is stable, but the hydraulic balance across the entire network can be challenging due to the hydraulic coupling among substations.
- Category III (2-pipe bidirectional parallel network): This system features pipes with changing flow directions according to load conditions, rather than constant directions. As a result, there are no fixed supply or return pipes in the traditional sense; instead, warm and cold pipes are employed. The temperature difference between the warm and cold pipes is generally controlled to remain constant. Distributed pumps, with two at each substation, one for heating and another one for cooling, are typically used to ensure an adequate flow to the substations. This configuration theoretically minimizes exergy loss by avoiding the mixing of cold and hot water but faces challenges related to hydraulic control and pump cavitation [54].
- Category IV (2-pipe unidirectional series network): Calise et al. [56,58,59] proposed a novel configuration that involves two water loops operating at different neutral temperature levels, interacting through a cross-flow heat exchanger. This configuration is considered more effective and flexible compared with two-pipe bidirectional networks but is also more complex and costly. In addition, it does not fundamentally address the thermal coupling issue caused due to substation connections in series.

It is important to acknowledge that there are additional 5GDHC system configurations not covered in this proposed classification. For example, a 3-pipe configuration was employed in a system that utilized abandoned coal mining fields as heating and cooling resources [60]. However, since these configurations are relatively uncommon and only applicable to specific application scenarios, this study does not provide an in-depth discussion of them.

2.3. Temperature control strategies for 5GDHC systems

In this section, we will summarize temperature control strategies commonly employed in current 5GDHC networks. It is found that three major temperature control strategies are dominant in the selected papers presented in Section 2.1, namely, constant operating temperature control strategy, multi-stage operating temperature control strategy, and free-floating with limits control.

2.3.1. Constant operating temperature control

The constant operating temperature control strategy is a widely adopted approach in district energy systems. It involves maintaining a consistent supply temperature at which the system delivers heating or cooling to substations within the district. In unidirectional medium flow systems, where each pipe's flow direction is predetermined, a single fixed supply temperature is typically employed. However, in bidirectional systems, where the medium's flow direction adapts to the dominant load condition, there are two constant temperature setpoints—one for warm pipes and another one for cold pipes. Table 4 provides examples of 5GDHC systems that employ the constant operating temperature strategy.

A heating-cooling demand ratio (HCDR), listed in Table 4, is calculated to assess the balance between annual heating and cooling demands, which can be determined using Eq. (1) [68]. HCDR ∈ [−1,1]. HCDR = 1 indicates that only heating demands exist, HCDR = −1 indicates that only cooling demands exist, and HCDR = 0 indicates that the annual heating demands are equal to the annual cooling demands.

$$HCDR = \frac{Q^{tot}_{dem,hea} - Q^{tot}_{dem,coo}}{Q^{tot}_{dem,hea} + Q^{tot}_{dem,coo}} \quad (1)$$

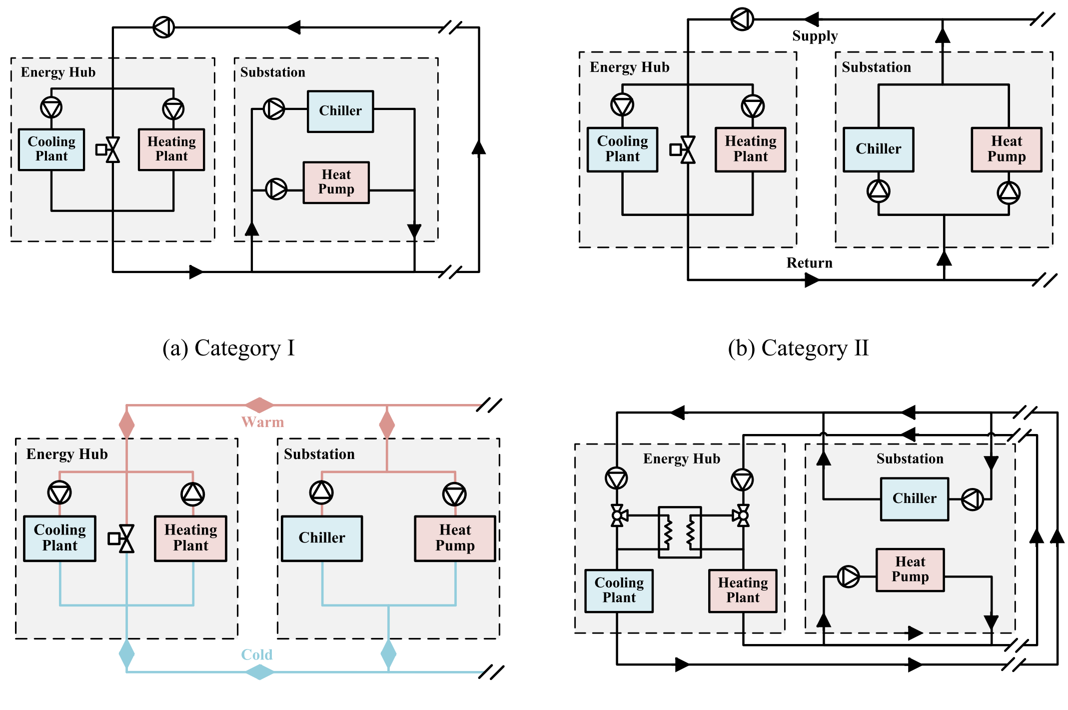
In which $Q^{tot}_{dem,hea}$/ $Q^{tot}_{dem,coo}$ are the annual heating and cooling demands.

Table 4 illustrates that the constant supply temperature strategy is prominently used in two types of 5GDHC systems. The first type encompasses systems equipped with mechanical heating or cooling devices in the energy hub that offer precise control over outlet temperatures, resulting in a constant supply temperature. The second type includes systems powered by energy sources with relatively stable thermal properties, such as deep geothermal sources or consistent waste heat. In these cases, the supply temperature can effectively be considered approximately constant.

It is important to emphasize that while the constant temperature control strategy is also commonly employed in conventional district

Table 3
Classification of 5GDHC Systems by Pipeline Configurations.

Category	Pipeline configuration	Medium flow	Substation connection	Pump placement	Terms
I	1-pipe	Unidirectional	Series	Both centralized and decentralized	Reservoir network [53], Uni-directional district heating and cooling system [54]
II	2-pipe	Unidirectional	Parallel	Centralized / Decentralized	Unidirectional parallel network
III	2-pipe	Bidirectional	Parallel	Decentralized	Bidirectional parallel network, Bidirectional low temperature network [55], Bi-directional district heating and cooling network [54]
IV	2-pipe	Unidirectional	Series	Both centralized and decentralized	Two rings network [56]



(a) Category I (b) Category II

(c) Category III (d) Category IV

Fig. 6. Schematic diagrams illustrating the proposed classification of 5GDHC network configurations.

Table 4
Review of 5GDHC systems that employ the constant operating temperature strategy.

Ref.	Location	Network configuration	HCDR	Thermal energy sources	Operating temperature	Direct cooling considered	Temperature Differential
[61,62]	Herne, Germany	III	0.26	Industrial waste heat, data center	Warm: 22 °C Cold: 12 °C	Y	10 °C
[63]	Warwickshire, England	III	0.73/ 0.43/0	Ground source heat pumps	23/ 46/ 58 °C	N	4 °C
[64]	Copenhagen, Denmark	II	0.86	Limitless renewable source	15 °C	N	4 °C
[65]	Helsinki, Berlinand, Strasbourg	II	0.60/ 0.40/ 0.19	Solar, ground source heat pumps	20 °C	N	Not mentioned
[66]	Milan, Italy	I	−0.06	Groundwater	15 °C	N	Not applicable
[67]	Germany	III	N/A	Combined heat and power (CHP), gas boiler, compression chiller, absorption chiller	Warm: 14 °C Cold: 10 °C	Y	4 °C

energy systems, the method for determining the operating temperature setpoint differs significantly between the conventional district energy systems and 5GDHC systems. In conventional district energy systems, substations only have limited control capability over water temperatures from the district networks, resulting in supply temperature setpoints primarily being determined based on user demand and equipment specifications [36]. In contrast, in 5GDHC systems, the installation of individual heat pumps at substations introduces a higher degree of flexibility in configuring operating temperatures. As demonstrated in Table 4, temperature setpoints in various 5GDHC cases can vary widely, ranging from 10 °C to 58 °C. However, this enhanced flexibility also raises questions regarding the optimal temperature setpoint for implementing the constant temperature control strategy, which requires further in-depth research and investigations.

While the constant operating temperature control strategy may appear to disregard users' demand conditions in 5GDHC systems, it does not necessarily imply inferior performance. As demonstrated by Büning et al. [69], the constant temperature approach resulted in reduced electricity consumption for heat pumps and chillers when compared to the free-floating approach, albeit with a focus on substation energy usage only. Due to its simplicity of implementation and potential energy savings, the constant operating temperature strategy should be seriously considered during the design of 5GDHC systems. This will be further substantiated through the case study presented in this paper.

2.3.2. *Multi-stage operating temperature control*

The multi-stage operating temperature control strategy involves the use of multiple temperature setpoints that vary based on changing load conditions, time of day, or other relevant factors. The most common multi-stage operating temperature control strategy is seasonal setpoint control. Due to the distinct differences between the load conditions of different seasons, different operating temperature setpoints are adopted to optimize the system's energy efficiency and occupant comfort.

Henchoz et al. [70] employed a 2-stage operating temperature control strategy for a double-pipe bidirectional parallel district water network. During the "cooling season" (when the outdoor air temperature is higher than 18 °C), the hot and cold pipe temperatures were limited to 16 °C and 12 °C, respectively, to enable direct cooling at the substations. For the rest of the time, the hot and cold pipe temperatures were set to 26 °C and 22 °C, respectively.

Bordignon et al. [65] introduced a variable setpoint temperature control strategy and compared it with a constant temperature setpoint of 20 °C through a TRNSYS simulation for a case study in Berlin, Germany. For the variable temperature settings, the supply temperature was kept at 8 °C during the heating period (from November to March) and 25 °C during the summer. During the intermediate seasons, the temperature setpoint was set to 20 °C. The simulation results showed that although the variable temperature setpoint strategy could lead to an increase of 8.5% in electrical demand at the substation level, a 21% reduction in electrical consumption of centralized heat pumps was achieved.

Brunt et al. [71] proposed applying seasonally variable temperatures and mass flow rates to a double-pipe bidirectional 5GDHC network. The hot and cold supply temperatures were maintained at 13 °C and 8 °C in the winter, respectively, and at 27 °C and 22 °C in the summer. What differs from the aforementioned studies is that during the shoulder seasons, supply temperatures were linearly scaled to follow outdoor air temperature trends. Compared with the reference case with constant supply temperature settings, the novel system achieved a higher annual system Coefficient of Performance (COP) and 32.16% annual emissions reduction.

From these reviewed case studies, it can be observed that there is a disparity in the settings of seasonal setpoint control strategy across various studies. While some studies advocated for lowering operating temperatures in summer and raising them in winter, others proposed the other direction, opting for higher summer temperatures and lower winter setpoints. This diversity likely arises from the opposing dynamics between the energy consumption of substations and temperatures, as compared to the relationship between the energy consumption of energy hubs and temperatures.

To elaborate, during periods when cooling demand dominates heating, which is often the case in summer, opting for lower operating temperatures can enhance energy efficiency in substations because it increases the COP of substations' chillers. Particularly, for substations equipped with heat exchangers, lowering the operating temperature to a point where direct cooling through heat exchangers is feasible can potentially eliminate the need for chiller operations, thus further reducing energy consumption in substations. However, maintaining lower operating temperatures is often not favorable for the cooling efficiency of energy hubs, as it might hinder the integration of free cooling sources such as cooling towers and decrease the COP of mechanical cooling devices.

2.3.3. Free-floating with limits control

The free-floating with limits control strategy permits the distribution network temperature to fluctuate within a defined range as the substation's demand changes. When the overall heating demand predominates over cooling, the network distribution temperature increases until reaching the upper limit, at which point auxiliary cooling devices activate to lower the temperature back within the range. Conversely, when the cooling demand surpasses heating, the distribution temperature tends to decrease until it reaches the lower limit, activating the auxiliary heating plant to raise the temperature. As long as the network temperature remains within this predefined range, the energy hub remains bypassed, and auxiliary heating and cooling systems can remain deactivated. The core concept behind the free-floating temperature control strategy is to leverage the synergy of simultaneous heating and cooling demand to minimize the reliance on auxiliary heating and cooling systems. This strategy's operational stability and simplicity have made it a popular choice in numerous projects and studies as listed in Table 5.

As indicated in Table 5, the free-floating temperature control strategy is widely employed in 5GDHC systems with diverse system configurations, climate zones, and thermal energy sources. Due to its broader temperature range, it can be applied to a wider range of thermal energy sources that may not have stable temperatures, such as shallow geothermal sources and dry coolers, in comparison to other temperature control strategies.

When it comes to determining the operating temperature range for the free-floating temperature control strategy, there are currently few established standards or guidelines. It is widely accepted that the lower limit of the temperature range should be above the freezing point to avoid freezing, typically being higher than 0 °C when water is used as

Table 5
Review of 5GDHC systems using free-floating with limits control strategy.

Ref.	*Location*	*Network configuration*	*HCDR*	*Thermal Source*	*Limits*	*Direct cooling considered*	*Temperature differential*
[69]	San Francisco, USA Cologne, Germany	II, III	0.40	Solar, river, lake or ocean	Cold: [8 °C,16 °C] Warm: [12 °C,20 °C]	N	4 °C
[72]	San Francisco, Phoenix, and Denver	III	Not mentioned	Air source heat pumps and chillers	Cold: [8 °C,16 °C] Warm: [12 °C,20 °C]	Y	4 °C
[55]	Lund, Sweden	III	0.42	Not mentioned	[20 °C,32 °C] [16 °C,36 °C] [24 °C,28 °C]	N	5 °C
[54]	Toronto, Canada	I, III	0.45	Sewage water, cooling tower	Cold: [8 °C,16 °C] Warm: [12 °C,20 °C]	N	4 °C
[53]	Sweden	I	0.70	Sewage water, geothermal	[6 °C,17 °C]	Y	4 °C
[73]	Lund, Sweden	III	0.41	Not mentioned	Cold: [6 °C,30 °C] Warm: [16 °C,40 °C]	Y	10 °C
[74]	Sweden	III	0.56	Air source heat pump, cooling tower	Cold: [8 °C,28 °C] Warm: [18 °C,38 °C]	Y	10 °C
[56]	Pantelleria Island, Italy	IV	Not mentioned	Sea water heat pump	[6 °C,17 °C]	N	Not applicable
[58]	Madrid, Spain	IV	0.60	Ground source heat pump	Ring 1: [15.5 °C,19 °C] Ring 2: [20 °C,23.5 °C]	N	Not applicable
[75]	Chicago IL, USA	II	0.21	Geothermal, cooling tower	[9 °C,16 °C] (heating season) [5 °C,16 °C] (outside of the heating season)	Y	4 °C
[76]	Seville, Spain	III	0.43	Compression chiller, heat pump	[6 °C, 25/40/55/70 °C]	Y	~3 °C
[77]	Spain	III	−0.33	Air source heat pump, compression chiller	Cold: [6 °C,14 °C] Warm: [6 °C,40 °C]	Y	Not mentioned
[23]	Spain	III, IV	0.43	Coaxial borehole heat exchangers	[6 °C,40 °C]	Y	~3 °C

the circulating medium. However, opinions differ regarding the determination of the upper limit. Some studies argued for a lower upper limit to enable direct cooling at substations with maximal energy savings [22], while other studies suggested that maintaining a higher maximum temperature limit could lead to better system performance [27].

Research on the influence of different temperature limits on the system performance is still limited. Blacha et al. [55] conducted a case study where three temperature limit combinations were simulated and compared within a bidirectional low-temperature network. The results indicated that a smaller spread between limit values (24/28 °C) could increase heat demand but decrease total electricity demand from decentralized heat pumps and chillers. However, this study did not account for the energy consumed by the central balancing unit. Quirosa et al. [76] analyzed the impact of four different upper limits (i.e., 25 °C, 40 °C, 55 °C, 70 °C) on the 5GDHC system. Their findings showed that temperatures around 40 °C resulted in the lowest annual grid energy consumption while allowing the cold pipes to maintain a maximum temperature of 14 °C for a constant direct cooling led to the highest energy consumption. However, this study lacked considerations of different system configurations and load conditions.

From the literature review summarized in this subsection, we have identified a list of factors that should be considered when determining temperature limits for free-floating temperature control:

1) Substation Configuration: When substations within the system are equipped with heat exchangers, it is common practice to establish a lower upper limit to enable direct cooling within the substation. This approach helps minimize energy consumption for substation cooling purposes [78].
2) Integration of Renewable Energy Sources: Adjusting temperature limits to align with renewable energy availability is crucial. This allows for a better utilization of renewable energy resources.
3) Sequence of Multiple Thermal Energy Sources: 5GDHC systems with multiple energy sources benefit from adopting different temperature limits to dictate the order of energy source activations. Typically, wider temperature spreads are employed for renewable energy sources and thermal energy storage, prioritizing their utilization when auxiliary heating or cooling is required. Conversely, narrower spreads are applied for mechanical heating and cooling devices.
4) Overall System Performance: Ideal temperature limits strike a balance between meeting substation requirements, ensuring users' thermal comfort, and maximizing energy efficiency and emissions reductions.

It is important to note that while the free-floating temperature control strategy theoretically reduces the operational time of auxiliary heating and cooling devices, this does not necessarily guarantee that 5GDHC systems using this strategy are inherently energy efficient because the temperature-dependent efficiencies of heat pumps and chillers are not considered [32]. The free-floating temperature control strategy tends to maintain a high network temperature during periods when the cooling demand is prevalent and a low network temperature during periods with dominant heating demand. As a consequence, substations within the system often operate less efficiently, which can lead to an increase in overall system energy consumption.

2.3.4. Other temperature control strategies

The temperature control strategies mentioned in subsections 2.3.1–2.3.2 are the most commonly employed rule-based approaches used in 5GDHC systems. However, beyond these strategies, there are more complex temperature control strategies that, while not yet widely adopted, have demonstrated good system performances. This subsection aims to provide a brief introduction to these temperature control strategies.

Quirosa et al. [76,77] introduced a storage strategy for 5GDHC systems integrated with Photovoltaics (PV) modules. Unlike the conventional free-floating temperature control strategy, where the energy hub only operates when the distribution temperature reaches certain limits, this strategy utilizes excess electricity generated from PV modules to run the auxiliary heating and cooling devices in the energy hub to either increase (when heating demands predominate) or decrease (when cooling demands predominate) the distribution temperature even when the network temperature is within the designed limits. In this way, this control strategy effectively converts PV excess into thermal energy and stores it within the system's network. Since the energy hub's electricity usage is covered by PV modules, this control strategy does not escalate the operational cost of the energy hub. Instead, it enhances the overall system efficiency by utilizing the stored energy within the network and providing more favorable inlet temperatures to substations.

While this control strategy has primarily been applied to 5GDHC systems integrated with PV modules, it has the potential for broader applications even in systems without PV fields. For instance, in 5GDHC systems operating in regions with time-of-use (TOU) electricity pricing, pre-cooling or pre-heating during off-peak electricity periods could be implemented. This involves leveraging the inherent heat capacity of the pipeline network or employing additional thermal energy storage to store excess cooling or heating. These stored resources can then be deployed during peak electricity periods. However, there is a limited depth of exploration on this topic in existing research and practice.

Bünning et al. [69] proposed a method to optimize the temperature setpoint for the network. The optimization was conducted based on the assumption that the optimized temperature setpoint is proportional to the nth power of the normalized heating and cooling loads. The simplex method Nelder-Mead was used to solve the optimization problem. The case study showed that 15% less electricity consumption could be achieved by the temperature setpoint optimization compared with the case with free-floating temperature control strategy. However, this method also assumed that non-balanced heat in the distribution network can be fully covered by renewable energy or storage that shifts loads, so only the electricity consumed by the substation heat pumps and chillers was considered when calculating the whole system energy consumption.

Wirtz et al. [32] proposed a mixed-integer linear program (MILP) for the short-term optimization of the network temperature in 5GDHC systems with the objective function of minimizing the operation costs of the entire system. Results from this case study indicated significant cost savings in two out of three investigated months, with savings reaching 10% and 60%, respectively, compared to the baseline scenario employing the free-floating temperature control strategy.

3. Case study

To demonstrate the distinctions among the various temperature control strategies employed in 5GDHC systems outlined in Section 2.3, a case study that involved three temperature control strategies, each with eight settings, as listed in Table 6, was designed. Notably, the temperature control strategies discussed in Section 2.3.4 have been excluded

Table 6
Case design for different temperature control strategies.

Temperature control strategy	Variable	Settings
Constant operating temperature	Operating temperature setpoint	From 12 °C to 40 °C, at an interval of 4 °C
Free-floating with limits	Upper limit	From 12 °C to 40 °C, at an interval of 4 °C
Multi-stage operating temperature	Operating temperature setpoint	Group 1: Summer: constant at 16 °C, Winter: from 28 °C to 40 °C, at an interval of 4 °C Group 2: Winter: constant at 16 °C Summer: from 28 °C to 40 °C, at an interval of 4 °C

from the case study due to their challenges in application or specific configuration requirements.

The network's operational temperature range is established between 12 °C to 40 °C, based on the operating temperatures of 5GDHC systems from the literature review. For the constant temperature control strategy, eight setpoints within this range were considered utilizing 4 °C intervals. For the free-floating temperature control strategy, similar settings were considered, but the upper limit was varied instead of changing the operating temperature setpoint. The lower limit was consistently maintained at 4 °C to prevent water from freezing.

For the multi-stage temperature control strategy, 8 operating temperature settings were proposed and divided into 2 groups to represent the two distinct patterns identified in the literature. In the first group, a lower temperature setpoint was employed during summer, and a higher temperature setpoint was used during winter. Conversely, in the second group, a higher temperature setpoint was employed during summer, and a lower temperature setpoint was used during winter. The lower setpoint was kept constant at 16 °C as this allows for the substation direct cooling, while the higher setpoint varied from 28 °C to 40 °C, with intervals of 4 °C. In our case study, the summer period spanned from June to August, the winter period from December to February, and the remaining months were considered transition periods. The operating temperature setpoint during transition periods was determined through linear interpolation between the summer and winter setpoints. Fig. 7 illustrates the operating temperature setpoint profiles with a multi-stage operating temperature control strategy for the entire year.

To explore the performance of different temperature control strategies in various system configurations, the two most common 5GDHC system configurations described in Subsection 2.2.2 were studied in the case study, i.e., unidirectional parallel networks (Category II) and bidirectional parallel networks (Category III). It should be noted that the same temperature settings may imply different meanings for these two different network configurations. For unidirectional systems, the operating temperature represents the supply temperature, or in other words, the leaving temperature from the energy hub. For bidirectional systems, as there are no constant "supply" or "return" pipes in the traditional sense, the operating temperature setpoint, in this case, represents the warm pipe temperature setpoint, thus cold pipe temperature should be maintained at the operating temperature setpoint minus the design temperature difference.

3.1. Modelica-based simulation environment

A virtual testbed of the 5GDHC system was developed using Dymola 2023, a modeling tool based on the Modelica language [79]. Modelica is a well-established equation-based, object-oriented modeling language renowned for its effectiveness in modeling physical systems and controls. It has previously demonstrated success in modeling and simulating district energy systems [37,72,74]. The fundamental component models employed within the virtual testbed primarily originate from the Modelica Building Library [80], which offers dynamic simulation models for building and district energy and control systems.

Fig. 8 illustrates the Modelica diagram view of the studied 5GDHC system. In both studied network configurations, the system comprises three substations and one energy hub. Annual simulations were conducted with a one-hour output interval, utilizing the CVODE solver [81] with a tolerance of 1×10^{-6}. Here, we provide a concise overview of the modeling process for the primary components within the systems, including substations, energy hubs and distribution pipelines. More detailed parameter values adopted in the models are provided in the tables in Appendix A.

1) Substation

In each substation, it is assumed that one water source heat pump is designated for heating and one chiller for cooling. Their operational COP can be calculated using Eq. (2).

$$COP_{heating} = \eta_{Carnot} \times \frac{T_{cond}}{T_{cond} - T_{evap}}$$

$$COP_{cooling} = \eta_{Carnot} \times \frac{T_{evap}}{T_{cond} - T_{evap}} \tag{2}$$

where T_{cond} is the temperature of the condenser, T_{evap} is the temperature of the evaporator, and η_{Carnot} is Carnot effectiveness.

Additionally, each substation is equipped with a heat exchanger that operates in parallel with the chiller. If the entering temperature is lower than the predefined setpoint, the substations can engage in the "direct cooling" mode, where cooling is exclusively provided by the heat exchangers. Conversely, when the entering temperature exceeds this setpoint, the "active cooling" mode is initiated, activating the chillers to provide cooling. In our case study, the building's chilled water supply temperature setpoint is set at 18 °C and the mode transition setpoint is established at 16.5 °C with a safety margin of 1.5 °C considered. Apart from the heating and cooling devices, distributed pumps are installed to ensure there is enough water flow for each substation. These pumps are controlled to maintain the design temperature difference of the substations, which is set to be 4 °C in the case study, a widely used setting according to the literature review.

Load profiles are defined by users as input information to the substations. In our study, the building loads come from simulations of prototype building models developed by the U.S. Department of Energy (DOE) [82] using EnergyPlus. Three commercial building types are involved in the case systems, including large office, hospital and high-rise apartment. The load of the high-rise apartment is multiplied by 10 times, loosely representing ten high-rise apartments that have similar load profiles, to balance the load from different building types. The domestic hot water load and space heating load are aggregated together as the total heating demand.

2) Energy hub

In this case study, the energy hub comprises an air-cooled chiller for auxiliary heating, an air-source heat pump for auxiliary cooling, and a variable-speed dry cooler to harness the "free" thermal energy provided by ambient air for auxiliary heating and cooling, as illustrated in Fig. 9.

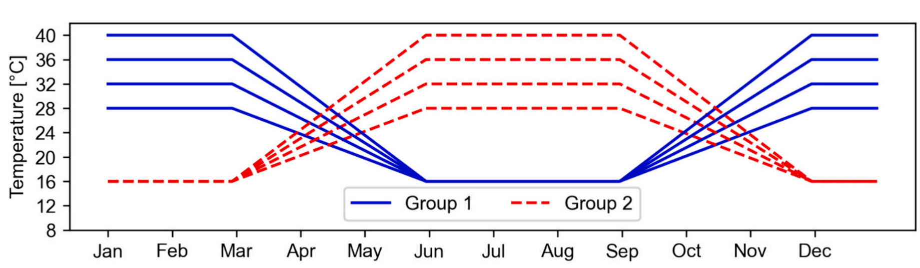


Fig. 7. Temperature setpoint profiles for the multi-stage operating temperature control strategy.

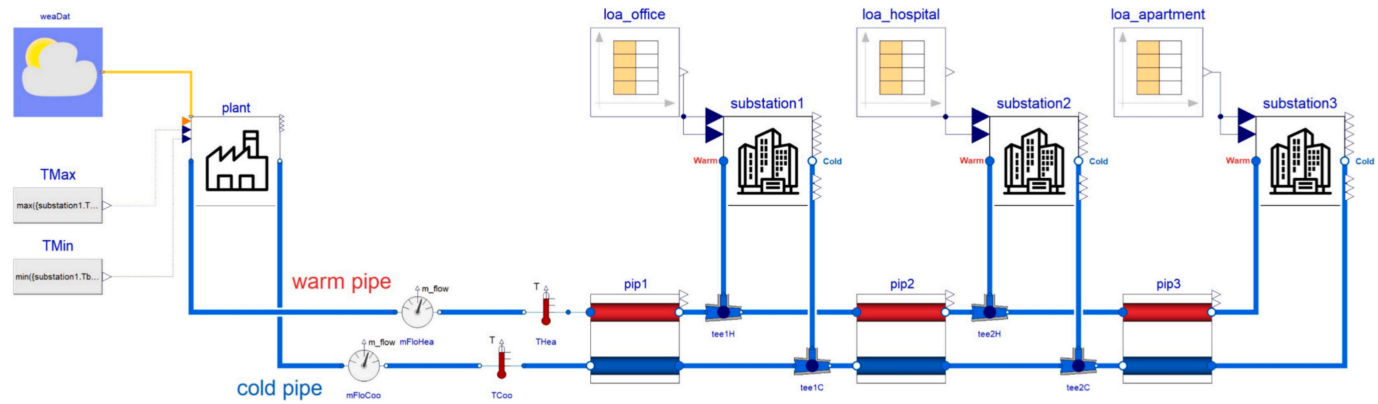


Fig. 8. Modelica diagram view of the studied 5GDHC system.

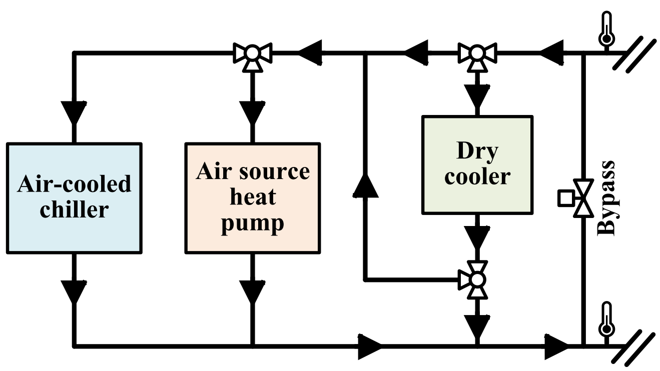


Fig. 9. Schematic representation of the energy hub configuration.

The decision to use a dry cooler instead of a cooling tower was motivated by its ability to prevent the introduction of pollutants and oxygen into the network [75], and its capability of acting as heat source or heat sink depending on ambient conditions and the system's requirements [18]. Different dry cooler control strategies are employed for different system temperature control strategies, as shown in Fig. 10. When the free-floating temperature control strategy is in use, the dry cooler controller employs a constant approach temperature (ΔT_{app}) control strategy to fully exploit the "free" heating or cooling when outdoor air temperature (T_{air}) is appropriate. On the other hand, when the constant or multi-stage temperature control strategy is employed, the dry cooler fan speed is adjusted to align the outlet temperature with the specified supply temperature setpoint as defined in Table 7. If using the dry cooler alone cannot meet the requirement, the air-source heat pump or air-cooled chiller will be activated to further increase or decrease the temperature to the setpoints. Both the air-source heat pump and air-cooled chiller operate based on the performance equation provided in Eq. (2).

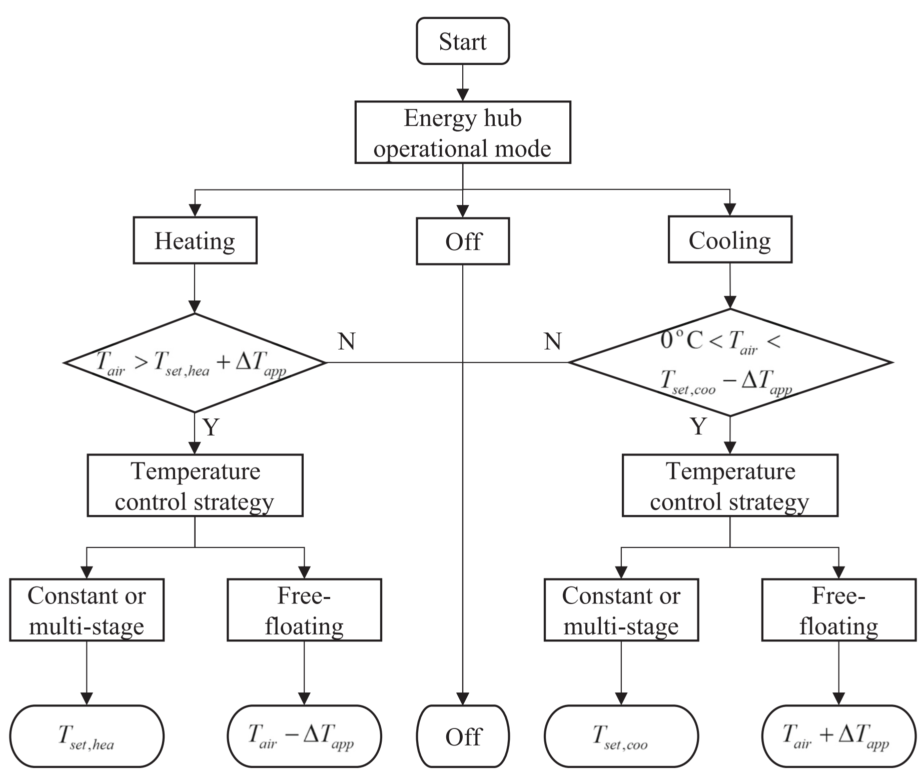


Fig. 10. Dry cooler operational status and supply temperature setpoint control logic.

Table 7
Energy hub activation conditions in different scenarios.

Temperature control strategy	Network configuration	Monitoring location	Heating activation conditions	Heating supply temperature setpoint ($T_{set,hea}$)	Cooling activation conditions	Cooling supply temperature setpoint ($T_{set,coo}$)
Constant / Muti-stage	II	·Energy hub entering temperature	$T_{ent,hub} < T_{set}$	T_{set}	$T_{ent,hub} > T_{set}$	T_{set}
	III	·Energy hub - warm pipe connection ·Energy hub - cold pipe connection	$\dot{m}_{warm,hub} > 0$ & $T_{cold,hub} < T_{set,cold}$	$T_{set,warm}$	$\dot{m}_{cold,hub} > 0$ & $T_{warm,hub} > T_{set,warm}$	$T_{set,cold}$
Free-floating	II	·All substations leaving temperature	$\exists T_{lvg,sub} < T_{lmt,low}$	$max(T_{lmt,low} + \Delta T, T_{air} - \Delta T_{app})$	$\exists T_{lvg,sub} > T_{lmt,high}$	$min(T_{lmt,high} - \Delta T, T_{air} + \Delta T_{app})$
	III	·Substations - warm pipe connection ·Substations - cold pipe connection	$\sum \dot{m}_{warm,sub} > 0$ & $\exists T_{cold,sub} < T_{lmt,low}$	$max(T_{lmt,low} + \Delta T, T_{air} - \Delta T_{app})$	$\sum \dot{m}_{cold,sub} > 0$ & $\exists T_{warm,sub} > T_{lmt,high}$	$min(T_{lmt,high} - \Delta T, T_{air} + \Delta T_{app})$

3) Distribution pipeline

The distribution pipe model used in this study is developed from the buried pipe model described in Abugabbara et al. [74]. The pipe model is capable of simulating both hydraulic pressure drops and heat transfer to the surroundings. In this study, the pipes are sized based on the criterion that pressure losses are 100 Pa per meter at design flow rates [35]. The pressure drop (Δp) can be calculated using Eq. (3).

$$\dot{m} = k\sqrt{\Delta p} \tag{3}$$

Where $\dot{m}$ represents the mass flow rate and k denotes the constant flow coefficient which can be calculated based on the nominal parameters.

For heat transfer from the pipe to the surroundings, an equivalent resistance model from van der Heijde et al. [83] is adopted to calculate the heat transfer to the surroundings, considering the interaction between pipes buried together. The undisturbed soil temperature ($T_{s,z}$) can be calculated based on Eq. (4) [84].

$$T_{s,z} = T_{ms} + A_s e^{-z\sqrt{\frac{\pi}{\alpha\tau}}} sin\left(\frac{2\pi(t - t_{lag})}{\tau} - z\sqrt{\frac{\pi}{\alpha\tau}}\right) \tag{4}$$

Where t is Julian date, days; $t_{lag}$is phase lag of soil surface temperature, days; z is depth, m; τ is annual period length, 365 days; α is thermal diffusivity of the soil, m^2/day T_{ms} is mean annual surface temperature, °C; A_s is surface temperature amplitude, °C.

Fig. 11 displays the hourly ambient air and soil temperature profile using ASHRAE climate zone 4C as an example.

3.2. Implementation of various temperature control strategies

This subsection introduces the implementation of various temperature control strategies within the 5GDHC system models. Since substation control remains consistent, the variations in temperature control strategies primarily manifest in the control methods applied to the energy hub. Table 7 provides a comprehensive overview of the energy hub activation conditions and supply temperature setpoints with different temperature control strategies.

For the constant temperature control strategy, the energy hub's inlet temperature ($T_{ent,hub}$) is continuously monitored in real-time and serves as the primary indicator for the energy hub's operational status. If it exceeds the predetermined setpoint (T_{set}), the cooling plants are activated to lower the water temperature to the specified setpoint. Conversely, if it falls below the setpoint, the heating plants are engaged to raise the water temperature to the desired setpoint. When this constant temperature control strategy is applied to bidirectional flow systems, as the flow direction may change according to the dominant load, the inlet temperature at the connection points between the energy hub and both the cold and warm pipes must be monitored. When the overall heating demand prevails cooling, the energy hub's inlet temperature should correspond to the temperature at the connection point with the cold pipe ($T_{cold,hub}$). If this temperature is lower than the cold pipe temperature setpoint ($T_{set,cold}$), the cooling plant will be activated. Conversely, when cooling demand dominates, the energy hub's inlet temperature should be the temperature at the connection point with the warm pipe ($T_{warm,hub}$). If this temperature exceeds the setpoint for the warm pipe ($T_{set,warm}$), the heating plant will be activated.

For the free-floating temperature control strategy, to ensure that the temperature across the entire system remains within the defined range,

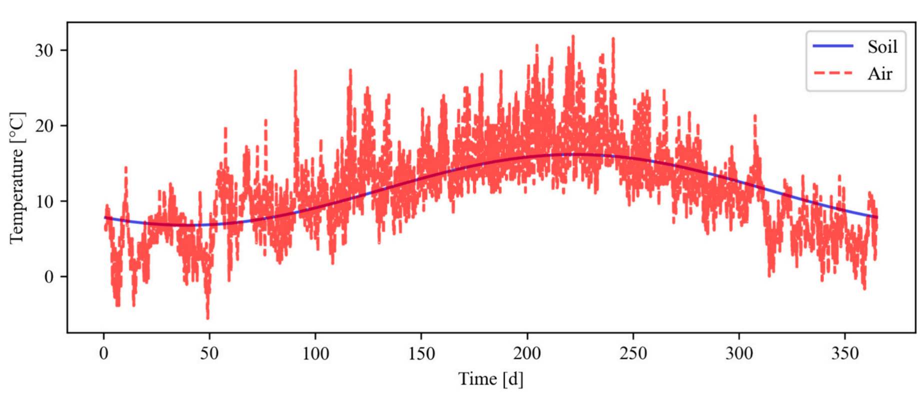


Fig. 11. Hourly ambient air and undisturbed soil temperature profile (climate zone: 4C).

temperature monitoring is extended to cover all substations, rather than solely focusing on monitoring the entering temperature of the energy hub. If the leaving temperature of any substation ($T_{lvg,sub}$) falls below the lower limit ($T_{lmt,low}$), the heating plant is activated, and if it exceeds the upper limit ($T_{lmt,high}$), the cooling plant is activated. For bidirectional systems, real-time temperature monitoring is required at all connection points between substations and both the cold and warm pipes. If the temperature at any substation's connection point with the cold pipe ($T_{cold,sub}$) drops below the lower limit, the heating plant is activated. If the temperature at any substation's connection point with the warm pipe ($T_{warm,sub}$) exceeds the upper limit, the cooling plant is activated.

In determining the supply temperatures of heating and cooling devices within the energy hub, the primary aim is to ensure that the network temperature aligns with the defined temperature control strategy while minimizing mechanical heating and cooling usage in the energy hub. Consequently, heating and cooling devices within the energy hub have their temperature setpoints positioned at the highest allowable temperature range (for cooling) or the lowest allowable temperature range (for heating) as specified by the temperature control strategy. Notably, when employing the free-floating temperature control strategy and 'free' heating or cooling is available through the dry cooler, the priority is given to the dry cooler for heating or cooling provision. This leads to adjustments in the supply temperature setpoint, as depicted in Fig. 10.

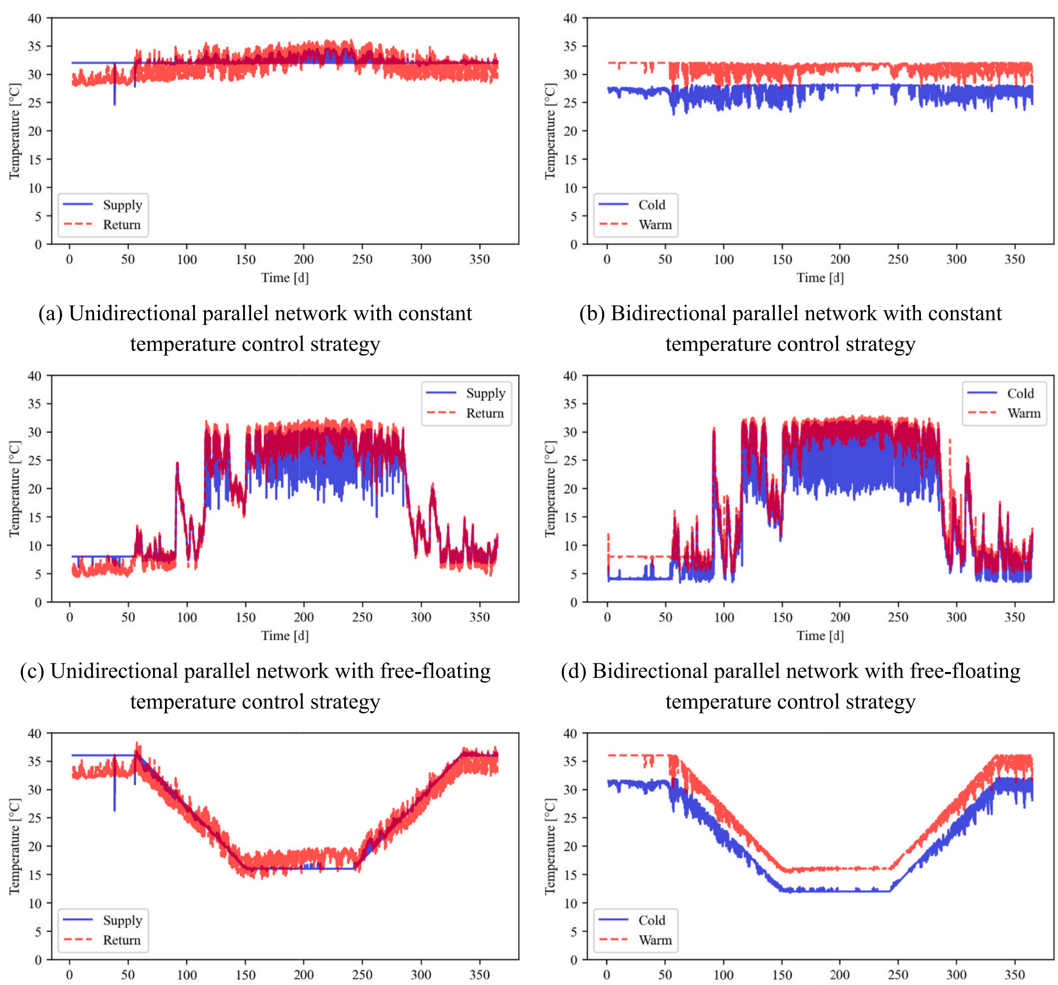
Fig. 12 illustrates annual temperature profiles simulated under different temperature control strategies for the two studied network configurations. It can be observed that in all simulation scenarios, the system's operating temperature generally aligns with the specified pattern, confirming the functionality of the developed virtual testbed. However, some temperature deviations can also be noticed. These deviations are attributed to the implemented delay timer. In our case study, a time threshold of 10 min is adopted to prevent short cycling of heating and cooling equipment. If the equipment's running time is <10 min, the equipment will continue running even if it receives a signal to shut down. While this may result in a slight deviation from the set strategy for system operating temperatures, it effectively mitigates issues caused by frequent starts and stops of equipment.

3.3. Performance indicators

In this study, five performance indicators were selected to evaluate the system including total energy consumption, system coefficient of performance (COP), carbon emissions, peak electricity demand, and plant sizing.

3.3.1. Total energy consumption

The total energy consumption, as shown in Eq. (5), is one of the most commonly used indicators to evaluate the district energy system



(a) Unidirectional parallel network with constant temperature control strategy

(b) Bidirectional parallel network with constant temperature control strategy

(c) Unidirectional parallel network with free-floating temperature control strategy

(d) Bidirectional parallel network with free-floating temperature control strategy

(e) Unidirectional parallel network with multi-stage temperature control strategy

(f) Bidirectional parallel network with multi-stage temperature control strategy

Fig. 12. Annual temperature profiles simulated under different temperature control strategies for the two studied network configurations.

performance as it is directly related to the system's operational cost [85]. For district energy systems, the annual system energy consumption can be divided into two parts: substation energy usage (E_{sub}) and energy hub energy usage (E_{hub}). The main energy-consuming equipment in the substation includes heat pumps, chillers and distributed pumps, while for the energy hub, the main energy-consuming equipment includes heating and cooling plants and pumps.

$$E_{tot} = E_{sub} + E_{hub} = \sum_{t_0}^{t_N} P_{sub}(t_i) \times \Delta t + \sum_{t_0}^{t_N} P_{hub}(t_i) \times \Delta t \quad (5)$$

Where N is the total sampling number for each operation timestamp t. P_{sub} and P_{hub} are the real-time power of substations and energy hub, respectively. Δt is the time interval. For the case study, since there are only electricity-using devices in the 5GDHC systems, the total energy usage equals the total electricity consumption.

3.3.2. System coefficient of performance (COP)

The annual system COP is defined as the ratio of the total thermal load (heating and cooling) to the required system input energy, as expressed as Eq. (6). A higher COP value is generally preferred as it indicates that the system is more energy-efficient because it can deliver more heating or cooling for a given energy input.

$$COP = \frac{\sum_{t_0}^{t_N} \left[\dot{Q}_{dem,coo}^{tot}(t_i) \times \Delta t + \dot{Q}_{dem,hea}^{tot}(t_i) \times \Delta t \right]}{E_{tot}} \quad (6)$$

Where $\dot{Q}_{dem,coo}^{tot}(t_i)$ is the total cooling demand at the time stamp t_i and $\dot{Q}_{dem,hea}^{tot}(t_i)$ is the total heating demand at the time stamp t_i.

3.3.3. Carbon emissions

Carbon emissions, often referred to as CO_2 emissions, entail the release of carbon dioxide gas into the atmosphere and can be estimated using Eq. (7). This performance indicator reflects the environmental impact of the 5GDHC system.

$$E_{CO_2} = c_{ele} \times E_{tot} \quad (7)$$

Where c_{ele} is the CO_2 emission factor, which reflects the CO_2 emission per electricity consumed on the site. According to the statistics provided by EIA [86], this factor is determined as 0.389 kg per kWh.

3.3.4. Peak electricity demand

The peak electricity demand, as calculated by Eq. (8), refers to the highest level of electricity that is consumed by the whole system within a specified period. Peak demand has significant implications for power systems and energy supply, as the system must be designed to be able to meet the peak demand to ensure the stability and reliability of power supply. Considering the 5GDHC systems in our case study are totally electricity-driven, this indicator will highly influence the system's total capital cost and energy prices.

$$P_{peak} = \max_{t_0 \le t \le t_N} P_{tot}(t_i) = \max_{t_0 \le t \le t_N} \left(P_{sub}(t_i) + P_{hub}(t_i)\right) \quad (8)$$

3.3.5. Plant capacity

Plant sizing is of great significance as an oversized plant results in higher capital and operational costs, while an undersized plant cannot meet the load requirements. In the study, the capacities of heating and cooling plants ($\dot{Q}_{cap}$) are determined based on fulfilling the peak heating or cooling demand ($\dot{Q}_{dem}$), as shown in Eq. (9).

$$\dot{Q}_{cap} = SF \times \max_{t_0 \le t \le t_N} \left(\dot{Q}_{dem}(t_i) \right) \quad (9)$$

where SF represents safety factor with 1.0 for default.

The plants sized in the case study consist of heat pumps, chillers in the substations, as well as the air-source heat pump and air-cooled chiller in the energy hub. The total system cooling capacity ($\dot{Q}_{cap,coo}$) encompasses the water-chillers chillers in substations ($\dot{Q}_{cap,sub,coo}$) and the air-cooled chiller in the energy hub ($\dot{Q}_{cap,hub,coo}$), which can be calculated using Eq. (10).

$$\dot{Q}_{cap,coo} = \dot{Q}_{cap,sub,coo} + \dot{Q}_{cap,hub,coo} \quad (10)$$

where $\dot{Q}_{cap,sub,coo}$ stands for the total cooling capacity of heat pumps in the substations and $\dot{Q}_{cap,hub,coo}$ stands for the total cooling capacity of the heat pump in the energy hub.

Similarly, the total system heating capacity ($\dot{Q}_{cap,hea}$) equals the sum of the heating capacity of the heat pumps in substations ($\dot{Q}_{cap,sub,hea}$) and the heat pump in the energy hub ($\dot{Q}_{cap,hub,hea}$).

It is important to mention that as the heating demands of water source heat pumps in substations are predefined inputs, their sizing is relatively straightforward and thus not discussed in this study. However, for other components within the system, such as chillers and heat pumps in the energy hub, their demands are unknown without any analysis. Therefore, a pre-simulation is necessary to ensure that the capacities of all these devices are sufficiently large to meet the required heating or cooling demands.

4. Results and discussions

4.1. Comparison under the same climate zone

In this subsection, various temperature control strategies are being implemented in 5GDHC systems located in Seattle, WA, which is in climate zone 4C. The annual cumulative heating demand is 4.53 GWh and the annual cumulative cooling demand is 4.38 GWh. Fig. 13 illustrates the hourly load profile for climate zone 4C.

The performance indicators, as detailed in section 3.3, have been calculated across various temperature control strategy settings and are depicted through color maps illustrated in Fig. 14 and Fig. 15. Each row corresponds to different settings within the context of three distinct temperature control strategies, while each column signifies different performance indicators. To vividly highlight the variations in performance indicators arising from distinct temperature control strategies, the indicator values have been normalized using Eq. (11) and then represented with varying cell colors. The color bar scale ranges from 0 (minimum normalized value) to 1 (maximum normalized value).

$$PI_{norm} = \frac{PI_{act} - PI_{min}}{PImin_{max}} \quad (11)$$

4.1.1. Unidirectional parallel network (category II)

Fig. 14 shows performance indicator values of the unidirectional parallel system under 18 different settings of three temperature control strategies as listed in Table 6.

In terms of **total energy usage/ system COP/ CO_2 emissions**, the best and worst-case scenarios both occur when the constant operating temperature control strategy is employed. Specifically, the constant operating temperature control strategy set at 16 °C demonstrates the best system performance. This is because this temperature setting represents the critical temperature point at which the direct cooling at substations is consistently enabled, leading to a significant reduction in substation energy consumption as presented in Fig. 14. In comparison to the worst-case scenario where the operating temperature is set at 40 °C constantly, the best case achieves a 16.9% reduction in both total energy consumption and carbon emissions. This underscores the significance of the role of temperature control strategies.

In the case of the free-floating temperature control strategy, its best setting does not outperform the constant supply temperature control strategy in terms of system performance. However, it exhibits better resilience to different temperature settings. Several factors contribute to

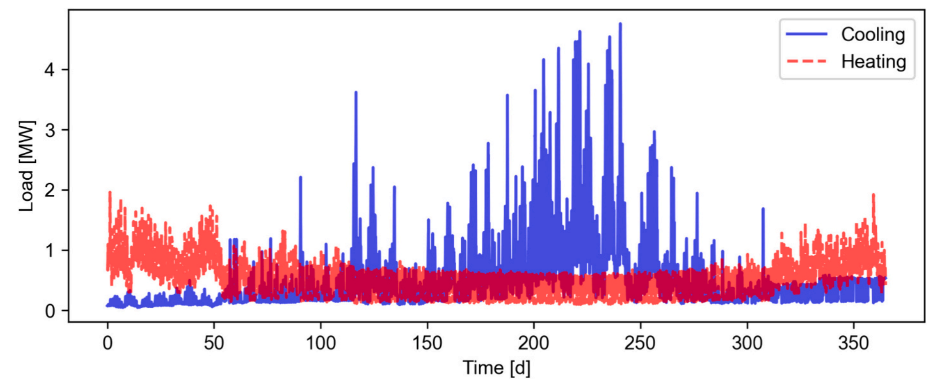


Fig. 13. Annual hourly demand profile (climate zone: 4C).

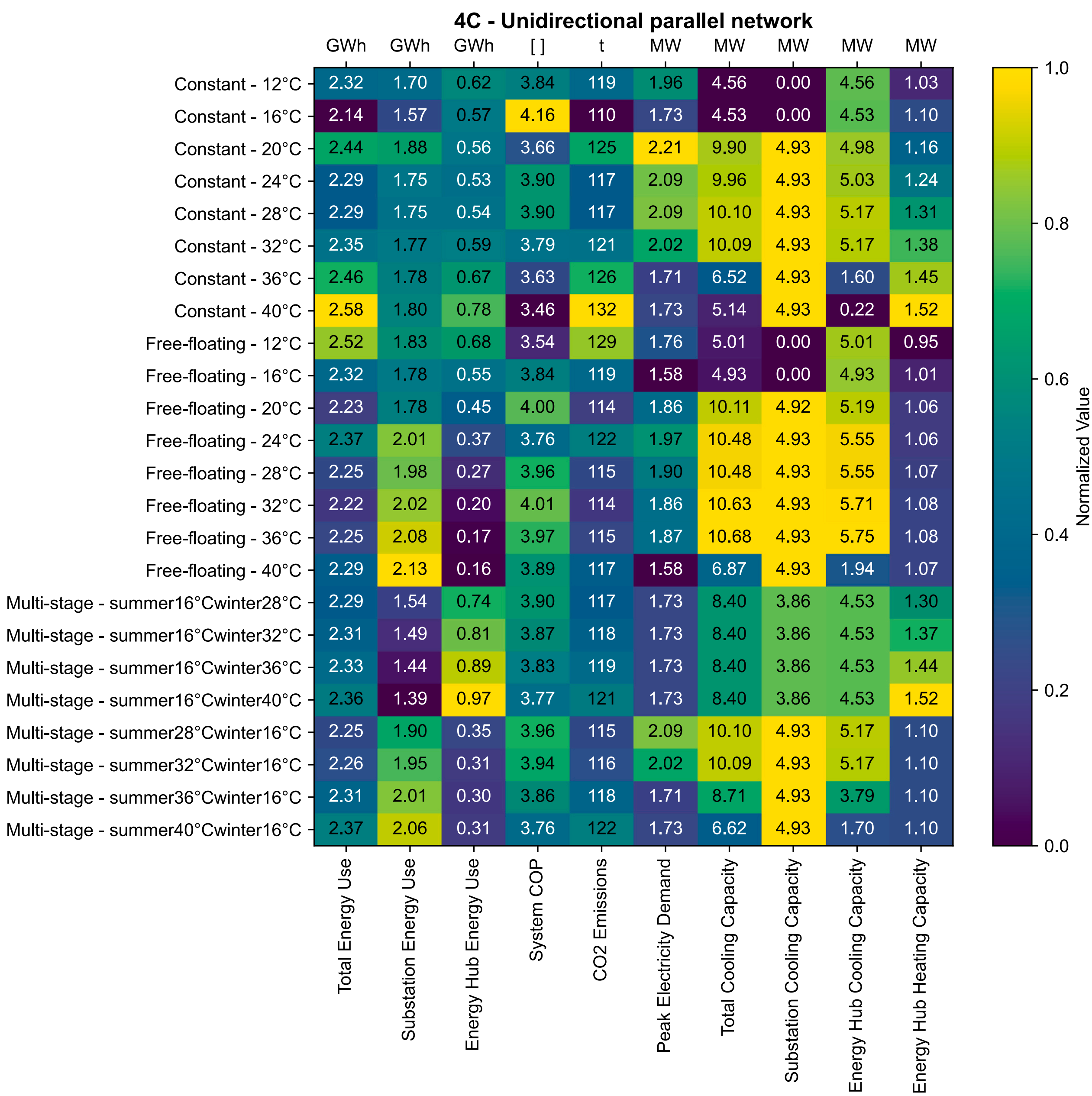


Fig. 14. Performance indicators of the unidirectional parallel system under different temperature control strategies.

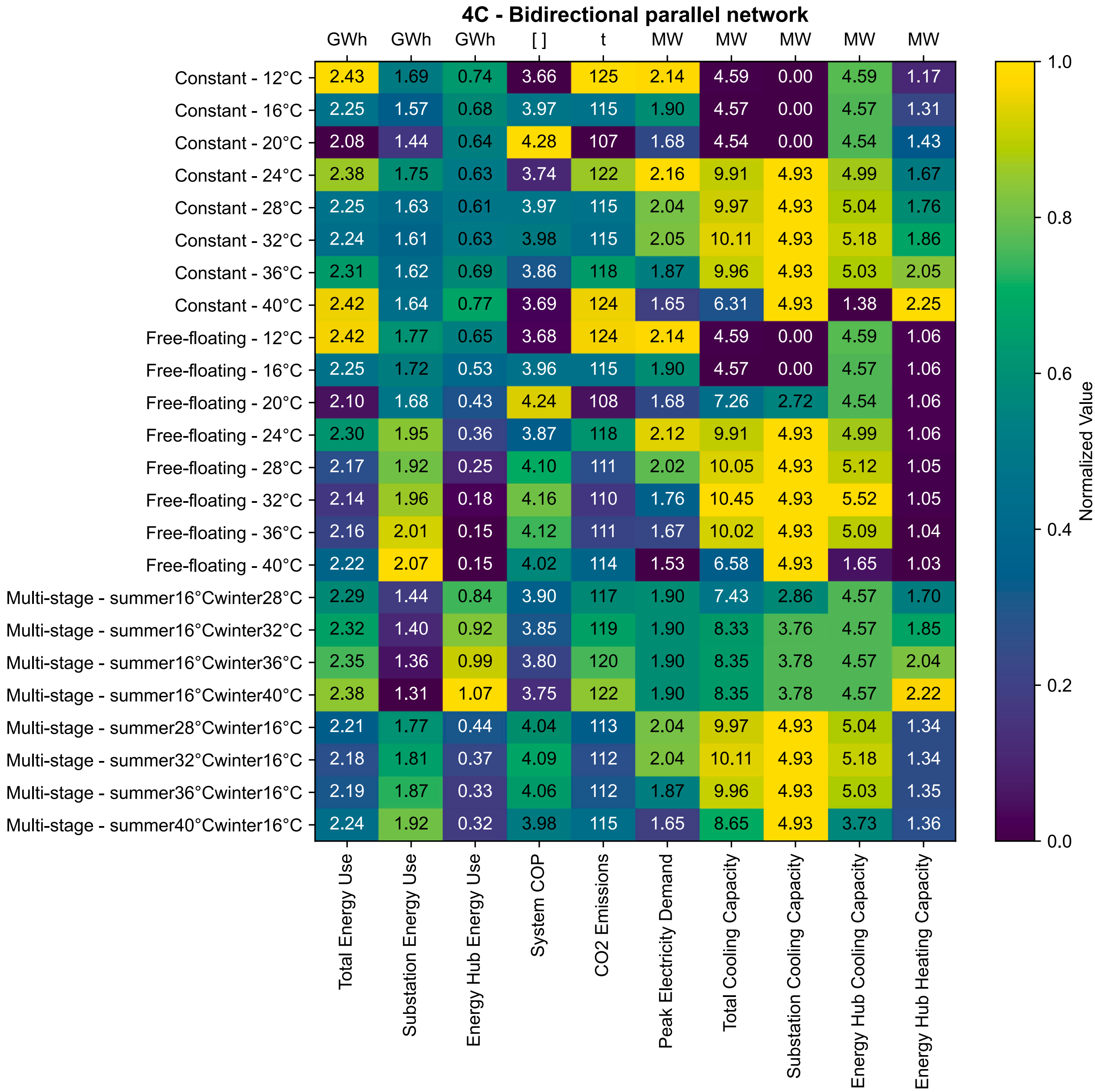


Fig. 15. Performance indicators of the bidirectional parallel system under different temperature control strategies.

this. Firstly, for the free-floating temperature control strategy, changes in the temperature upper limit only affect the system performance during cooling-dominant periods, while the performance during heating-dominant periods remains largely unaffected since the lower limit remains constant throughout the study. Besides, increasing the upper limit may reduce the COP of substation chillers, and it simultaneously enhances the system heating COP and increases the availability of the dry cooler. In addition, the wider temperature spreads resulting from the increased operating temperature upper limit can reduce the total operation time of the energy hub. As a result, the net effect, i.e. the total system energy use and carbon emissions, remains relatively stable as the upper limit changes. As illustrated in Fig. 14, increasing the upper limit results in higher substation energy consumption, lower energy hub energy consumption, and an overall energy usage trend that initially decreases and then increases. The best upper limit setting for the system with the free-floating control strategy is observed at approximately 32 °C.

In the case of the multi-stage temperature control strategy, none of the eight studied temperature settings has been able to achieve the best system performance attained by the constant temperature control strategy. This highlights the fact that simple seasonal setpoint control may not lead to improvements in the overall energy efficiency of 5GDHC systems. However, it is worth noting that the lowest substation energy usage is achieved when implementing the multi-stage temperature control strategy. This finding holds particular significance for 5GDHC systems that have access to inexpensive or free thermal energy sources, such as lake water or industrial waste heat. In such systems where energy hub operational costs can be regarded as negligible, substation energy usage becomes a critical determinant of total system energy consumption. When considering the two distinct group settings within this strategy (as shown in Table 6), it is evident that there is a significant difference in substation energy usage and energy hub energy usage between the two groups. However, the overall system performance remains similar, and neither pattern exhibits a clear advantage over the other.

Regarding the **system's peak electricity demand**, in all scenarios considered in the study, the peak electricity demand coincides with the peak cooling demand. The lowest peak load occurs when employing the free-floating temperature control strategy with either the upper limit set at 16 °C or 40 °C. This is because 16 °C is the maximum temperature that enables substation direct cooling, eliminating the need to operate chillers at substations. Setting the upper limit at 40 °C is also effective because this temperature is higher than the outdoor air temperature when the peak demand occurs, allowing the dry cooler to provide free cooling to the network instead of using the air-cooled chiller. However, in cases with temperature settings between these two extremes, both the substation chillers and energy hub chiller need to operate during peak cooling demand, leading to higher peak electricity demand. Overall, the peak electricity demand in cases with the constant temperature control strategy is higher than those with free-floating temperature control strategies under the same temperature setting. This is mainly because the wider temperature spreads of the free-floating temperature control strategy enable intermittent operation of energy hubs, reducing the hourly average power consumption. As for all cases with the multi-stage temperature control strategy, their peak electricity demands are the same as those with the corresponding constant temperature control strategies with their summer temperature setpoint. This is because peak electricity demand is solely determined by the network's temperature condition when the peak cooling demand occurs.

Regarding the **plant sizing,** both the constant temperature control strategy and the free-floating temperature control strategy show a pattern where the total cooling capacity is lower at both ends of the temperature setting range and higher in the middle of the range. In cases where the maximum operating temperature remains below 16 °C, the required cooling capacity for substation chillers becomes zero. This implies that there is no necessity to incorporate chillers in substations, resulting in significant cost savings in both initial installation and maintenance for the substations. For temperature control strategies with an operating temperature exceeding 16 °C, the direct cooling at the substations becomes unfeasible. Consequently, the required cooling capacity of substation chillers equals the peak cooling demand. When it comes to the higher end of the studied temperature range, the dry cooler takes on the primary role of providing cooling, which substantially reduces or even eliminates the need for the air-cooled chiller in the energy hub. For the multi-stage temperature control strategy, as the peak cooling demand happens in summer, the total system cooling capacity remains almost the same for the group 1 settings while in the case of the group 2 settings, the required total cooling capacity is lower as the summer temperature setpoint is raised higher.

Regarding the heating capacity of the heat pump in the energy hub, it can be noticed that as the temperature setting increases, the required heating capacity by the system with the constant temperature control strategy also increases. However, for the systems employing the free-floating temperature control strategy, the heating capacity remains almost constant because the change in the upper limit of the free-floating control strategy has a minimal influence on the system performance when the heating demand dominates over cooling. In the case of the multi-stage temperature control strategy, the group 1 settings demonstrate that as the winter temperature setpoint increases, the required heating capacity also increases, while the group 2 settings achieve similar heating capacity. However, the group 2 settings result in a lower required heating capacity compared to the group 1 settings.

4.1.2. Bidirectional parallel network (category III)

Fig. 15 shows performance indicator values of the bidirectional parallel system under 18 different settings of 3 temperature control strategies as listed in Table 6.

In terms of **total energy usage / system COP/ CO_2 emissions**, the best system performance is achieved with the constant temperature control strategy set at 20 °C, followed by the free-floating temperature control strategy with an upper bound of 20 °C. This can be attributed to the consistent utilization of substation direct cooling in these strategies. Regarding the multi-stage temperature control strategy, the group 2 settings (lower temperature setpoints in winter and higher setpoints in summer) outperform the group 1 settings (lower temperature setpoints in summer and higher setpoints in winter), despite the latter taking better advantage of free cooling at substations. However, the multi-stage temperature control strategy still falls short of achieving the best system performance.

When compared to the unidirectional parallel system, the best case's system performance improves by 2.82%, and the average system performance increases by 2.73%. A more detailed analysis of each temperature control strategy reveals a 2.68% average performance increase for the constant operating temperature control strategy, a 3.69% boost for the free-floating temperature control strategy, and a 1.84% enhancement under the multi-stage operating temperature control strategy. These findings demonstrate that bidirectional parallel systems tend to exhibit superior system energy performance as compared to unidirectional parallel systems even when the same temperature control strategy is employed.

In terms of **peak electricity demand**, the free-floating temperature control strategy with an upper bound of 40 °C demands the least electric power during the peak cooling demand. As the bidirectional parallel system can better leverage the simultaneous heating and cooling demand, the best peak electricity demand is reduced by 3.43% compared to the unidirectional parallel system. Examining each temperature control strategy individually, the best peak electricity demand decreases by 3.28%, 3.43%, and 3.29% for the constant operating temperature control strategy, the free-floating temperature control strategy, and the multi-stage operating temperature control strategy, respectively.

In terms of **plant sizing**, the trend in plant sizing with changes in the temperature setting exhibits a similar pattern to that of the unidirectional parallel system. The lowest total system cooling capacity is achieved when the network operating temperature is maintained below 16 °C, while the highest occurs when the operating temperature is around 32 °C during the peak cooling demand. It is worth noting that, in terms of plant sizing, the bidirectional parallel network doesn't demonstrate a better performance compared to the unidirectional parallel network, unlike other performance indicators. In comparison to the unidirectional parallel network, the lowest total cooling capacity of the bidirectional parallel network remains nearly constant (increases by 0.1%), while the lowest total heating capacity increases by 2.4%.

4.2. Comparison among different climate zones

To facilitate a comparative analysis of temperature control strategies across diverse climate zones, four representative climate zones with distinct characteristics in the U.S. are selected and outlined in Table 8. Demand Overlap Coefficient (DOC) [68] is introduced as a metric to evaluate the overlap of simultaneous heating and cooling demand profiles, which can be calculated by Eq. (12). The DOC ranges between 0 (no overlap) and 1(match exactly).

$$DOC = \frac{2\times\sum_{t_0}^{t_N} min\left\{\dot{Q}_{dem,coo}^{tot}(t_i),\dot{Q}_{dem,hea}^{tot}(t_i)\right\}}{\sum_{t_0}^{t_N}\left[\dot{Q}_{dem,coo}^{tot}(t_i)+\dot{Q}_{dem,hea}^{tot}(t_i)\right]} \tag{12}$$

Table 8
Characteristics of the four climate zones in the study.

Thermal zone	Representative city	Total heating demand [GWh]	Total cooling demand [GWh]	HCDR	District DOC
3 A	Atlanta, GA	3.83	12.65	−0.54	0.30
4C	Seattle, WA	4.53	4.38	0.02	0.54
5 A	Buffalo, NY	6.55	6.81	−0.02	0.38
6B	Great Falls, MT	6.87	5.40	0.12	0.37

Table 9
Optimal and average system COP for different temperature control strategies, system Configurations, and thermal zones.

Thermal zone	System configuration	Temperature control strategy	Optimal setting	Optimal system COP	Average system COP
3 A	II	Constant	16 °C	4.16	3.73
		Free-floating	20 °C	3.97	3.66
		Multi-stage	summer36°Cwinter16 °C	3.94	3.83
	III	Constant	20 °C	4.26	3.77
		Free-floating	20 °C	4.27	3.82
		Multi-stage	summer40°Cwinter16 °C	4.05	3.78
4C	II	Constant	16 °C	4.16	3.79
		Free-floating	32 °C	4.01	3.87
		Multi-stage	summer28°Cwinter16 °C	3.96	3.86
	III	Constant	20 °C	4.28	3.89
		Free-floating	20 °C	4.24	4.02
		Multi-stage	summer32 °Cwinter16 °C	4.09	3.93
5 A	II	Constant	16 °C	3.55	3.28
		Free-floating	32 °C	3.45	3.33
		Multi-stage	summer32 °Cwinter16 °C	3.43	3.35
	III	Constant	20 °C	3.61	3.33
		Free-floating	20 °C	3.60	3.42
		Multi-stage	summer32 °Cwinter16 °C	3.52	3.37
6B	II	Constant	16 °C	3.13	2.90
		Free-floating	20 °C	3.07	2.98
		Multi-stage	summer32 °Cwinter16 °C	2.99	2.95
	III	Constant	20 °C	3.16	2.93
		Free-floating	20 °C	3.18	3.05
		Multi-stage	summer32 °Cwinter16 °C	3.04	2.96

Within each climate zone, two network configurations were considered, resulting in a total of eight 5GDHC systems. For each system, 18 different temperature control strategy settings were implemented respectively. In this section, the system COP has been selected as the primary performance indicator.

Table 9 presents the best and average system COP for each temperature control strategy, considering different system configurations and climate zones. It can be noticed that for all the studied systems, the best scenarios involve a temperature control strategy that limits the cold pipe operating temperature to not exceed 16 °C. This emphasizes the energy-saving benefits linked to substation direct cooling. Therefore, when designing 5GDHC systems, it is advisable to equip substations with heat exchangers for cooling purposes and implement an operating temperature control strategy that facilitates direct cooling at the substations.

To facilitate further comparisons of different temperature control strategies across various climate zones, violin plots are employed to depict the distribution of the system COP, as illustrated in Fig. 16 and Fig. 17. Within these plots, the interior points located within the violins represent the data samples, while the width of the violin at any specific point indicates the density of data values at that location. Wider sections correspond to a higher data density, whereas narrower sections indicate a lower data density. It is worth noting that the temperature settings of 12 °C for both the constant temperature control strategy and the free-floating temperature control strategy have been excluded from the figures, as the settings of the multi-stage temperature control strategy studied in this paper do not allow temperatures below 12 °C.

From Fig. 16 and Fig. 17, it becomes evident that across all studied climate zones, the bidirectional parallel network consistently outperforms the unidirectional parallel network in terms of optimal energy performance. Among the four climate zones examined, climate zone 4C stands out, characterized by the most balanced heating and cooling demand both hourly and annually. In this climate zone, the bidirectional

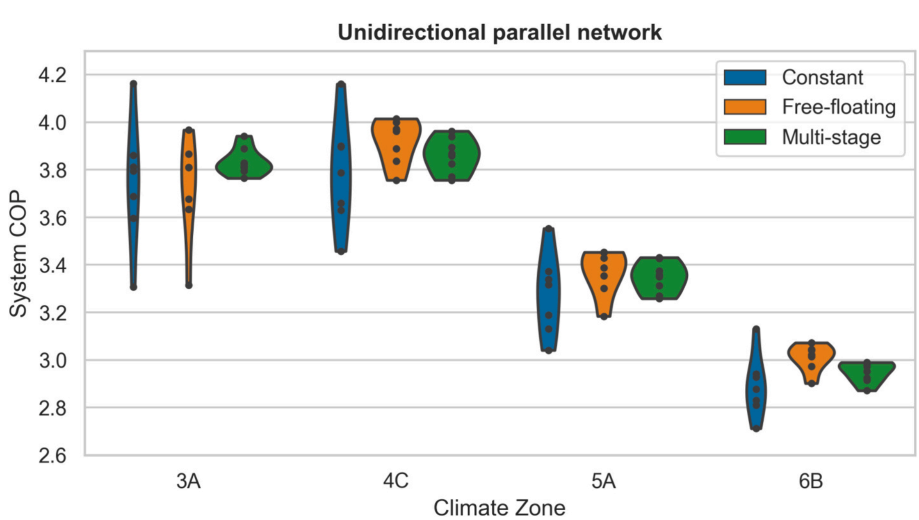


Fig. 16. The system COP of the unidirectional parallel system with various temperature control strategies in different climate zones.

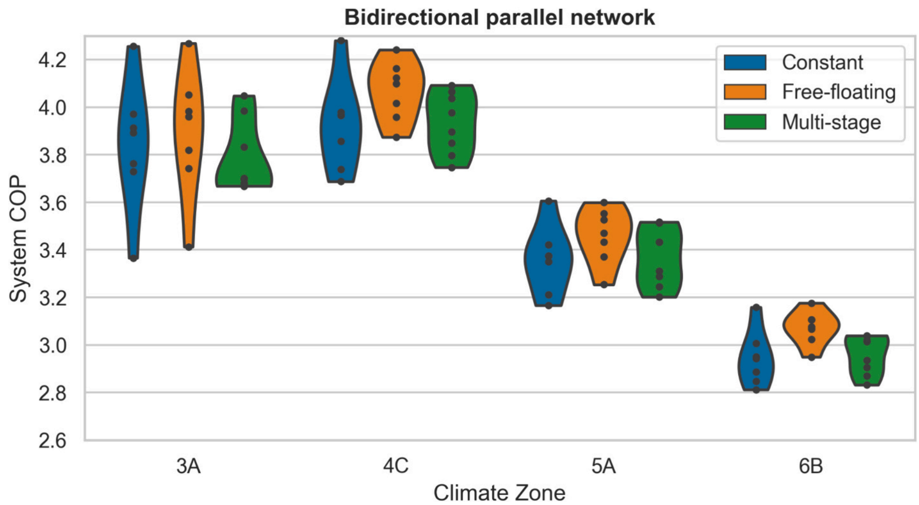


Fig. 17. The system COP of the bidirectional parallel system with various temperature control strategies in different climate zones.

parallel network demonstrates the most significant improvement, achieving an average energy performance increase of 2.80% compared to the unidirectional parallel network. In climate zone 3 A, which shares a similar system COP with climate zone 4C but exhibits the lowest DOC values, the average system COP of the bidirectional parallel network is only 1.32% better than that of the unidirectional network. This observation highlights the ability of bidirectional parallel networks to effectively harness the synergies arising from simultaneous heating and cooling demands, resulting in enhanced exergy efficiency.

As for the comparison among different temperature control strategies, it can be observed that the constant operating temperature strategy, despite lacking the consideration of load variations, does not necessarily result in poorer performance compared to other temperature control strategies. In fact, the best scenario frequently arises when the constant operating temperature strategy is employed. Out of the eight systems, six exhibit the highest system COP when the optimal constant operating temperature control strategy is applied. Even for the two systems that fall short of showing higher energy efficiency, the system COP with the optimal constant operating temperature control strategy is only slightly worse than the best case. This proves the idea that a well-designed constant temperature control strategy is able to achieve greater energy savings [32].

However, in comparison to the free-floating temperature control strategy, the system's energy performance with the constant operating temperature strategy appears to be more sensitive to temperature settings, evident in its slimmer violin shapes and lower average system COP. With the exception of the unidirectional parallel system in climate zone 3 A, all other systems exhibit better average system COP when using the free-floating temperature control strategy. Therefore, careful considerations are essential when determining operating temperature setpoints, particularly when employing the constant operating temperature strategy.

For the multi-stage temperature control strategy, the simulation results reveal that the scenario with higher operating temperatures in summer and lower operating temperatures generally leads to superior performance compared to the scenario with the opposite temperature settings, even though the system substations are equipped with heat exchangers. This highlights the importance of considering the overall system performance when selecting temperature control strategies for 5GDHC systems, rather than solely focusing on substation energy usage. In addition, neither of the two group settings shown in Table 6 presents a significant advantage over other temperature control strategies. In theory, a well-designed 5GDHC system employing a multi-stage operating temperature control strategy could surpass the best performance case of the constant operating temperature control strategy because the multi-stage operation involves multiple constant temperature levels in nature. In this study, it is evident that the full energy-saving potential of the multi-stage operating temperature control strategy has not been fully explored.

5. Conclusions and future work

This study aims to review and compare different temperature control strategies commonly applied to 5GDHC systems. Through an extensive literature review, three temperature control strategies were identified: the constant operating temperature control strategy, the multi-stage temperature control strategy, and free-floating temperature control strategy. Then, we developed a virtual testbed of detailed 5GDHC system models using Modelica. The virtual testbed was used to implement and assess the performance of multiple temperature control strategies and their various settings, both within the same climate zone (4C) and across different climate zones (i.e., 3A/4C/5A/6B). The following conclusions can be drawn:

- The performance of temperature control strategies can significantly vary based on load profiles, system configurations, temperature settings, and optimization objectives. Therefore, it is crucial to consider these factors and conduct a detailed annual system performance analysis when designing a suitable temperature control strategy for a 5GDHC system.
- While the constant operating temperature control strategy can yield better energy performance when appropriately configured, it is notably more sensitive to temperature settings compared to the free-floating temperature control strategy. Consequently, careful selection of settings is imperative when opting for the constant operating temperature control strategy.
- While simple seasonal setpoint control is frequently employed as a multi-stage temperature control strategy in 5GDHC systems, the simulation results in this study suggest that it may not achieve the desired system performance. In-depth analyses are required to enhance its energy efficiency.
- Substation direct cooling holds the potential to enhance overall system performance by substantially reducing the energy use consumed by substation cooling and eliminating the need for the installation of any mechanical cooling devices at substations, but it requires effective coordination with an appropriate temperature control strategy to maximize its energy-saving potential.

The following aspects will be addressed in future work:

- Real-World Data Integration: Due to the current limitations in data availability from real 5GDHC systems, this study utilized a synthetic 5GDHC system as the basis for the case study. Moving forward, we will strive to gather more real-world data from operational 5GDHC systems. Integrating this real-world data into our studies will enhance the reliability and applicability of our research findings.
- Temperature Control Optimization: The study revealed limited research on optimizing the temperature control of 5GDHC systems. Future work will delve into this area to further enhance the system performance through more rigorous optimization procedures instead of rule-based controls.
- Diverse Building Types and Energy Sources: This study uses the same building types (office/hospital/apartment) and thermal energy sources (air-source heat pump/dry cooler) for all the studied cases. Considering the broad applicability of 5GDHC systems, we will explore a broader range of building types and thermal energy sources in future investigations.
- Integration of Thermal Energy Storage (TES): This study assumes that no additional thermal energy storage systems, such as boreholes, are integrated into the 5GDHC system. However, incorporating TES can significantly enhance system performance [87]. Future studies should consider the integration of TES to evaluate its impact on the efficiency and effectiveness of 5GDHC systems.
- Hydraulic Control: In this study, we assume a ‘perfectly controlled’ hydraulic network for a precise flow delivery to each substation. However, achieving such precision in practice is challenging. Future research will discuss the impact of different hydraulic control strategies on 5GDHC systems.
- Scenario Analysis for Urban Climate and Extreme Weather Events: In this study, we have explored the performance of 5GDHC systems across various climate zones. However, the impact of local urban climates and extreme weather events has not been addressed. The proposed virtual testbed could be utilized in combination with urban climate models and Urban Building Energy Modeling (UBEM) models to further investigate the performance and resilience of 5GDHC systems under these conditions. This approach would provide a more comprehensive understanding of how the system performs in diverse urban environments and extreme weather scenarios.

CRediT authorship contribution statement

Yuhang Zhang: Writing – original draft, Methodology, Investigation, Formal analysis, Conceptualization. **Mingzhe Liu:** Writing – review & editing, Investigation, Formal analysis, Conceptualization. **Zheng O’Neill:** Writing – review & editing, Supervision, Resources, Project administration, Funding acquisition, Conceptualization. **Jin Wen:** Writing – review & editing, Investigation, Formal analysis.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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Appendix A. System design parameters

This appendix provides design parameter values used in the 5GDHC network models. These parameters were established based on the combination of literature review results and engineering experience.

Table A1
Design parameters of substations.

Parameter	Value
Design temperature differential	4 °C
Building hot water supply temperature setpoint	18 °C
Building chilled water supply temperature setpoint	55 °C
Heat pump/chiller Carnot effectiveness	0.4
Heat exchanger effectiveness	0.8
Heat exchanger approach temperature	1.5 °C
Nominal pressure drop over the water side	20,000 Pa

Table A2
Design parameters of the energy hub.

Parameter	Value
Design temperature differential over the water side	4 °C
Design temperature differential over the air side	10 °C
Approach temperature of the dry cooler	5 °C
Heat pump/chiller Carnot effectiveness	0.4
Nominal pressure drop over the water side	20,000 Pa
Nominal pressure drop over the air side	300 Pa

Table A3
Design parameters of pipelines.

Parameter	Value
Length of each pipe	300 m
Buried depth	1 m
Nominal pressure drop per length	100 Pa/m
Horizontal distance between pipe walls	0.2 m
Thermal conductivity	0.17 W/m•K

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