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THERMODYNAMIC MODELING OF A FOURTH GENERATION DISTRICT HEATING SYSTEM IN BRAZIL

ALI MOALLEMI

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Ali Moallemi

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Supervisor: Prof. Dr. Fabrício José Pacheco Pujatti

Co-Supervisor: Prof. Dr. Ramón Molina Valle

Federal University of Minas Gerais

Belo Horizonte

Engineering School of UFMG

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UNIVERSIDADE FEDERAL DE MINAS GERAIS
PROGRAMA DE PÓS-GRADUAÇÃO EM
ENGENHARIA MECÂNICA
Av. Antônio Carlos, 6627 - Campus Universitário
31270-901 – Belo Horizonte – MG
Tel.:+55 31 3409.5145
E-mail: cpgmec@demec.ufmg.br

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ALI MOALLEMI

Tese submetida à Banca Examinadora designada pelo Colegiado do Programa de Pós-Graduação em Engenharia Mecânica da Universidade Federal de Minas Gerais, como parte dos requisitos necessários à obtenção do título de **"Doutor em Engenharia Mecânica"**, na área de concentração de **"Energia e Sustentabilidade"**.

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Por:

Prof. Fabricio Jose Pacheco Pujatti
Orientador - Departamento de Engenharia Mecânica/ UFMG

Prof. Ramon Molina Valle
Co-orientador - Departamento de Engenharia Mecânica/ UFMG

A. KOOhsar
Prof. Ahmad Arabkoohsar
Technische Universiteit Eindhoven

Wadaed Uturbey
Prof.ª Wadaed Uturbey da Costa
Departamento de Engenharia Elétrica/ UFMG

Prof. Jose Guilherme Coelho Baeta
Departamento de Engenharia Mecânica/ UFMG

Prof.ª Tatiana Paula Alves
Centro Universitário UNA

Gratefully dedicated to my beloved Wife, Naier, who changed my world. For her love and endless efforts in this challenging path.

And my dear late father, may he rest in peace…

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ABSTRACT

Heating and cooling is accounted for about half of the global energy use where domestic heating is the main part of this energy consumption. Due to the importance of heating in the global energy crisis, district heating systems are on the verge of a transition to their fourth generation from in-operation third generation. In a fourth generation district heating system, the penetration of renewables is supposed to be much higher (up to 100%), there is strong synergy between the energy sectors, the rate of losses are considerably lower and the whole system is remarkably more energy efficient. In spite of the extensive advancement made in district heating systems, this technology has not been practically introduced to the Brazilian energy system yet. The main reasons for this are the low demand of space heating in the country due to the moderate to hot climate, and the highly renewable-based electricity sector of the country that it has made the use of electricity for heating application rational. In this project a district heating system, based on the latest standards of the fourth generation will be designed, sized and modeled for a residential region in the south of Brazil. The main objective of the project is to evaluate the technical feasibility of the implementation of such advanced district heating systems in the future energy matrix of Brazil. For this, various possible district heating technologies that are compatible with the 4GDH concept will be investigated and an innovative solution will be proposed. The performance of these systems will be comprehensively compared in terms of heat losses, the pipeline sizing, the pump work, and etc. Therefore, a through technical comparision of the existing generation of district heating, so-called the 3rd generation of district heating, ultra-low temperature system and proposed non-uniform temperature system will be done. The results show that the energy system of the case study area can effectively host a district heating system. The considered cases include the third generation, the low-temperature and an innovative proposed district heating systems. These are the best solutions that can be practically implemented by today. Among these cases, the proposed system so-called non-uniform temperature district heating system can effectively outperform the other solutions in terms of an overall energy-economy point of view. In addition, the project assesses the compatibility of the twin pipes, being used in the currently in-operation district heating systems, with the proposed district heating scheme thermo-hydraulically.

Keywords: District Heating Systems, the 4th Generation of District Heating, Thermodynamic Modeling, Thermo-hydraulic Performance, Twin-pipes.

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ABBREVIATIONS AND NOMENCLATURES

Abbreviations

3GDH	3rd Generation of District Heating
4GDH	4th Generation of District Heating
CHP	Combined heat and power
COP	Coefficient of Performance
CV	Control Volume
DCW	District Cold Water
DH	District Heating
DHSU	District Heating Storage Unit
DHW	Domestic Hot Water
Eq	Equation
ESS	Energy Storage System
EU	European Union
FC	Flow Controller
HE	Heat Exchanger
HPFU	Heat Pump Furnished Unit
IEA	International Energy Agency
IEHU	Individual Electrical Heat Unit
IHEU	Instantaneous Heat Exchange Unit
LTDH	Low Temperature District Heating
NUTDH	Non-Uniform Temperature District Heating
P	Pump
PCM	Phase Change Material
PUR	Polyurethane
RES	Renewable Energy Sources / Systems
SHW	Space Heating Water
ST	Storage Tank
SH	Space Heating
TES	Thermal Energy Storage
THS	Thermal Heat Storages
TSV	Thermostatic Valve
ULTDH	Ultra-Low Temperature District Heating
VTDH	Variable Temperature District Heating
WtE	Waste-to-Energy

Nomenclatures

a	Air	
b	Building	
b_m	Building Stuck	
A	Heat transfer area	(m^2)
C	Specific thermal capacity	(kJ/kg.^{o}C)
c_p	Specific thermal capacity in constant pressure	(kJ/kg.^{o}C)
D, d	Diameter	(m)
D_h	Hydraulic diameter	(m)
E	Overall energy of control volume	(kJ)
$\overline{E}$	Heat exchanger effectiveness	

f	Darcy Friction Factor	
g	Gravitational acceleration	(m/s^2)
G	Production of Turbulence Energy	(J)
H	Specific enthalpy	(kJ/kg)
$\bar{h}$	Convective heat transfer coefficient	$(W/m^2.K)$
h	Enthalpy	(kJ)
j	Node Number	
k	Thermal conductivity	$(W/m.^oC)$
KE	Kinetic Energy	(J)
M	Mass	(kg)
$\dot{m}$	Total air mass flow rate	(kg/s)
N	Number	
NU	Nusselt Number	
P	Pressure	(kPa)
PE	Potential Energy	(J)
Pr	Prandtl number	
Q	Heat	
$\dot{Q}$	Heat transfer rate	(kW)
R	Radius	(m)
Re	Reynolds number	
S	Specific entropy	(kJ/kg.K)
S	Entropy	(kJ/K)
S	Strain Tensor	
S_h	Heat Within the Solid	(J)
T	Time	(s)
T	Temperature	(^{o}C or K)
U	Internal Energy	(J)
UA	Overall heat transfer coefficient	$(W/m^2.C)$
V	Volume / Velocity	m^3
$\dot{V}$	Volume flow rate	(m^3/s)
W	Specific work	(kJ/kg)
W	Work	(kJ)
$\dot{W}$	Work rate	(kJ/s)
X	Exergy	(kW)
Z	Elevation	(m)

Greek Symbols

β	Coefficient of heat pump performance / of thermal expansion
Δ	Change in parameter
Ε	Dissipation of turbulence energy
Η	Efficiency
λ	Time step
μ	Dynamic viscosity
ν	Specific volume
ρ	Water density
σ	Turbulent Prandtl number
τα	Transmission-Absorption coefficient

ψ	Stream exergy
υ	Kinematic viscosity
φ	Exergy
ω	Angular velocity
Ω	Rotation

Sub(super)scriptions

0	Dead state
1	Initial state
2	Final state
a	Air
act	Actual
amb	Ambient
avg	Average
b	Buoyancy
CO_2e	CO_2-equivalent
cold	Cold fluid stream
comp	Compressor
cond	Condenser
cv	Control Volume
dh	District heating
dhw	District hot water
dest	Destroyed
e	External
ex	Heat exchanger
eff	Effective
f	Working fluid
fu	Fuel
gen	Generated
h	Heat, Heater
hl	Heat loss
hp	Heat pump
I	Internal
ins	Insulation
In	Entering parameter
k	Kinetic
lam	Laminar
lm	Logarithmic mean
mix	Mixing
max	Maximum
n	Node
out	Exiting parameter
p	Pipe
r	Return
rev	Reversible
s	Supply
sp&hw	Space heating and hot water
st	Storage tank

std	Stored
sys	System
th	Thermal
tur	Turbulent
ven	Ventilation
w	Wall

1 Introduction

In the today's modern world, the energy crisis and discharging fossil fuels together with the global warming and pollution problems are of the most critical issues. These challenges are all caused by the broad use of fossil fuels. The use of Renewable Energy Sources (RES) and technologies, instead of traditional fossil fuel based systems, to a large extent, is the best solutions for addressing the aforementioned challenges. Renewable energy sources are playing a progressively substantial role in the energy sector in the world. Many countries and international organizations have made decisions and are planning and setting goals to facilitate a renewable-based future energy system. The share of renewable energy in gross final energy consumption in EU countries, for example, reached 14.1% in 2012 and it should be increased to 20% by 2020 [1]. The future energy system in which there is a high share of renewables is so-called smart energy system.

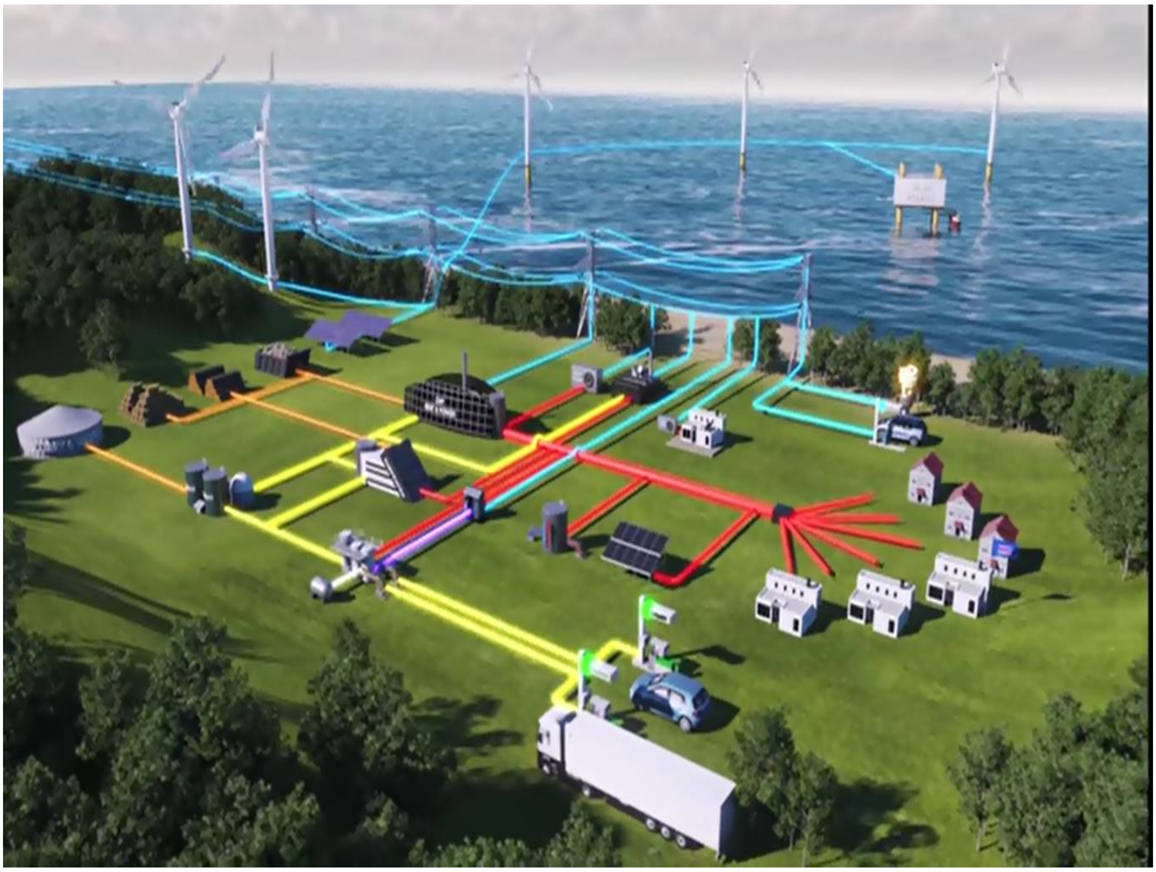

Figure 1-1 Schematic of a smart energy system: Red heat pipes, Yellow fuel pipes, Blue electricity lines

Introduction

The term smart energy system, as can be seen in Figure 1-1, was proposed as an integrated future system that provides a sustainable energy system for the society [2]. This promising system would be integrally including Smart Thermal Grid, Smart Electricity Grid, Transportation System, and Gas Grid of the cities and towns. To allow this, all of these smart systems should work together to coordinate the energy between different heat and electricity sectors, as well as different fuel infrastructures as an integrated system to achieve as high efficiencies as possible. One of the main objectives of this developing system is the substitution of fossil fuels by renewable energies for decreasing energy crisis, and pollutant emissions problems to find the best options.

According to the International Energy Agency (IEA) report, heating and cooling was taken into account for about 46% of the total global energy use in 2012 and 50% of energy consumption in Europe are used for heating [3]. A district heating system (or district heating network) is a system to distribute generated heat from one or various heat sources for residential, commercial, and industrial building for both space heating and water heating purposes [4]. District heating infrastructures also have a big effect on energy efficiency and the possibility of using heat resources to meet demands in the near future. District heating includes a network of pipes connecting the buildings in a neighborhood, town center or the whole city, so that they can be provided by a centralized plants or distributed heat-producing units. This allows any available source of heat to be used, making the supplied heat as cheap and efficient as possible [5]. The fundamental idea of district heating is to use local fuel or heat resources that would otherwise be wasted, in order to satisfy local customer demands for heating, by using a heat distribution network of pipes as a local marketplace. A district heating system supports this idea by using local resources with respect to the market. Also, it can be supposed as a substitution of ordinary primary energy supply for different heat demands in society to achieve lower environmental impacts. In addition to energy efficiency, and environmental impacts, district heating systems will improve safety and reduce fuel transportation problems since there is not any combustion system for heating at the end-user location.

In countries with strong driving motivations and forces, district heating systems supply heat to even more than half of the buildings. For example, Denmark supplies heat for more than 60% of the country’s land via district heating networks[6]. Or as another example, in Finland, the share of space heating increased from 21% to 25% during 2005–2012, out of which almost 50% of the total heating market was accounted for district heating [7]. On the other hand, district heating systems may be seen rarely in other countries because of the low knowledge or competitiveness of district heating. Establishment and experiences of district heating systems have paved the way for introduction and implementation of district cooling systems, mostly for space cooling in buildings. However, this development is younger compared to the heating systems and are, therefore, neither as common nor as extensive as district heating systems [8].

Today, most of district heating systems work with a supply temperature completely above 70 °C and a return temperature of above 40 °C [9]. Consequently, heat losses occur during the heat transfer as the effect of the temperature difference with the surroundings. The exergetic losses also happen especially at the location of the heating plant because of the temperature difference between the combustion temperature and the temperature of the heat transfer fluid. Moreover, there are additional heat losses in the user side to reduce the domestic hot water temperature to a maximum degree of 60 °C in a usable level. The current district heating systems are fossil-based for heat supply and as mentioned above, they work in high temperatures with high heat and exergy losses. For this reason, a generation change from the current nuclear and fossil fuel-based energy system to a more efficient and renewable energy-based generation is required to meet the future energy demands. Also, future district heating infrastructures should not only be planned for the current energy system and energy conservation measures in the existing buildings but also for the future system and low-energy buildings.

The term of the 4th Generation of District Heating (4GDH) was first employed by Prof. Sven Werner as a part of the work of the Strategic Research Centre for 4th Generation District Heating Technologies and Systems in Denmark [10]. This research center emphasizes on the important role of district heating and cooling systems in future sustainable energy systems, however, the technologies should be advanced to accomplish such roles to develop into a new 4th generation. In the previous years, the Heat Plan Denmark studies showed the possibility of this new system in Denmark and similar Heat Roadmap Europe studies were subsequently formed for the whole of Europe [11]. The 4th generation approach points out those technologies should be developed and adjusted to the future conditions of renewable energy-based supply. This new generation of heating system will contribute to the heating section of cities and urban areas to heat the industrial and residential buildings as well as houses. Accordingly, the study of such systems is valuable to improve the heating part and overall energy system of the civilization in both terms of economy and efficiency. Consequently, a 4GDH may connect to other parts of the smart energy system to improve the life quality of the human societies. Therefore, the development of 4^{th} generation district heating is essential for the implementation of smart energy systems since both 4GDH and district cooling system can be supposed as an important part of the future smart energy systems [12]. Therefore, the original motivation of a 4GDH is a final integrated sustainable energy system to supply 100 % renewable non-fossil heating as a part of the future smart energy system.

On the other hand, although Brazil is a country with very high rate of renewable energy use and lies among the top countries of the world in this regard, does not take advantage of district energy systems. This might include both of district heating and cooling systems as most of the land has a hot climate and a part of that has cold climate.

Brazil has almost no district heating system but there is a large domestic market, with total demand of more than 575 TWh. During the recent years, low prices of the goods, poor credit rating, and political crises have led to the downturn and shrinking power consumption. The economic crisis has entailed increased private involvement in the transmission and gas markets. Despite this short-term trend, it is reasonable to expect fundamental long-term growth. Nowadays, in Brazil, the system is dominated by hydropower that is vulnerable to variations in water inflow. However, significant new capacity is added to the energy system, 9 GW has come online recently, mainly wind (2.9 GW) and hydro (5.4 GW) [78].

Brazil possesses different RES like solar, wind and geothermal energy capabilities and the possibility of large biomass CHP. Consequently, there is a good potential to develop more energy efficient systems increasingly such as 4GDH. On the other hand, Brazil has a lot of dense cities. These large and small cities are scattered in all of this large country since almost everywhere of this country is suitable for cultivate and living. Many of these cities have cold winters and it enables using district heating systems in these cities. As mentioned, 4GDH is a reasonable system for dense urban areas with access to different RES and heat sources. Therefore, this new generation district heating system can be very extremely beneficial for Brazil. However, it requires an exact and detailed management for planning, regulations, and financial and non-financial supports by Federal and Provincial Governments to hire these large potential capacities of this natural-rich country.

According to these facts, this thesis studies the technical feasibility of developing a district heating system for a case study in Brazil. For this, the performance of various district heating designs to meet 4GDH requirements, including those already proposed by others, plus a specific concept of non-uniform temperature district heating (NUTDH) system, designed by ourselves, and it is simulated and analyzed thermodynamically in the case study. The results are presented and discussed in details and finally, the conclusion of the study is presented. The aim of the simulation model is to cover the heat demand of the case study.

The main purpose of this research as the societal objective is to develop the concept of the 4th Generation District Heating in order to provide the understanding of the important role of this future infrastructure. However, a further perspective is required to facilitate the development of additional national and international research projects as well as the link to the industrial area and universities.

The concept of district heating and specifically, the 4GDH is not a well-known approach in Brazil. For this reason, a small cold city was chosen to evaluate the implementation of the latest generation of district heating systems as a pilot study in Brazil.

Also, the compatibility of twin pipes with the proposed system is studied since this technology has been widely used for currently in-operation district heating systems. This compatibility is a very important factor for feasibility of the system because it can show the ability of the system to be run with the updated technologies.

This thesis text is organized in five chapters as follows. An introduction to the work was introduced as the first chapter. In chapter 2, the 4GDH as a new generation of heating systems and the history of the prior generations will be explained. Also, a review literature of the preceding studies will be described in this chapter. In chapter 3, the methodology of the research will be described. Obtained results from the modeling and simulations will be presented in chapter 4 and finally, the conclusion will be discussed in the last chapter.

Introduction

It should be mentioned that the main results of this thesis were published as a paper by title of “Non-uniform temperature district heating system with decentralized heat storage units, a reliable solution for heat supply” in Energy Journal on January 2019.

2 Literature Review

As the main goal of this work is modeling and simulation of the performance of a district heating system, the fundamental knowledge of this technology should be first presented. This chapter presents a detailed background and information of district heating systems up today.

2.1 Previous Generations

The 1st generation: In the first generation of district heating systems, introduced in the USA around 140 years back, steam was the main heat carrier through the pipes. The main reason for introducing this technology at the time was reducing the risk of fire or explosion of the individual boilers in the buildings and increasing comfort. Steam pipes through concrete ducts, steam traps, and compensators were the key components of this high temperature district heating system, and surprisingly there was no substation heat exchanger for the buildings. The steam pipe material was normally steel and had insulations around. For providing the hot water required for the consumers, hot water tanks heated directly with steam or from a secondary water circuit were used and high-temperature radiators (around 90 °C) using steam or water were the medium heat distribution systems in the buildings. Coal steam boilers and some CHP plants were the main heat supply units in this system [4].

The principal drawback in this technology was the very high rate of losses through the transmission/distribution system making it outdated. The risk of pipeline explosion and damage to facilities and people as well as corroded return pipes were further important failures of this system. The first generation of district heating was only the dominant in-operation system worldwide for less than 50 years. Very few locations, like part of USA and France, may be found with this technology today [8].

The 2nd generation: This system, in which pressurized hot water is the heat carrier, was introduced in the 1930s and this technology also continued to be the rampant district heating system worldwide for almost the following half-century. The supply temperature was usually above 100 °C and the transmission was carried out through the insulated steel pipes in concrete ducts. Tube-and-shell heat exchangers were used for the substations and central pumps were the circulator systems. Domestic Hot Water (DHW) was supplied by shell and tube heat exchangers for a larger group of consumers while in case of supplying on building level, the DHW storage tank principle was applied [6]. Just like the 1st generation, high-temperature radiators were the main components of the system for space heat within the buildings. Coal and oil based CHP units and heat-only boilers were the main heat production systems [9].

Although a big portion of the supplied heat to the transmission pipeline was still lost in this system, the rate of losses was much less than of the 1st generation. The other motivation for transition from the first to the second generation of district heating system was to achieve fuel savings and better comfort by utilizing CHP [4].

<u>The current in-operation 3rd generation:</u> The two main characteristics of the 3rd generation district heating system, which emerged in the 1980s and is in its latest lifetime period, are the material/components applied and the lower operating temperatures than the previous two systems. The supply temperature is below 100 °C where pressurized water is used as the heat carrier. The key components are the pre-insulated pipes buried directly into the ground, the prefabricated compact substations, the compact stainless steel plate heat exchangers and the material of other components such as valves etc. The circulation is carried out by central pumps, and direct/indirect supply of district heating water through medium-temperature radiators (around 70 °C) or under-floor heating are the main space heating methods. DHW is supplied either directly by plate heat exchangers (instantaneous heat exchange units) or by employing both storage tanks and plate heat exchangers (district heating storage unit). A wide range of technologies including large-scale or distributed CHPs, biomass and waste incineration plants, biomass or fossil fuel boilers, heat pumps etc. comprise the heat supply chain in this system. The main advantages of the 3rd generation compared to the 2nd generation are lower temperature levels leading to lower losses and higher production efficiencies, utilizing renewable energy sources, using insulation material with much better performance (making the losses even less), using pre-insulated pipes with higher lifetime and lower installation cost/time instead of in-situ insulation pipes, more advanced components with higher efficiencies and qualities and keeping the system in the optimal operation bound. Besides, pipe leakage detection systems and using energy meters are further progresses made in the district heating system in this generation. [10]

The primary motivations for this generation were the increased focus on lower cost and higher energy efficiency caused by the oil crises leading to higher fuel price at the time. That is why a great expansion in the use of CHP systems and employing biomass and waste incineration plants is observed in this generation. [45]

Comparison of the first three generations of district heating systems: Figure 2-1 shows a schematic of the first three different generations of district heating systems and the involved technologies in the supply chain based on the presented information. As can be seen, the heat supply systems in the first generation (left figure) were only coal or waste fired heat production plants where there were huge steam storage tanks for supposing the network during peak demand times. In the middle figure, it is observed that in addition to the storage units and the coal/waste fired plants, CHP systems have been widely used. The schematic of a 3rd generation district heating system can be seen on the right side of the figure in which biomass sources and surplus industrial heat are among the main heat supply ways and the heat carrier is medium temperature pressurized water. Besides, the graph presented at the top of the figure compares the three generations of district heating in terms of temperature and thermal efficiency. As discussed and also the graph approves, the temperature of supply is lower for the later generations (red curve). Expectedly, the losses decrease significantly with lower temperatures and more efficient materials. Therefore, the efficiency is the highest for the 3rd generation and the lowest for the 1st generation (blue curve).

Table 2-1 summarizes the information given in the previous section in order to make a comparison of the three generations of district heating systems in substantial details and characteristics.

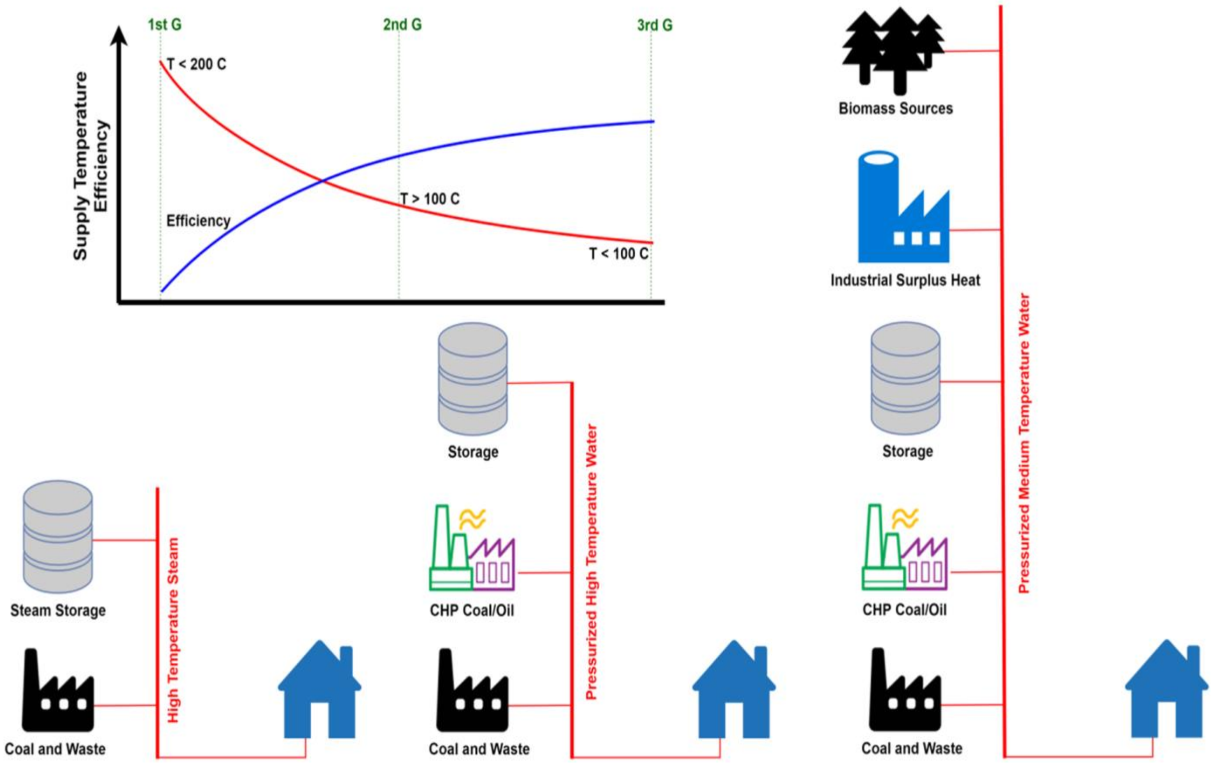

Figure 2-1 Schematic of the three generations of the district heating systems and the involved technologies; left: 1st generation, middle: 2nd generation, right: 3rd generation, graph: comparison of efficiency and temperature in the three district heating systems.

Table 2-1 The comparison of the three generations of DH systems in terms of the main characteristics.

	1st Generation	2nd Generation	3rd Generation
Heat Suppliers	Coal steam boilers and some CHP plants	Coal and oil based CHP and some heat-only boilers	Large-scale CHP, distributed CHP, biomass, waste and fossil fuel boilers
Heat carrier	Pressurized steam	High temperature pressurized water	Pressurized medium temperature water
Temperature of supply	200-300 ˚C	100-200 ˚C	70-100 ˚C
Pipe shape	In-situ single pipes	In-situ single pipes	Pre-insulated twin pipes
Pipe material	Steel	Steel	Copper-steel-poly ethylene
Insulation material	Wool	Wool	PUR Foam
Pipe casing	Concrete ducts	Concrete ducts	Plastic
Substation heat exchanger	No substation heat exchangers	Shell and tube heat exchanger	Razed stainless steel plate heat exchangers
Space heating system	High temperature radiators	High temperature radiators	Medium temp. radiators or under floor heating
DHW system	Hot water tanks heated directly with steam or from a secondary water circuit	Shell and tube heat exchangers for a larger group of consumers and DHW storage tank for building level	Instantaneous Heat Exchange Unit (IHEU) or DH Storage Unit (DHSU)

2.2 The new generation of district heating

In order to be able to fulfill its role in future sustainable energy systems, district heating will have to transit to its 4th generation. The 4GDH will be indeed a main part of future energy systems comprising smart heat and electricity grids. The main challenge of smart thermal grids is the utilization of low-temperature heat sources and the interaction with low-energy buildings while the main challenge of smart electricity grids is the integration of fluctuating and intermittent renewable electricity production. The major objective of both of them is a large-scale integration of an increasing level of intermittent renewable energy in an efficient way and these two concepts complement each other as necessary technologies to establish the sustainable energy systems. Anyway, there are some principal challenges for a successful transition in the smart thermal grids from the existing district heating systems to the 4GDH systems that they should be addressed [45]:

a) Low Temperature District Heating (LTDH) for both space heating and DHW for the existing and the new low-energy buildings,

b) low grid losses,

c) recycling heat from low grade sources and integrating renewable heat sources,

d) integration to other energy sectors including electricity, gas, and cooling grids,

e) optimal planning as well as cost and motivation structures

Figure 2-2 illustrates a sketch diagram of the 4GDH system including the involved technologies in the production of the heat demand and those making synergy with the DH network. As seen on the top of the figure, the possible supply and return temperature ranges have been mentioned as 35-55 $^{\circ}$C and 20-25 $^{\circ}$C, respectively. It has also been pointed out that the losses are lower and thermal efficiency is higher compared to all of the previous generations of district heating. According to the Figure 2-2, there are a large number of highly efficient facilities (including biomass CHP, large scale heat pumps, waste incineration plants, solar and geothermal plants and future energy production technologies) contributing in the heat supply chain and various sorts of synergies between the district heating and other thermal sources, e.g. seasonal and regular heat storage units, surplus electricity of wind farms, 2-way heat exchange with industrial buildings, utilization of any source of surplus heat and any possible sustainable synergy.

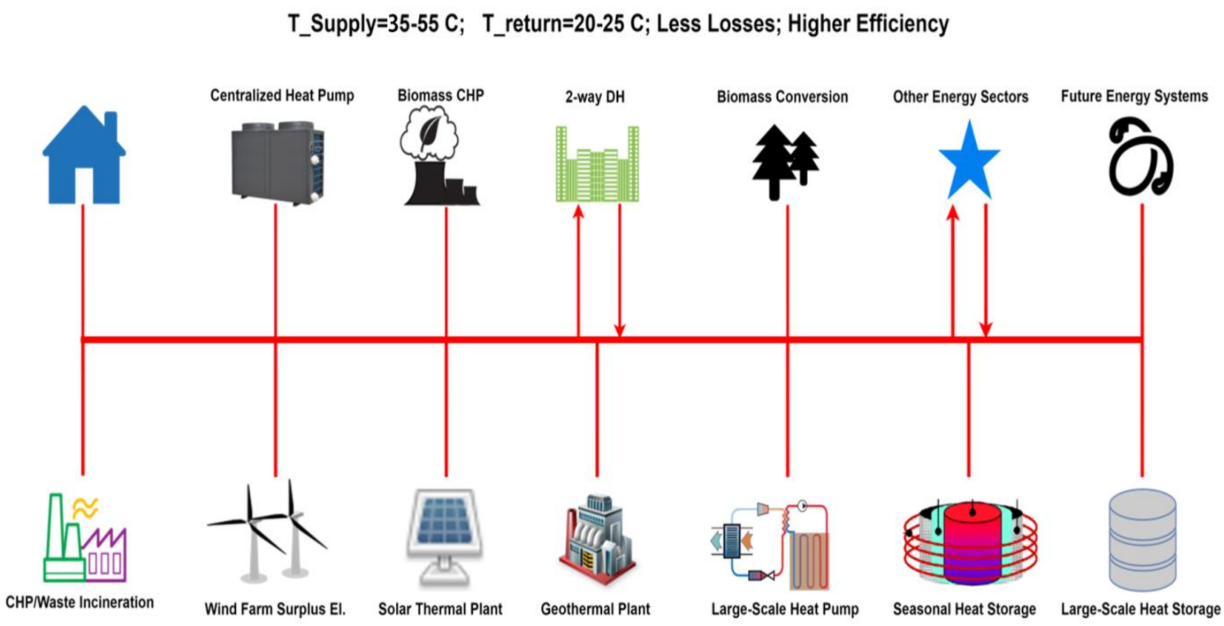

Figure 2-2 Schematic diagram of a 4GDH.

Following, the challenges and the relevant aspects of the 4GDH are elaborated in a more detailed way.

<u>LTDH for both space heating and DHW:</u> As discussed, in the concept of the 4GDH, efficient energy performance of buildings is a crucial parameter. Therefore, development of sustainable buildings plays an important role in the transition of district heating system generation. Making a balance between the summer and winter heat demand of the building is a design feature of the future buildings. This, which can be achieved by reducing the energy use for space heating to the level of energy required for DHW supply, can result to a constant low energy use level at a lower cost. The demand temperature may further decrease by more efficient heating systems capable of using supply temperatures around 40 °C and return temperatures near room level. Low temperature floor/wall heating and oversized water heating panels (radiators) with proper flow control systems are also further possibilities. In addition, the smaller volume of water in the pipes and the taps, the lower risk of legionella problem and the lower rate of heat losses. Intelligent control, employing weather forecast data and peak shaving will improve the energy efficiency of the district heating system [4].

It should be mentioned that legionellosis is a collective term for diseases caused by legionella bacteria including the most serious Legionnaires' disease, as well as the similar but less serious conditions of Pontiac fever and Lochgoilhead fever. Legionnaires' disease is a potentially fatal form of pneumonia and everyone is susceptible to infection. This bacterium is common in natural water sources such as rivers, lakes and reservoirs, but usually in low numbers. They may also be found in purpose-built water systems such as pipelines, cooling towers, evaporative condensers, hot and cold water systems, and heat storages [80].

Low grid losses: The focus of smart thermal grids is on lower cost and lower heat losses by improving the components and decreasing the buildings' heat demand. This paves the way for better utilization of low-grade renewable heat, increases the thermal efficiency of CHP and large-scale heat pumps and for using heat storage systems. The possible solutions to lower the rate of heat losses in the grid are lower temperature network (annual average of 40-45 °C in supply and 20 °C in return, for instance), smaller pipe dimensions, better insulation materials and a loop layout of supply and return pipes (in order to establish circulation of supply pipe during summer) are [45].

Low-grade source utilization: As mentioned, better utilization of available low-grade heat sources especially renewable heat sources and increasing the thermal efficiency of CHP and large-scale heat pumps together with the use of thermal storage is a key feature of the future smart energy system. The heat from CHP and waste incineration units, waste/surplus heat from industrial processes and commercial buildings, the heat supplied by geothermal plants or solar thermal plants with seasonal storage are the main possible ways of low-grade source utilization.

Waste incineration is very useful for district heating systems, especially for base load coverage, provided that it is used in an optimal way. In order to make use of all the heat production from waste incineration, it is necessary to have a sufficiently large district heating network coupled to the incineration plant. With a LTDH network, there is much higher potential for usable waste heat from industrial processes. In most areas of Europe, usable hot water is available in the ground. Considering the temperature of the earth's layers, an LTDH makes it much easier to construct geothermal plants for use in district heating systems in direct or indirect, e.g. geothermal heat pumps. Large solar thermal plants may be placed in the cities nearby and supply the district heating distribution lines. However, due to the seasonal mismatch of solar availability and the heat demand of the grid, a seasonal storage is more advantageous for solar thermal fields in the district heating systems. In addition, lower supply and return temperatures in the district heating systems will also result to higher power-to-heat efficiencies in steam CHP plants, higher heat recovery from flue gas condensation, higher coefficients of performance in heat pumps, higher conversion efficiencies in solar collector fields, and higher capacities in thermal energy storages [45].

<u>Integration to other energy sectors:</u> The most important sources of renewable energies, e.g. solar and wind, are intermittent and fluctuating. This is the main challenge of integrating these sources into the existing energy systems. Meeting this challenge is even more vital for electricity as there should always be Balance between generation and demand of electricity must be verified at each instant, and occurs in real time. Employing large-scale heat pumps, electrical boilers or Energy Storage Systems (ESS) in the district heating system is a smart measure that may facilitate balancing the demand and production of the renewable-based electricity grid. However, the district heating system needs to be redesigned to be appropriate for this integration and meet the required standards. It can also be so beneficial to make such a properly designed synergy between the district heating system and other energy sectors, e.g. gas sector. The practical activities for making this reliable interaction between the sectors can be achieved by, for instance, regulation of CHP plants by use of thermal heat storage, use of large-scale heat pumps in CHP systems, involving CHP, heat pumps, and the electrification of transport (batteries and electrolyzers), in securing grid stabilization tasks, etc [45].

<u>Optimal planning and cost structure:</u> This item is not that technical, rather more management. This actually declares that technological change from the current energy systems (i.e. the nuclear and fossil fuel-based energy systems) to the future energy systems (the renewable-based smart energy system), specifically transition from the 3rd generation of district heating to the 4GDH, has various management aspects to consider as well. This transition must be accomplished in a way that suitable planning, cost and motivation structures in both supply and consumer chains is ensured in the 4GDH [4].

2.2.1 Heat supply technologies

In a 4GDH, there are a number of key technologies that are involved in the heat production and supply process. Figure 2-2 illustrates a schematic of the involved technologies. The fundamentals of these technologies are discussed hereunder. Note that as neither analysis nor optimization of the heat supply chain of the 4GDH is in the scope of this project, no thermodynamic models are presented in this section.

CHP: A large portion of the input energy for generating electricity is wasted, mainly as heat, in the conventional power generation technologies. The idea of capturing this wasted heat for other uses than electricity production, like steam or hot water production for thermal uses, resulted to the concept of CHP. In this way, a CHP unit may produce energy at high overall thermal efficiency of up to 90%. An optimal CHP system is designed to meet the heat demand of the energy user, whether at building, industry or city-wide levels, since it costs less to transport surplus electricity than surplus heat from a CHP plant. For this reason, CHP can be viewed primarily as a source of heat, with electricity as a by-product, for a 4GDH system heat production chain.

A CHP system may come into various configurations but the principle is always the same and based on an efficient system that combines electricity production and a heat recovery system. CHP plants consist of four main components, namely, a prime mover, an electricity generator, a heat recovery system and a control system and may be classified by the type of application, prime mover and fuel used. Almost any fuel can be used in a CHP system, including fossil-fuels, municipal solid waste, biomass, etc. Even multi-fuel CHP systems are possible. Waste incineration CHP plants and biomass CHP plants will be the crucial components of the 4GDH.

Figure 2-3 compares a sample CHP system performance and individual heat and power generation systems. This can clearly show how a CHP may result in much higher efficiencies and savings. In addition, one should add the savings achievable in the transmission system as CHP plants are usually built near the consumption locations. Thorough information about various CHP types may be found in [13].

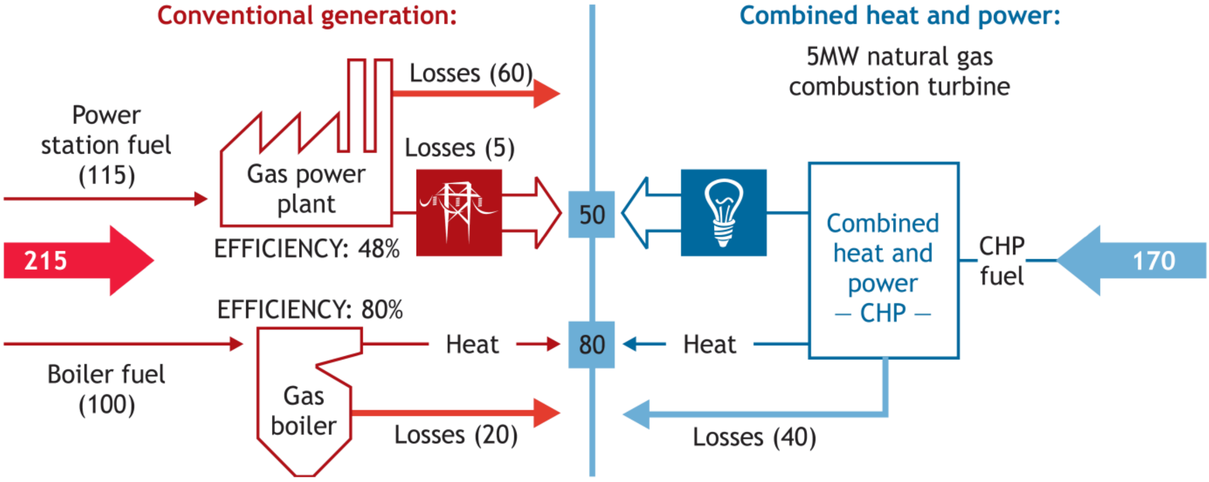

Figure 2-3 Comparison of individual heat and power generation methods with combined production.

Waste incineration: One of the interest solutions in the future energy systems is increasing the share of waste-to-energy technologies. Among waste-to-energy technologies, incineration is more popular than others, e.g. gasification, anaerobic digestion, etc. As the title implies, waste incineration is the process of burning municipal waste with the main goal of high temperature gas flue production. The hot flue gas can then be utilized for just heat production or cogeneration of heat and electricity. Waste incineration is a popular solution because not only energy can be freely produced, but the critical problem of waste disposal can be resolved to a very large extent, as it reduces the solid mass of the original waste by 80–85% and the volume of compressed waste in the trucks by 95–96%, depending on composition and degree material recovery such as metals from the ash for recycling. Currently, the year-round base load of district heating system is supplied by waste incineration plants in Denmark and the plan is to increase this share in the future. Many other European countries are recovering heat and electricity from waste today. In modern incinerators, flue gas cleaning is effectively done in order to minimize the environmental impact of the process [14]. Waste incineration can be in combined configuration, i.e. cogeneration of heat and power, or in heat only production scheme. There are a wide range of waste incineration technologies.

Renewable energy plants: Solar thermal and geothermal plants are the technologies that may directly contribute in the heat preparation of the 4GDH while wind farms can take part in indirect form by running electricity-driven heat production systems like heat pumps, electrical boilers, etc.

In a solar thermal system, which may come in either large-scale or stand-alone form, sunlight is converted into heat in the form of a hot working fluid. A solar thermal system consists of the collector array, the storage system and the solar regulator system (e.g. temperature difference control). The collector array is the key element of solar thermal system absorbing the solar energy and transferring it into a working fluid. This hot fluid be transferred to the heat exchanger via pumps with a minimal heat loss. Then, it will heat the secondary fluid via heat exchanger either in direct use or in the storage unit. The distance between collector and storage unit should be as short as possible to minimize heat loss. In solar thermal units, in contrast with the solar thermal power systems in which concentrating collectors are employed, either flat plate collectors or evacuated tube collectors are used. Evacuated tube collectors are generally more efficient as they minimize the heat losses due to the vacuum space between the absorber in the cover around it [15].

In a geothermal system, heat can be produced directly by digging geothermal boreholes and extracting the thermal energy of the lower layers of the earth. There are two main possible ways of using geothermal energy as direct and indirect methods. In a direct use of geothermal system, hot pressurized water is directly produced through via the heat exchangers supplied by the geothermal boreholes. As uniform-continuous energy may be supplied by such a system, no heat storage unit is required here. Therefore, the main elements of a direct geothermal heat plant are the deep vertical boreholes, the pipes used to transfer the working fluid down to the earth's depth, pumps, and a heat exchanger to transfer the collected heat to the pressurized water of the district heating system. On the other hand, in the indirect use of geothermal energy, usually, horizontal boreholes are employed as the evaporator of a heat pump. In this way, the system can be in both large-scale and stand-alone types depending upon the size of the heat pump. The main components are a series of underground pipes (the geothermal loop), the heat sink and a geothermal heat pump to move heat between the heat sink and the ground [16].

The next type of renewable plant that can be integrated into the 4GDH is wind power. Although, wind farm output, i.e. electricity, cannot be directly injected into the district heating network, it can be used for driving electrical heat production systems such as heat pumps or electrical boilers in both large-scales and small-scales. This can be widely used in the future energy systems as wind power is fairly cheap compared to other electricity produced by other systems. Specially, in off-peak times, surplus wind power can be used by these electrical driven facilities to immediately support the local district heating network or heat up the heat storage facilities for peak-shaving [17].

Heat pump: Naturally, heat can only flow from an environment with higher temperature to another with lower temperature. The reverse path is only possible if a mechanical machine receiving work is used. Heat pump is the machine that receives electricity, as work to drive a pump, to transfer heat from a cold ambient to a warm/hot space. Like every other thermal engine, heap pump also works between two thermal sources, i.e. the lower temperature source which is called heat source and the higher temperature source to which heat is delivered and is called heat sink. The very positive point about heat pump is that like other refrigeration systems it offers a high efficiency (coefficient of performance - COP), usually well above 100%. Depending on the operation conditions and temperature of heat sources, heat pumps may offer COP as high as 4-5. In means the device can produce 4-5 times more heat than the electricity it has received to operate. Generally, the lower temperature of supply and the lower temperature difference of the heat sources for a heat pump, the higher efficiency it offers [18].

In a heat pump, there are four main components. These are the compressor that runs the working fluid in the system consuming electricity, the condenser through which the hot high pressurized working fluid exchanges heat with the heat sink (it is a heat exchanger), the expansion valve that depressurizes and cools down the working fluid and finally the evaporator through which the low temperature, low pressure working fluid exchanging heat with the heat source and evaporates (another heat exchanger). The working fluid of the heat pump is called refrigerant and it is usually ammonia, R-134a, R-410a, CO_2, water, etc. depending on the application of the heat pump. Typically, there are three main types of heat pump as air source, ground source and water source, referring to the substance of the heat source.

Heat only boilers: As the name implies, this device produces heat as the only and main product in the form of hot water by burning fuel or consuming electricity. It can be integrated into the district heating system in both large and small scales. This device can be a very good candidate for the 4GDH system, especially in large-scales, as not only it can be fed by biomass feedstock, e.g. wood pellets or biogas, the electrical type of this device can make an integration of the district heating system and the renewable based electricity grid which is very crucial in the future energy systems [19].

Flue gas condensation: Although flue gas condensation is not a specific technology but only an approach to get higher efficiency from the fuel-fired energy production systems, it is going to be briefly discussed due to its importance in the future energy systems. In all of the fuel-based heat production technologies, such as CHP plants, boilers and waste incineration systems, there should be a chimney to get rid of the combustion/incineration products in the form of flue gas after being utilized for energy extraction. In the conventional technologies, the gas flue has still high temperatures so that the water content of the combustion/incineration products is in vapor phase, wasting a considerable amount of energy to the environment. Flue gas condensation is a process in which the flue gas, after the primary energy exchange process, moving toward the chimney is cooled below its water dew point. In this way, the heat released resulting from the condensation of water is recovered as a low temperature heat flow. This flue gas cooling process can be fulfilled directly with a heat exchanger or via a condensing scrubber. The excess condensed water is continuously removed from the process. Therefore, flue gas condensation is an extremely cost-effective method of recovering energy for the district heating system heat production chain. The amount of energy recovered depends on the desired temperature of the district heating system so that the lower temperature of condensation, the more energy can be recovered. As such, the heat recovery potential of flue gas condensation is higher for fuels with higher moisture content like biomass and municipal waste. Thus, flue gas condensation is more popular for biomass-fired boilers and waste incinerators plants connected to district heating systems in which relatively low temperature heat is required. The 4GDH implementation increases the amount of heat recovery from the flue gas as even lower temperatures than before is required there. It is also possible to improve energy recovery by using a heat pump. For this, the flue gas is considered as the evaporator, i.e. heat source, of the heat pump, condensing the flue gas and releasing significant amount of energy for the heat pump. Overall, a flue condensation system without a heat pump can increase energy recovery by up to 15% while the increment for an installation with a heat pump is over 20% [20].

Heat storage: Clearly, heat storage is not a way of heat generation but storing heat in off-peak hours and supplying the stored heat back to the end users when appropriate. Therefore, it is one of the most important components of heat supply chain of the 4GDH and this is why it is discussed in this section. Heat storage is possible into different schemes of seasonal storage and short-term storage each of which may be in various configurations.

In short-term storage, sensible heat storage or latent heat storage are possible. In latent heat storage systems heat is stored as latent heat during change in the phase of a material. These materials are called Phase Change Materials (PCMs) which are categorized as organic and inorganic materials. Selection of material depends on the application and temperature range of heat storage unit. Sensible heat storage is increasing the temperature of storage medium without any change in its phase or chemical components. Generally, sensible heat storage is more appropriate for district heating applications, both in small and large scales, due to the simpler design and application in the low required temperatures. Here, water, industrial oil or brine can be used as the storage medium.

In short-term sensible heat storage, one of the popular options for low temperature grades is hot water storage tank (container). Charging and discharging of this storage tank can be implemented in various ways. One of these configurations, which is the favorable type of storage tank for variable temperature HD system, is a tank with one helical coil (heat exchanger) with the hot water coming from the district heating supply line while the DHW stream is a mixing flow through the tank. The spiral coil is used here because the DH supply water is not potable and cannot be mixed with the DHW flow. The DHW flow is mixing (not through another coil) because having an extra heat exchanger for the DHW flow reduces the efficiency of heat exchange. Generally, there may be or not be an auxiliary heater in the tank (usually electrical coil), however, for LTDH system working below the desired storage temperature, there must be an auxiliary heater to heat up the water for higher levels. In addition to this configuration, the other possibility is having a tank in which the district heating supply flow is mixing and the heat exchange between the district heating flow and the DHW flow is done through an extra internal/external heat exchanger. Figure 2-4 illustrates these storage configurations which are used for DHW preparation of the end users. A comprehensive discussion of storage configurations in the district heating substations is presented in [21].

As such, seasonal storage is the storage of heat (or even cold) for long periods, e.g. several months. The idea behind this is to store the free/cheap and surplus heat available in a season, typically warm months, and use it during the cold months of the year. For example, solar energy which is naturally absorbed by soil or the solar heat collected by active solar thermal systems during summer may be stored to be used during winter. Any other heat stream or the output of any heat production system, such as waste heat from industrial process or heat pump output energy etc., can similarly be stored for long period later use. Seasonal heat storage can be a very effective method at service of district heating systems for single buildings or complexes, moderating the demand of the consumers. Depending upon the application and size, the seasonal heat storage system may be designed for a maximum temperature of 27-80 °C. Seasonal storage can also be in both small-scale and large-scale, however, the large the system is the better because the size picks up, the energy efficiency of the system increases and the cost falls [22].

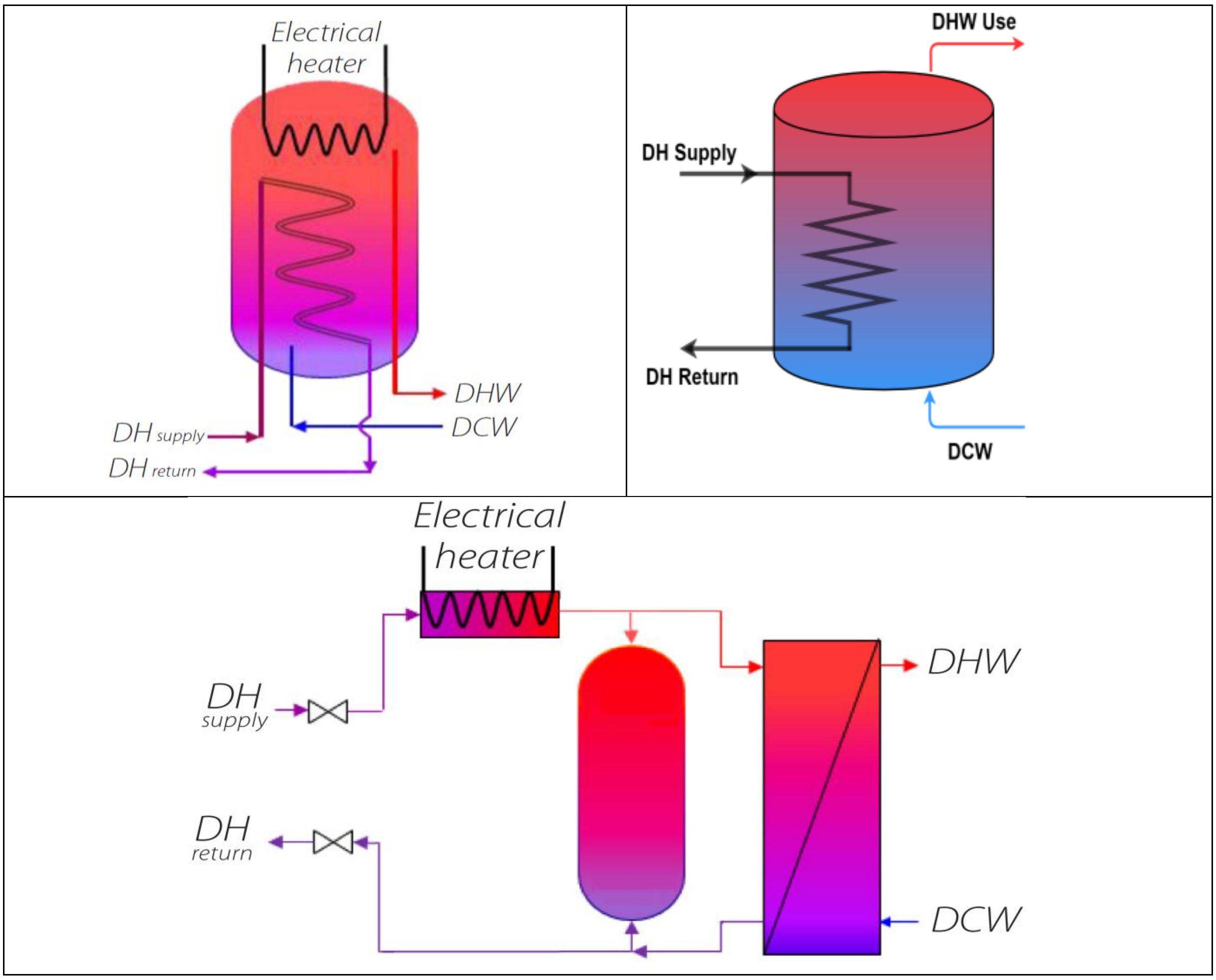

Figure 2-4 Different configurations of storage units with indirect and direct charging/discharging for DHW

2.3 Recent studies

In this section, a comprehensive literature of the most recent studies on the various relevant aspects of the 4GDH is presented. Basically, these studies can be categorized into different classes based on their main focus and scope, namely, LTDH and the relevant aspects, integration of a district heating system into other energy sectors, legionella issues, the ways and methods for transition from current district heating system to the 4GDH, utilization of renewable and recycled heat form low temperature sources and integration of more advanced technologies in the supply chain of the district heating.

Feasibility of lower temperatures: There are a large number of studies focusing on the feasibility of lower temperatures in the district heating system. Olsen et al. [21] presents different ideas and recommendations for DH systems to enable a low supply temperature. The goal is to support the consumers with 50 °C district heating supply line. Different technical solutions are presented within the work together with identified critical components in a district heating system. One conclusion shows that it is possible to guarantee an efficient operation but it is very critical to obtain the proper functions, i.e. high effectiveness of heat exchangers, in each substation. Otherwise, not enough low return temperatures could be reached. Another conclusion is that there is no superior substation concept, but the best system should be adapting to the specific characteristic of the site and heat demand. The presented concepts could be used for a range of supply temperatures from 50-55 °C up to 60-70 °C. Winger et al. [23] implemented a combination of high and LTDH networks connected to an organic Rankine cycle for an energy efficient building. The Organic Rankine Cycle (ORC) is named for its use of an organic, high molecular mass fluid with a liquid-vapor phase change, or boiling point, occurring at a lower temperature than the water-steam phase change. The fluid allows Rankine cycle heat recovery from lower temperature sources such as biomass combustion, industrial waste heat, geothermal heat, solar ponds etc. The low-temperature heat is converted into useful work that can itself be converted into electricity. In this system, the organic Rankine cycle generates electricity using high-temperature district heat and the excess heat of the system is used to heat the building via a low-temperature heat distribution system. The combined system of this building is also integrated with micro-scale wind turbines and PV systems to achieve higher levels of sustainability and energy efficiency. Relying on the latest state-of-the-art and state-of-practice of district heating knowledge, the

Literature Review

R&D section of Aarhus district heating Company (AVA) presented a thorough review study aimed at providing a guideline for standardizing the LTDH features, including the facilities, characteristics, etc. The report includes recommendations for establishment, design and operation, specifications for consumer connections and substations as well as DHW of a LTDH [21].

Persson and Munster [24] presented another study on the future district heating systems in Europe aiming at estimating the total volume of municipal solid waste available for heat recovery in European district heating systems in 2030. The main conclusion of this study is that the effectiveness of lower temperature district heating system, which results to more efficient heat production process in incineration plants, will even increase in future when more advanced incineration technologies come up. Ommen et al. [25] employed a Mixed-Integer Programming optimization model to investigate the impact of lower temperature district heating system on the existing and future energy systems. Their results showed that by decreasing the district heating operation temperature instead of 80-85 °C, the primary fuel consumption is reduced 5–7% at temperatures in the range of 60–70 °C while surprisingly the operation temperature below 60 °C will result in increasing cost and emissions. This was mainly due to the fact that low water flow rate through the district heating pipeline during summer leads to too large temperature drop along the pipeline, necessitating heat recovery in the supply line. Yang et al. [26] carried out a thorough energy, economy and exergy evaluation of the solutions for supplying DHW from LTDH. With the main goal of providing possible solutions, the study presented various propositions for district heating systems with the low supply temperatures of 65 °C, 50 °C and 35 °C and for two different building topologies. Based on the results obtained for the energy, exergy and economic performance analyses, the configurations of the devised DHW substations were optimized to fit with both low and ultra-LTDH systems. The benefits of lower return temperatures were also analyzed compared with the current district heating situation. Ostergaard and Svendsen [27] assessed the size of radiators and the level of heat demand in a few existing Danish single-family houses from 90 years back. The results showed that there is a large potential to use LTDH in these houses and critical radiators must be replaced in order for obtaining the full potential of LTDH. If these radiators were replaced, it would have been possible to lower the average heating system temperatures to 50 °C in supply line and 27 °C in the return line for all of the houses. Köfinger et al. [28] examined the feasibility of district heating networks for areas with low heat demand and low-energy building in Austria. The results of the study showed that the availability and economic conditions of low-temperature heat sources are a key factor in facilitating LTDH networks. In rural areas, lower heat losses due to lower network temperatures are beneficiary for the performance of the LTDH network.

Schmidt et al. [29] highlight the advantages and restrictions on the way to go for LTDH systems in Germany. They comprehensively discuss how significantly a lower temperature of supply may result to more efficient use of energy sources and better integration of renewable energy in the district heating system. Reduction of losses in the distribution and transmission lines, increasing the efficiency of the supply units and minimizing the irreversibility in the district heating system are the further mentioned advantages of utilization of the LTDH in this work. Kallert et al. [30] made a thorough exergy-based analysis of renewable multi-generation units for small-scale LTDH with the main target demonstrating the advantages of exergy-based assessment for increased efficiency of small-scale district heating supply schemes. For this objective, the authors investigated different supply scenarios, based on fossil and renewable energy sources regarded individually or in multi-way combinations. The work presents a prospect for an optimized LTDH and offers a more holistic understanding of the energy conversion chain of LTDH. Zuhlsdorf et al. [31] investigated the optimal integration of booster heat pumps in ultra-LTDH (35-50 $^{\circ}$C) and compared to the performance of their proposed system with conventional LTDH. They found out that by using ultra-LTDH with booster heat pump the performance of the network will improve by 12%. In another work, Yang and Svendsen [32] analyzed the return temperatures of different types of substations for DHW preparation with ultra-LTDH in temperature ranges 35-45 $^{\circ}$C, and developed improvements in the substation for better energy efficiency. They analyzed the seasonal impacts of the return temperature from the DHW loop on the overall return temperature of district heating. To achieve lower return temperature and higher efficiency for DHW supply, an innovative substation was devised. Vetterli et al. [33] studied the feasibility of using LTDH for a district called “Suurstoffi” in Central Switzerland. The main goal of this work was to ensure a fully renewable energy supply and CO_2-neutral operation. In this work, the monthly analysis of more than 300 measuring points and complementary user surveys resulted in numerous findings, the most relevant observation being the performance gap identified for the heat demand and electricity consumption of the heat pumps and auxiliary equipment. Flores et al. [34] explored the advantages in district heating operation by connecting the new energy-efficient building loads via LTDH subnets to a conventional district heating system, supplied by a CHP plant. A techno-economic analysis was performed to estimate the annual district heating operating costs and revenues achieved by the LTDH. The savings were due to the reduction in distribution heat losses, lower pumping power demand, improved power-to-heat ratio for electricity production and enhanced heat recovery through flue gas condensation.

Legionella problem: As discussed in the previous section, one of the main concerns of lower temperatures in district heating systems is the legionella generation problem. In addition to the technical studies on this issue, there are a number of works implemented on legionella problem specifically in district heating systems. Leoni et al. [35] presented an investigation of the outbreak of legionella bacteria in hot water distribution systems in Italy and found out that the water recirculation system used by centralized boilers spreads the bacteria throughout the entire network. Averfalk and Werner [36] presented the effective solution of low water volume at the secondary side and fast DHW preparation. The idea is to have instantaneous water discharge by the heat exchanger, giving a short residence time to the water flow. Therefore, individual district heating substations for each apartment may avoid the requirement of large DHW circulation which further relieves the issue of legionella. Finally, the standard DHW temperature of 60°C is suggested while temperatures below 50°C must strongly be avoided. In another work in this area, Yang et al. [37] addressed the concern about legionella when applying LTDH in conventional systems with DHW circulation. In this study, a system with decentralized substations was analyzed as a solution for legionella generation problem was proposed. Also, the arrangement of the decentralized substations was modified to decrease the average return temperature. The models of conventional system with medium-temperature district heating, decentralized substation system with LTDH, and innovative decentralized substation system with LTDH were built and the distribution heat loss of these scenarios were compared. This research is only research that studied decentralized heat storage units similar to this thesis research. However, it mainly emphasizes on solving legionella problem in a low temperature system while in our proposed system with non-uniform temperature system there is no concern about this problem. A report published by the Swedish District Heating Association [38] points out to towel dryers and floor heating coils connected to district heating as the high risks systems for legionella generation, when the devices are turned off. This problem can be solved by running the devices with separate circuits from the original DHW system.

District Heating in future energy systems: There are some studies that highlight the importance of district heating in the future energy systems in which the penetration of renewable energy is extremely high and there is integration with other energy sectors, especially, the electricity grid. Neuman et al. [39] studied the principles of smart grids on the generation of electrical and thermal energy and control of heat consumption within the district heating network. In the work, the control of electricity and heat is carried out by using advanced optimal control methods, including different configurations of facilities and the substations. Ancona et al. [40] presented several different layouts for the utility substations in smart district heating networks with distributed heat generation systems, such as solar thermal panels or micro-CHP units, etc. The layouts were developed to allow the bidirectional heat exchange at the utilities, optimizing the heat exchange as a functional of the network design temperatures, utilities thermal energy requirements and the characteristics of the production system. The simulation was carried out in IHENA software and the results obtained were analyzed considering the implementation of the elaborated layouts. Schüwer et al. [41] analyzed the impact of CHP systems on the German real time electricity market and showed how decentralized CHP units are flexible and suited to provide balancing power, thereby contributing to the integration of renewable energy. In this way, not only CHP can be an efficiently active renewable power balancing source in the grid, but also it can cost-effectively support the district heating system integrated with the electricity grid. Xiong et al. [42] indicated that more flexible dispatch rules and integration between the electricity sector and the district heating networks can be very promising for increasing wind integration in the future energy systems. Lund and Persson [43] investigated the use of large heat pumps for district heating and showed that potential heat sources are present for 99% of district heating networks in Denmark, where the concentration of heat sources is bigger around larger cities. They anticipated that sea, wherever available, will play a key role as a heat source of heat pumps in the 4GDH. Chiu et al. [34] investigated the use of mobile thermal energy storage employing phase change materials for transporting of industrial surplus heat for use in LTDH networks, finding it effective to save the costs of district heating networks by 25%. Lake et al. [44] presented a review of district heating and cooling systems for a sustainable future, including various relevant topics such as the history of district energy systems, system identification, energy sources, design considerations, environmental impact, economic feasibility, performance analysis and the role of energy policy.

Characterizing the 4GDH: There are a number of works that specifically use the term “the 4GDH” and conduct research projects on this issue. The concept of 4GDH was first introduced in 2014 and a clearer definition of it was presented later in the same year. In 2014, Lund et al. [45] presented the first comprehensive work on the definition of the 4GDH and its objectives. In addition to identifying the future challenges of district heating technologies, they explain which components should be removed from the 3rd generation of district heating and which ones should take a higher share in the future district heating system. Then, in 2016, they presented a summary of the most important works presented in the 1st international conference of the 4GDH [12]. In two works, Ziemele et al. [46,47] studied and modelled the transition to the 4GDH in Latvia. For this, a system dynamics model was developed to analyze the current district heating system of Latvia and various policies, such as risk reduction, efficiency improvement and subsidies were suggested for this transition. The results showed that such policies may accelerate the transition. Lund and Mohammadi [48] studied the insulation standards for the transmission systems of the 4GDH. For this, a new method, combining a detailed heat loss analysis with an integrated energy system analysis, for assessment of the insulation level for district heating was proposed. The method was investigated for the case study of Denmark’s district heating and showed that pipes with higher insulation standard (series 3) are generally preferable, but the highest insulation standard available today (series 4) might be better in the future if either fuel prices increase or investment costs decrease. Verda et al. [49] presented a study on the effect of thermo-economic cost assessment in the future district heating networks. Simple examples were analyzed to provide quantitative evaluations of the various cost terms, depending on the operating conditions, topology and characteristics of the users and the producers. Tereshchenko and Nord [50] discussed factors associated with the decisions on energy supply unit of the future district heating systems considering various highly efficient energy conversion technologies taking into account the economic aspects and technical limitation of the technologies. They suggested a new heat supply optimization based on detailed performance simulation models developed in Aspen HYSYS software and data post-processing in MATLAB. Weidlich [51] highlights the test requirements for the joints of pipes in the future district heating systems. Some recommendations for the improvement of the test procedure were given and expected operational loads in modern district heating networks, where the share of CHP systems and renewable energy is high, were discussed. Paiho and Reda [7] discussed the relevant issues for a smooth transformation of the current district heating system towards the 4GDH for Finland. The authors emphasized on the crucial role of energy-efficient buildings, high level of insulation, low temperature indoor distribution systems etc. in the 4GDH. Kamal [52] did a feasibility study and an investigation of the potential for LTDH systems and integrating to the 4GDH with the existing technology. He proposed, analyzed and discussed four different integration solutions between old and new networks. The results showed that first, LTDH could lead to reduction in the initial cost for the network by using PEX instead of steel as pipe material, and second, the cost of the retention flow linked with 4GDH is almost 20-30% of the total cost. Also, the study showed that 55 °C is a satisfying temperature during winter season where 60-65 °C is suitable for other seasons. The variation depends directly on the temperature drop through the supply pipes to the consumers. Averfalk and Werner [53] highlighted the essential improvements in the future district heating systems, providing seven recommendations on the design and construction strategies of the 4GDH. The recommendations are associated with the seven factors of temperature levels for the heat distribution system, recirculation, metering, supervision, thermal lengths for heat exchangers and heat sinks, hydronic balancing and legionella.

Mapping and planning: Another category of works in this framework is mapping and planning. Nielsen and Moller [54] presented an article focusing on developing a method for assessing the costs associated with supplying all the buildings in Denmark by expanding the existing district heating networks based on a geographic information system. Stennikov and Iakimetc [55] focused on the need to solve problems of planning and presented a complex methodology which allows defining the locations of heat sources. The authors concluded that the less value of heat density in the system, the higher specific costs for generation, distribution and transmission of heat energy. Petrović and Karlsson [56] presented an energy atlas based on a highly-detailed geographic information system for a given municipality in Denmark. The paper showed that significant heat saving potential lies in farmhouses and detached houses as well as in buildings built before 1950. The biggest part of heat saving potential is found around biggest towns and heat savings should be done in buildings supplied by oil and natural gas boilers. Overall, it is concluded that changing from the current district heating system to the 4GDH requires institutional and organizational changes that address the implementation of new technologies and enable new markets to provide feasible solutions to society as well. It is important to mention that management issues are out of scope of this technical project and therefore, will not be addressed in this work.

3 Methodology

3.1 Various Design Possibilities

In this chapter, the most likely yet feasible design of district heating networks that is compatible with the 4GDH concept is introduced. The objective is to finally compare the performance of this technology with the innovative one proposed by this thesis. Therefore, the existing generation of district heating, so-called the third generation of district heating (3GDH), is also compared and introduced in details to finally have a comprehensive comparison of the proposed system, the other feasible 4GDH solution and the currently in operation district heating systems.

3.1.1 The 3GDH system

The 3GDH technology is, in fact, the currently in-operation district heating scheme in most of the world's energy systems. In this system, the design supply temperature is mainly in the range of 70-85 $^{\circ}$C, while the design return temperature is in the range of 35-45 $^{\circ}$C [57]. Apart from the dramatic change of heat production units, which is out of the scope of this work, the lower supply and return temperatures, the new generation of pipes, and the configuration of the substations are the main differences of the 3GDH system with the second generation of district heating systems [58]. Two substation designs are normally used in the 3GDH system. These substations are the Instantaneous Heat Exchange Unit (IHEU) and the District Heating Storage Unit (DHSU). In an IHEU, the DHW section has just a plate heat exchanger, while in a DHSU, besides that, there is a buffer tank that stores heat for peak shaving. The storage tank in the substation results in a significant reduction of the piping size and heat exchangers, however, it causes a bigger rate of heat loss in the system [21]. The space heating section in both of these substations has only a plate heat exchanger. Figure 3-1 shows the schematic of an IHEU (a) and a DHSU (b). Figure 3-1 presents information about the temperature values before and after the substations based on the Brazilian comfort standards. T_A is the temperature of domestic cold water, which is equal to room temperature.

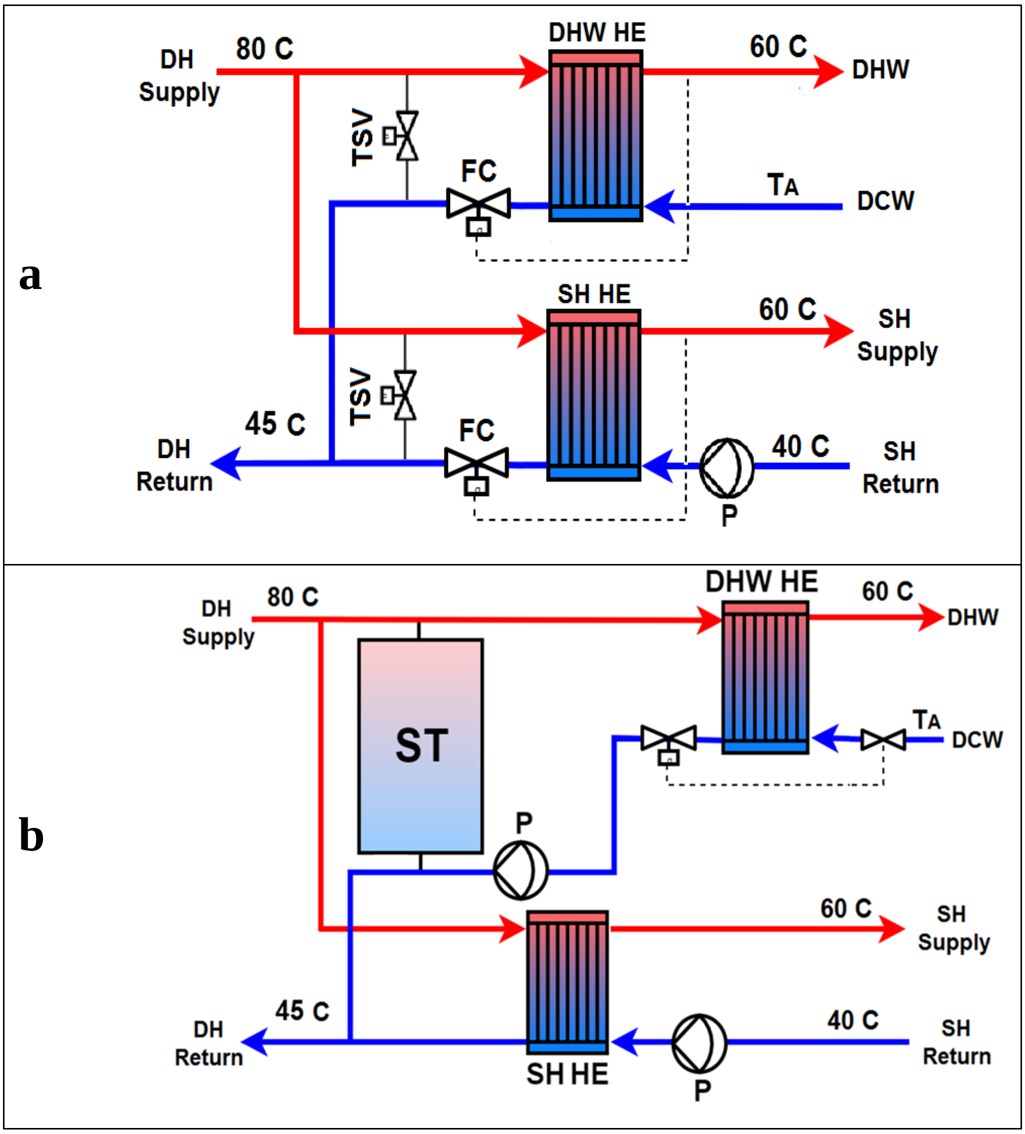

Figure 3-1 Sketch diagram of the two main substation types in the 3GDH systems; the IHEU (a), the DHSU (b); DH: district heating, DHW: domestic hot water, DCW: domestic cold water, HE: heat exchanger, TSV: thermostatic valve, FC: flow controller, P: pump, SH: space heating.

3.1.2 The ULTDH system

The ULTDH system is similar to the 3GDH system in various aspects, e.g. the piping system, heat suppliers, etc. The only difference between these systems is the extremely lower temperatures of the ULTDH scheme, where the supply and return temperatures are in the range of 40-45 $^{\circ}$C and 20-25 $^{\circ}$C, respectively. Naturally, as the supply temperature is below the comfort standard of DHW, direct supply of hot water may not be possible. Therefore, either the ULTDH system is for space heating only (an auxiliary heater in the building covers the DHW demand) or the substation is equipped with a heat pump to increase the supply temperature to 60 $^{\circ}$C. This substation design is called heat pump furnished unit (HPFU). Figure 3-2 represents the schematics of a substation with an individual electrical heater (IEHU) (a), and an HPFU (b). As seen in Figure 3-2-a, the hot water supplied by district heating is used for space heating only (via a plate heat exchanger) while in Figure 3-2-b, in addition to space heating, the district heating water temperature is boosted to 60 $^{\circ}$C by a heat pump for covering the DHW demand of the building. A thorough explanation of various HPFU designs may be found in [59].

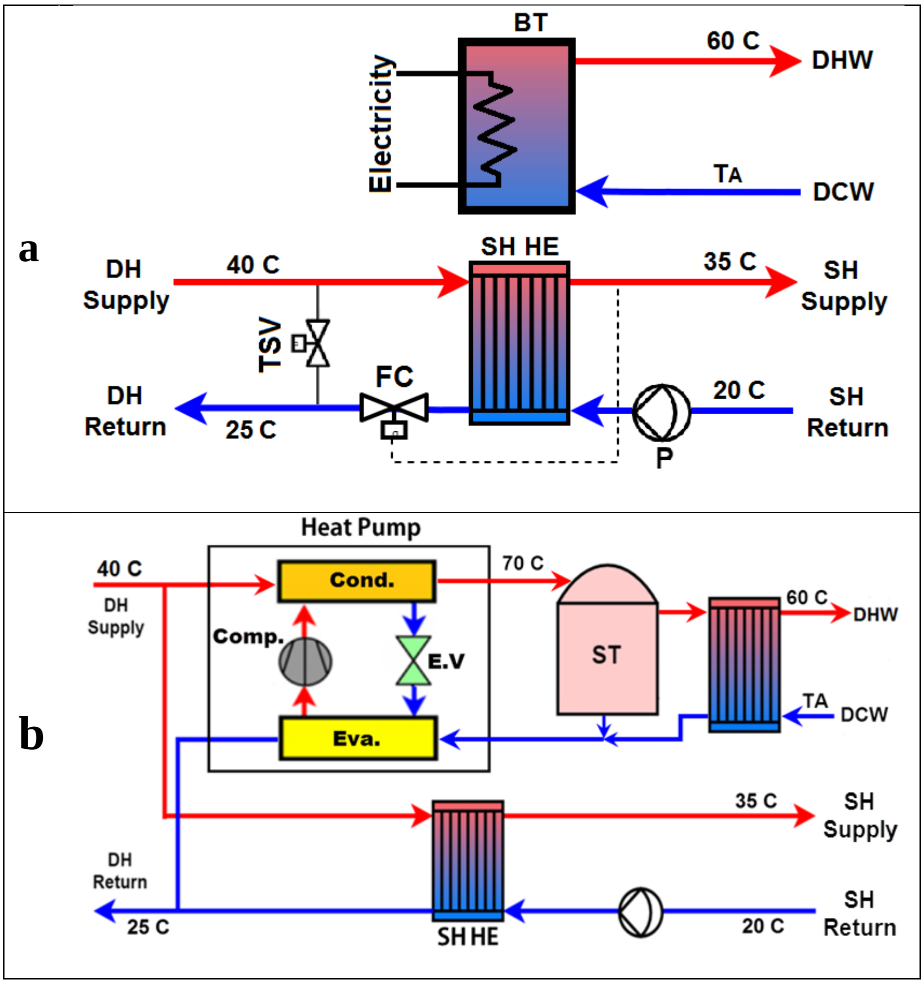

Figure 3-2 Sketch diagram of the substations in a ULTDH system; an IEHU (a), an HPFU (b), ST: buffer tank.

3.1.3 The Proposed Concept

As explained, one of the focuses of the studies on district heating technologies is to reduce the rate of heat losses for which the primary measure is to decrease the supply temperature as much as possible. This, however, can be challenging due to the main reasons of (i) the comfort standard DHW temperature, and (ii) the risk of the legionella. The legionella problem appears as a concern when the water temperature is between 25-45 ^{o}C and the standing time is longer than 2 days [60]. Since there is an extensive literature about the legionella problem in heat distribution systems, no further information about this issue is presented here. The DHW standard temperature varies from one country to another. For Brazil, the standard comfort DHW temperature is 60 °C [61]. Also, the standard comfort room temperature in residential buildings must be set at 26°C during summer and at 22°C during winter [62].

In this work, the new concept of non-uniform temperature district heating (NUTDH) system is proposed. This system works based on the predominant supply temperature of 40-45 $^{\circ}$C (for 20 hours a day) and the periodic high temperature of 70-75 $^{\circ}$C (for four hours a day). In the NUTDH system, the low-temperature supply is mainly for space heating, and the high-temperature supply aims at providing the DHW of the network. Since the high-temperature supply is not available all the time, the network is equipped with decentralized heat storage units to be charged during the high-temperature supply mode and be discharged during the regular operation of the system. In other words, the NUTDH system aims at providing the DHW demand of the network at the standard comfort temperature and preventing the legionella risk, while a very low rate of loss is achieved. The design low and high supply temperatures are chosen based on the comfort standard temperatures, the temperature drop along the pipeline and the expected effectiveness factor of the heat exchangers.

As the NUTDH system is shortly on the high-temperature mode, the system needs heat storage units to be able to provide hot water for all day long. This work proposes the use of neighborhood-scale heat storage units. Figure 3-3 presents the schematic diagram of the NUTDH system. According to Figure 3-3, the main district heating lines bring the heat to the neighborhood (the streets here) and the street branch pipes distribute the heat along each neighborhood. There is a further street connection line for connecting the storage tanks to the end-users. This supply/return twin pipe is much smaller than the main street connection pipe because not only it just provides the DHW demand, but also the simultaneity factor is quite low (due to the large number of houses in each neighborhood).

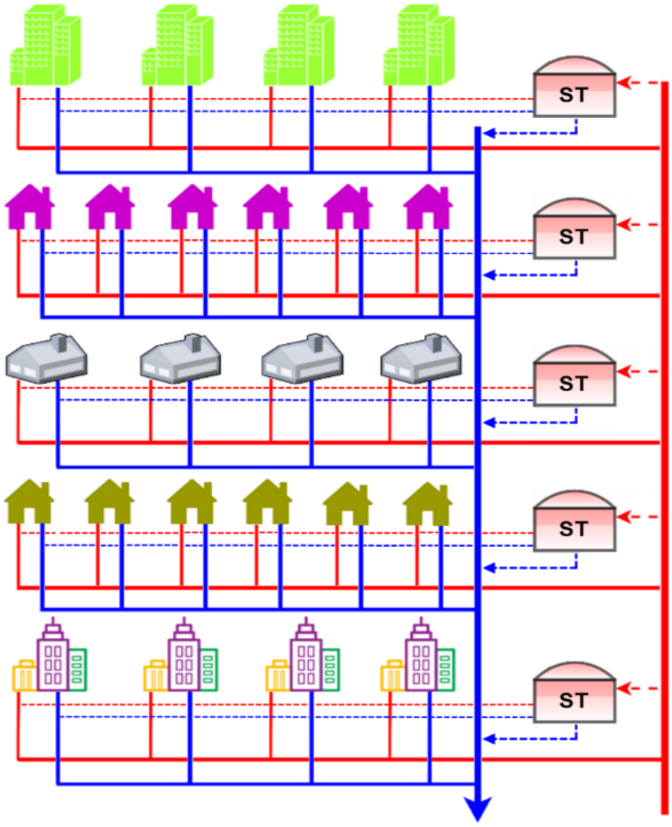

Figure 3-3 Schematic of an NUTDH system; ST: storage tank, red line: supply pipe, blue line: return pipe, solid-lines: flows directly from/into the district heating system, dashed-lines: flows from/into the storage tank.

A further important point about this design is the way that the storage tanks are charged and discharged. As the storage tanks are to store district heating water, buffer tanks can be employed. Heat storage buffer tanks are stratified in various temperature levels (low temperatures below and higher temperatures above). The tank receives hot water from the top and the cold water goes out (to the return line) from the lower part of the tank. As the tank is charged, the temperature of the bottom of the tank also increases. If the control system does not work efficiently, the storage tank may discharge a considerable amount of heat back to the return line, causing a high rate of loss. For solving this problem, the storage tank employs a control valve that operates as a linear function of the temperature difference between the tank inlet and outlet flows, such that the lower the temperature difference gets (i.e. higher temperature of the bottom of the tank), the less the valve opens. Figure 3-4 shows the storage tank configuration (a), and the control valve operating curve (b). In this figure, CV-1 is the electrical flow control valve for controlling the charging process, and CV-2 is the control valve that opens as much as the neighborhood demands. As seen in Figure 3-4-b, the maximum temperature difference is 30 $^{\circ}$C. As the temperature difference gets lower, the valve proportionally closes. The valve will be fully open for the higher temperature difference values than 30 °C.

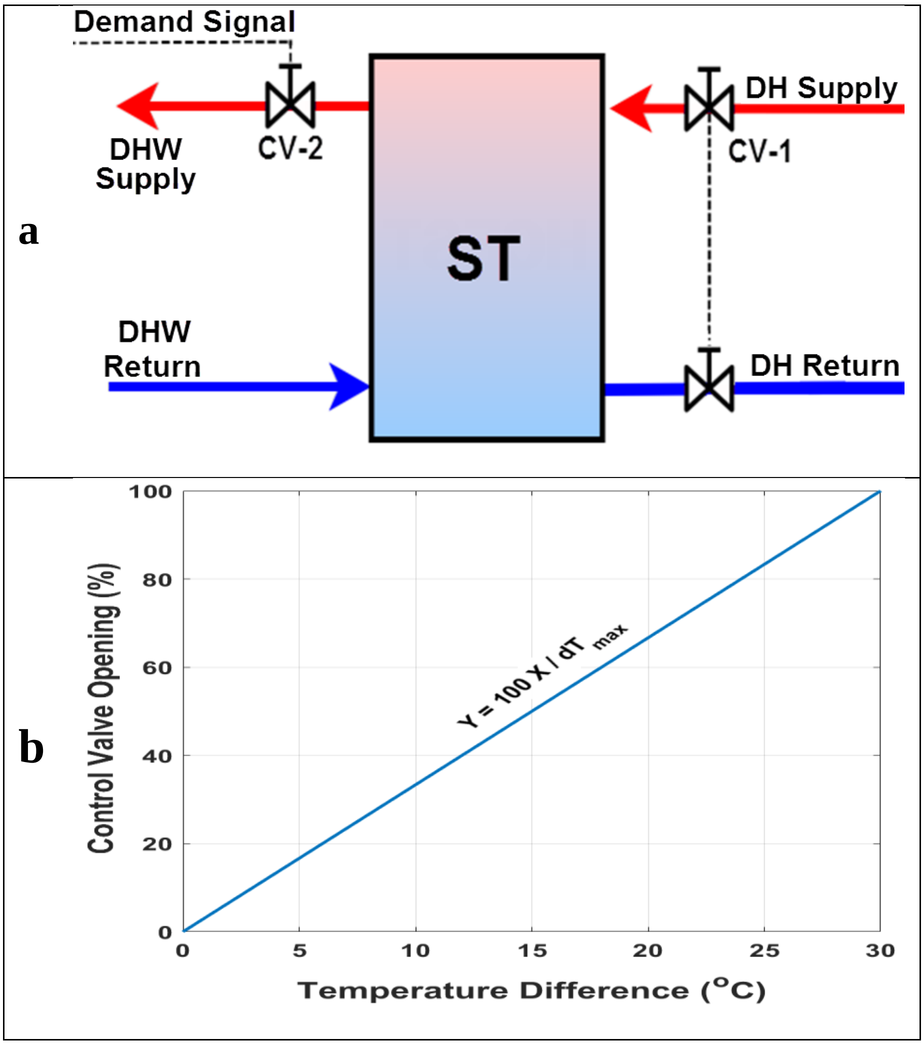

Figure 3-4 The storage tank operation/control strategy; the tank configuration (a), the control valve opening strategy (b), CV: control valve.

Finally, Figure 3-5 shows the operation strategy of the NUTDH system for a sample of three-tank network when working in high-temperature mode (a) and low-temperature mode (b).

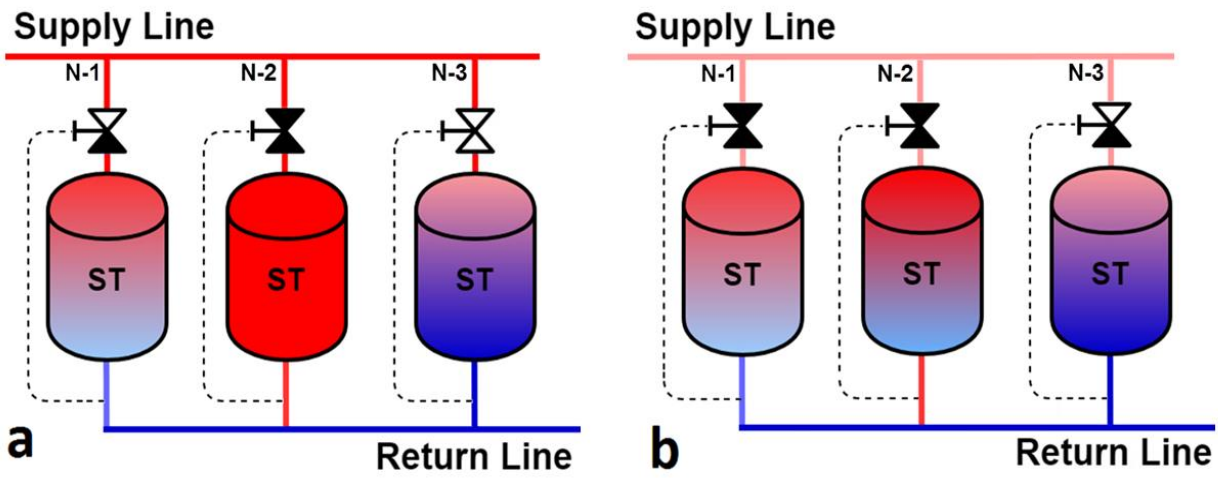

Figure 3-5 Sketch of the NUTDH system operation strategy;
high-temperature supply mode (a), low-temperature supply mode (b), N: neighborhood number;
black valve: fully closed, white valve: fully open, black-white valve: partially open.

According to Figure 3-5-a, if the system is in the high-temperature mode, the supply line charges all the tanks until they are fully charged. Figure 3-5-a shows that the storage tank of the second neighborhood is fully charged and as a result, its control valve is fully closed. In this case, neighborhoods 1 and 3 are being charged yet. As the temperature of the third tank is lower than the first one, its control valve opens more than the other one. Figure 3-5-b shows the operating strategy of the system when the district heating system is on the low-temperature mode. In this case, the storage tanks are only charged if they are sufficiently cold. In the illustrated case, it is only the control valve of the third tank which is partially open due to the too low temperature of this tank.

It should be mentioned that the district heating system could be off during most of the days in summer as there will not be any space heating demand and the system only comes into operation to charge the storage tanks during the high-temperature mode.

Table 3-1 presents information about the characteristics considered for the 3GDH, the ULTDH and the NUTDH systems in this work. It is worth noting that the maximum pressure for convention piping system is 15 barg and on the other hand, there is no need to heat exchanger for space heating in pressure values under 10 barg. For this reason, an average value of 12 barg was chosen for all three systems.

Table 3-1 The features considered for the three different district heating schemes of this work.

Parameter	Information		
	3GDH	ULTDH	NUTDH
Space heating temperatures (°C)	70/40	35/25	60/40 or 35/25
Design supply temperature (°C)	80	40	75/45
Design return temperature (°C)	45	25	35/25
Substation design (DHW/SH)	IHEU or DHSU/IHEU	IEHU or HPFU/IHEU	IHEU/IHEU
Nominal pressure (barg)	12		
Space heating system	Radiator		
Pipe type	Twin [27]		
Insulation class	Series II and series III [27]		
Min pressure difference at the substation (bar)	0.3		
Max volume of each pipe after the substation (lit)	3		
Maximum media speed (m/s)	2		
DHW supply temperature (°C)	60		
Room comfort temperature (°C)	22		

3.2 Modeling and Simulation Method

As explained, this work includes a thermodynamic modeling and analysis of a 4GDH system being implemented in a case study. In addition, the work will assess the compatibility of the twin pipes currently being used in the 3GDH system with the proposed thermo-hydraulically system. Thus, the mathematical model required to accomplish the simulations for these two objectives is presented in this chapter.

3.2.1 Thermodynamic Model

The general format of the first law of thermodynamics or energy conservation law for a control volume is as Eq. 3-1 [63]:

$$\frac{dE_{C.V}}{dt} = \dot{Q}_{C.V} + \dot{W}_{C.V} + \sum \dot{m}_i \left(h_i + \frac{Ve_i^2}{2} + gz_i \right) - \sum \dot{m}_e \left(h_e + \frac{Ve_e^2}{2} + gz_e \right) \tag{3-1}$$

where, $\dot{Q}$ and $\dot{W}$ refer to the amount of heat and work exchanged between the control volume and the environment respectively. $\dot{m}$, h, V_e and z also represent the mass flow rate, enthalpy, velocity and potential term of the flows incoming or outgoing into/from the control volume. $E_{C.V}$ also refers to the total energy of the control volume including internal energy (U), kinetic energy (KE) and potential energy (PE). Note that the above equation subscripts i and e represent the inlet and outlet conditions respectively. In addition, the mass conservation law for the control volume (if applicable) can be written as Eq. 3-2:

$$m^{\lambda+1} = \left(\sum \dot{m}_i - \sum \dot{m}_e\right)^{\lambda} \Delta t + m^{\lambda} \tag{3-2}$$

where, m is the total mass of the control volume and the superscript λ counts the operational time steps of the control volume.

The first law efficiency, based on the definition, shows what portion of the inlet energy in the system has been effectively employed to produce power output by the control volume. Thus Eq. 3-3:

$$\eta_I = \frac{\text{Useful energy output}}{\text{Energy input}} \tag{3-3}$$

For thermodynamic modeling of a district heating, specifically, first, one should calculate the rate of heat loss and pressure drop along the pipeline. This would reveal at what temperature level the district heating water flow may reach the end-users, and how much work is required to drive the booster pumps for recovering the pressure drop along the way. Assuming a constant surface temperature for the pipe carrying the district heating water underground, Eq. 3-4 gives the temperature distribution profile of the water in the pipe [64]:

$$T(x) = (T_i - T_s)\, exp\left(\frac{UA_l x}{\dot{m}c}\right) + T_s \tag{3-4}$$

where, T is the temperature, and the subscripts i and s refer to the pipe inlet condition and the soil surrounding the pipe. UA_l is the overall heat transfer coefficient from the water within the pipe to the soil, $\dot{m}$ is the mass flow rate, c is the thermal capacity of water, and x is the length of the pipe. Calculating the temperature distribution profile, one may calculate the rate of heat loss from x meter of the pipe by Eq. 3-5:

$$\dot{Q}_l(x) = UA_l(x)(T - T_s);\ \boldsymbol{where}: UA_l(x) = \left(\frac{1}{h_{in}\pi d_{in}x} + \frac{ln\left(\frac{r_{out}}{r_{in}}\right)}{2k_p\pi x} + \frac{ln\left(\frac{r_{ins,out}}{r_{ins.in}}\right)}{2k_{ins}\pi x}\right)^{-1} \quad (3\text{-}5)$$

where, r_{out}, r_{in}, $r_{ins,out}$ and $r_{ins,in}$ refer to the external and internal radiuses of the pipe and the insulation, respectively. k_p and k_{ins} are the conductivity factors of the pipe and the insulation material. h_{in} is the convective heat transfer coefficient. This parameter for a turbulent and a laminar flows through a pipe with a constant surface temperature may be, respectively, given by Eq. 3-6 and Eq. 3-7 [64]:

$$h_{in-tur} = \frac{0.023Re_D^{0.8}Pr^{0.4}k}{d_{in}} \quad (3\text{-}6)$$

$$h_{in-lam} = \frac{3.66k}{d_{in}} \quad (3\text{-}7)$$

where, Re_D is Reynolds number, Pr is Prandtl number, k is the conductivity of the fluid and d_{in} is the internal diameter of the pipe. The rate of pressure drop through the pipe is given by Eq. 3-8:

$$\Delta P_t = \Delta P_f + \Delta P_{minor} = 1.2\ \Delta P_f;\ \textbf{where: } \Delta P_f(x) = \frac{fx\rho u^2}{2d_{in}} \quad (3\text{-}8)$$

in which, ΔP_f and ΔP_{minor} are, respectively, the major pressure losses due to friction and the minor pressure losses through the pipe (approximately equal to 20% of the friction losses). ρ represents water density and u is the flow velocity. *f* is Darcy friction factor which is a function of the pipe roughness and Reynolds number as Eq. 3-9:

$$f = \begin{cases} 0.316Re_D^{-0.25} & if \quad Re_D \leq 20000 \\ 0.184Re_D^{-0.2} & if \quad Re_D > 20000 \end{cases} \quad (3\text{-}9)$$

Having the value of the pressure losses, one may calculate the rate of work used by the booster pumps in the network as Eq. 3-10 [63]:

$$\dot{W}_p(x) = \dot{m}v\Delta P_t = \frac{0.6f\dot{m}xu^2}{d_{in}} \tag{3-10}$$

where, v is the specific volume of water.

In the previous chapter, one saw that the proposed district heating system takes advantage of decentralized heat storage units. The type of storage tanks employed in this system is a simple buffer stratified heat storage tank. Then, the next step is modeling of the storage tanks. Practically, a storage tank may operate with significant degrees of stratification, i.e. the top of the tank hotter than the bottom. In a multi-node tank simulation approach, a tank is modeled as divided into N nodes (sections), with energy balances written for each of the nodes. The result is a set of N differential equations that can be solved for the temperature of the nodes as a function of time. Figure 3-6 illustrates the arrangement of the nodes as well as the energy flows into/out of such a storage tank.

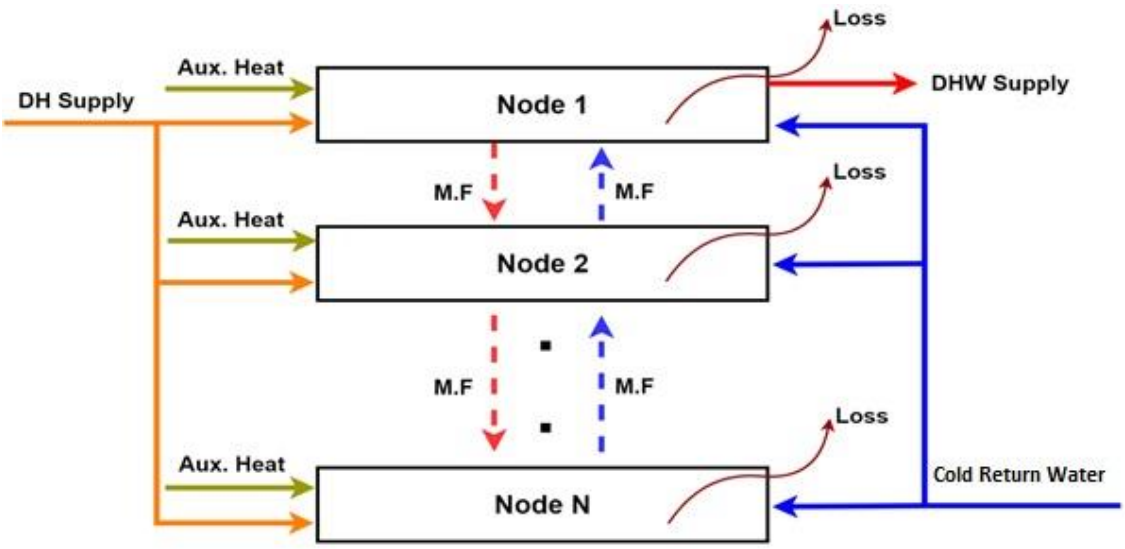

Figure 3-6 Schematic of a multi-node heat storage tank;
Aux. Heat: auxiliary heat source if any, M.F: mixing flows.

In this thesis, a five-node model is considered for modeling the neighborhood-scale storage tanks. For these tanks, the district heating water enters the tank from the top at the temperature of $T_{dh,s}$. The coming hot water flow lies in the first layer of the tank. Then, the same amount of water initially residing in the first node goes to the second node. Similarly, the water in each node moves to the lower node, and the same amount of water goes out to the district heating return line from the bottom node at $T_{dh,r}$. When discharging, the hot water supplied to the substations exits from the first node and it comes back to the tank through the bottom node. In addition to the energy transfer between the nodes due to mass transfer, the water layers exchange heat via thermal conduction due to the temperature differences.

Based on the first law of thermodynamics, the energy balance on each node of the tank can be written as Eq. 3-11 [65]:

$$\overbrace{m_j c \frac{dT_{st,j}}{dt}}^{\dot{Q}_{std}} = \overbrace{F_j^{dh} \dot{m}_{dh} c\left(T_{dh,s} - T_{st,j}\right)}^{\dot{Q}_{sp}} - \overbrace{UA_{l,j}\left(T_{st,j} - T_a\right)}^{\dot{Q}_l}$$

$$- \overbrace{F_j^{dhw} \dot{m}_{dhw} c\left(T_{st,j} - T_{dhw,r}\right)}^{\dot{Q}_{ld}} + \overbrace{kA_j \frac{\left(T_{st,j-1} + T_{st,j+1} - 2T_{st,j}\right)}{y_j}}^{\dot{Q}_{ex}} \tag{3-11}$$

$$+ \overbrace{\begin{cases} \dot{m}_{mix,j} c\left(T_{st,j-1} - T_{st,j}\right) & if \quad \dot{m}_{mix,j} > 0 \\ \dot{m}_{mix,j} c\left(T_{st,j} - T_{st,j+1}\right) & if \quad \dot{m}_{mix,j+1} < 0 \end{cases}}^{\dot{Q}_{mix}}$$

In this equation, the subscripts/superscripts j, dh, dhw, mix, st, l, s and r refer to the number of the node, district heating water flow, DHW flow, mixing flows between the nodes, the storage tank, heat losses, supply line and return line, respectively. Also, m, dt, A, y are, respectively, the mass of water in each node, the duration of each time step (10 s), the cross-section area of each node and the height of each node.

Besides, two important parameters in this equation are $\mathrm{F_j^{dh}}$ and $\mathrm{F_j^{dhw}}$. The value of these parameters is either 1 or 0. $\mathrm{F_j^{dh}}$ indicates which node the district heating water lies into, and the $\mathrm{F_j^{dhw}}$ reveals which node the DHW return flow resides in. During the high-temperature mode, according to the explanation given, $\mathrm{F_j^{dh}}$ is equal to 1 for the first node and 0 for the other nodes, while $\mathrm{F_j^{dhw}}$ should be 1 for the bottom node and 0 for the others. Overall, these two factors may be calculated by Eq. 3-12 and Eq. 3-13:

$$F_j^{dh} = \begin{cases} 1 & if \quad j = 1 \textbf{ and } T_{dh,s} > T_{st,j} \\ 1 & if \quad T_{st,j+1} < T_{dh,s} < T_{st,j-1} \\ 0 & if \qquad otherwise \end{cases} \tag{3-12}$$

$$F_j^{dhw} = \begin{cases} 1 & if \quad j = N \textbf{ and } T_{dhw,r} < T_{st,j} \\ 1 & if \quad T_{st,j+1} < T_{dhw,r} < T_{st,j-1} \\ 0 & if \qquad otherwise \end{cases} \tag{3-13}$$

As marked in Eq. (3-11), each of the statements on the left and right sides of the equation represent an energy flow in the control volume. Here, $\dot{Q}_{std}$, $\dot{Q}_{sp}$, $\dot{Q}_{l}$, $\dot{Q}_{ld}$, $\dot{Q}_{ex}$ and $\dot{Q}_{mix}$ refer to the rate of heat stored in each control volume, the injected heat into each node, the heat loss from each node, the heat load flow for DHW preparation, the heat exchanged between the nodes via conduction and the heat transferred among the nodes due to the mixing flows effect.

The above equation set can be solved by several numerical techniques such as the explicit Euler, the implicit Crank-Nicolson and Runge-Kutta methods. In this case, a fourth order Runge-Kutta method was employed [66].

Regarding the substations, as discussed, each substation includes two plate heat exchangers. Considering counter-flow plate heat exchangers and an effectiveness factor of 0.8, Eq. 3-14 may be applied [67]:

$$\dot{Q}_{hx} = \dot{Q}_{dhw} \text{ } or \text{ } \dot{Q}_{sh} = UA_{hx}\Delta T_{lm} \tag{3-14}$$

where, Q_{hx} is the heat transferred from the district heating water to the secondary side of the heat exchanger, and should be equal to the DHW demand (Q_{dhw}) or space heating demand (Q_{sh}) of the buildings. UA_{hx} is the overall heat transfer coefficient of the heat exchanger and ΔT_{lm} is the logarithmic mean temperature difference. The temperatures of the hot and cold flows outgoing from each heat exchanger are calculated by Eq. 3-15 and Eq. 3-16:

$$T_{max,e} = \frac{\varepsilon_{hx} \overbrace{\dot{m}_{min} c_{min} \left(T_{max,i} - T_{min,i}\right)}^{\dot{Q}_{max}}}{\dot{m}_{max} c_{max}} + T_{max,i} \tag{3-15}$$

$$T_{min,e} = T_{max,i}\varepsilon_{hx} + (1 - \varepsilon_{hx})T_{min,i} \tag{3-16}$$

in which, the subscript min refers to the fluid with the lower value of $C = \dot{m}c$ and the superscript max refers to the other fluid. In this equation, $\dot{Q}_{max}$ is the maximum possible heat transfer rate through the heat exchanger and ε_{hx} is the heat exchanger effectiveness factor given by Eq. 3-17 [67]:

$$\varepsilon_{hx} = \frac{1-exp[-(UA_{hx}/C_{min})(1-C_r)]}{1-C_r exp[-(UA_{hx}/C_{min})(1-C_r)]} \text{ ; Where } C_r = \frac{C_{min}}{C_{max}} \quad (3\text{-}17)$$

The heat demand in the DHW section depends on the hot water draw-off the building. This includes the small hand-washing tapings, the draw-offs for taking a shower, dishwashing, etc. and cannot be calculated or estimated. Instead, there are some standard taping profiles for families with different sizes that can be used as sample draw-off patterns. The energy demand for space heating, on the other hand, depends on the local standard comfort temperature, the ambient temperature, the building stuck energy performance, the performance of the heat distribution facilities and personal preferences. Eq. 3-18 calculates the rate of demand for space heating:

$$\dot{Q}_{sh,b} = \overbrace{\rho_a V_{a,b}\left({T_{in}}^{t+1} - {T_{in}}^{t}\right)}^{\dot{Q}_{s,b}} + \overbrace{\rho_a \dot{V}_{ven}\left(T_{in} - T_{out}\right) + UA_{l,b}(T_{in} - T_{out})}^{\dot{Q}_{l,b}} + \overbrace{\rho_{bm} V_{bm}\left({T_{bm}}^{t+1} - {T_{bm}}^{t}\right)}^{\dot{Q}_{std,b}} - \overbrace{\sum_{n=1}^{M}\left(A_n \mathrm{I_T}(\tau\alpha)_{avg}\right)}^{\dot{Q}_{g,b}} \quad (3\text{-}18)$$

in which, the subscripts a, in, out, b and b_m refer, respectively, to the air within the building, the indoor condition, the ambient condition, the building and the building stuck material. The superscript t is the time step counter (in seconds), and V represents the volume. As marked, each of the statements on the right side of the equation represents an energy flow coming in or going out of the building. The first term on the right is the rate of heat required to increase the indoor temperature. The summation of the second and the third terms calculate the total rate of heat losses from the building via ventilation and heat loss to the ambient. In these two terms, $\dot{V}_{ven}$ is the volume flow rate of air replaced by fresh air for ventilation, and $UA_{l,b}$ is the overall heat loss coefficient of the building through the walls and windows. The fourth term indicates the rate of energy stored in the building stuck which will be zero in steady state conditions, and the last item calculates the rate of heat gain of the building due to solar irradiation. In this item, I_T is the solar irradiation through the windows, A_n is the area of each of the windows and $(\tau\alpha)_{avg}$ refers to the average transmission-absorption coefficient of the windows and the internal elements of the building exposed to the sun rays. M is the total number of apertures letting sun rays coming into the building.

Calculating the energy demand of the network and the temperature drop through the pipe, one could size the pipeline of the district heating system. This includes dimensioning of the main distribution pipes, the street branches and house-connection pipes. For this, information about the supply and return temperatures are required. Naturally, the pipeline is such sized that it may carry the maximum heat load of the network based on the maximum allowed velocity through the pipes, i.e. 2 m/s. Eq. 3-19 is used for sizing the pipeline:

$$d_{n,p} = \left(\frac{4\dot{Q}_{max}}{\rho\pi u_{max} c\Delta T}\right)^{0.5} \tag{3-19}$$

where, $d_{n,p}$ is the nominal diameter of each section of the pipeline, $\dot{Q}_{max}$ is the maximum heat load of the given pipeline section, u_{max} is the maximum velocity of water in the pipe, and ΔT is the temperature difference between the return and supply lines.

It bears mentioning that the simulation of the performance of the system via the above presented mathematical model is accomplished via programming in MATLAB.

3.2.2 Twin Pipe Thermo-Hydraulic Model

As it was mentioned, in addition to the thermodynamic modeling of the district heating system, with the aim of assessing the feasibility of employing the existing pipe technology for the proposed district heating system, this study aims at a thorough thermal and hydraulic performance simulation of twin-pipes. Therefore, in this section, the governing equations on the thermal and hydraulic behavior of the pipes and the heat carrier (the solid and the fluid sides) are presented. Then, the numerical solution method is explained in details.

Governing equations: The governing equations on the motion of an incompressible flow are Reynold-Averaged Navier Stockes, which can be written in Eq. 3-20 and Eq. 3-21 in a Cartesian coordinate [68]:

$$\frac{\partial \rho u_i}{\partial x_i} = 0 \tag{3-20}$$

$$\frac{\partial u_i}{\partial t} + \frac{\partial u_i u_j}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\upsilon\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l}\right)\right] + \frac{\partial y}{\partial x}\left(-\overline{u_i' u_j'}\right) \tag{3-21}$$

in which, u_i, u_j, and u_l are the velocity components in the directions on i, j, l respectively. x_i, x, and x_l are the cartesian coordinates in the directions on i, j, l. δij is the Kronnoecker delta function, υ and μ are the kinematics and dynamic viscosity terms and ρ is density. The Reynolds stresses term $(-\overline{u_i' u_j'})$ is evaluated as Eq. 3-22:

$$-\overline{u_i' u_j'} = \upsilon_t\left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right) - \frac{2}{3}k\delta_{ij} \tag{3-22}$$

where, v_t is turbulent viscosity. For considering the effect of turbulency the Realizable k-ε model is used in this work. This model is based on the solution of the two heat transfer equations of turbulent kinetic energy (k) and dissipation of turbulence energy (ε) as Eq. 3-23 and Eq. 3-24 [69]:

$$\frac{\partial k}{\partial t} + \frac{\partial k u_j}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\upsilon + \frac{\upsilon_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \varepsilon \tag{3-23}$$

$$\frac{\partial \varepsilon}{\partial t} + \frac{\partial \varepsilon u_j}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\upsilon + \frac{\upsilon_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1 S\varepsilon + C_2\frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}} \tag{3-24}$$

Where Eq. 3-25:

$$C_1 = \max\left[0.43, \frac{\eta}{\eta + 5}\right], \eta = S\frac{k}{\varepsilon}, S = \sqrt{2S_{ij}S_{ij}} \tag{3-25}$$

In these equations, C_2 is a constant, σ_k and σ_ε are turbulent Prandtle numbers for k and ε. Also, G_k and G_b are the productions of turbulence kinetic energy due to the average velocity gradients and buoyancy, which can be given by Eq. 3-26 and Eq. 3-27, respectively.

$$G_k = -\overline{u_i' u'_j}\frac{\partial u_j}{\partial x_j} = \upsilon_t S^2 \tag{3-26}$$

$$G_b = \beta g_i \frac{\nu_t}{Pr_t}\frac{\partial T}{\partial x_i} \tag{3-27}$$

where β and g_i are coefficient of thermal expansion and the gravity acceleration component in each direction. Pr_t is the turbulence Prandtl number for energy considered as 0.85 here [68]. S is the modulus of the mean rate-of-strain tensor, defined as Eq. 28 and Eq. 3-29:

$$S \equiv \sqrt{2S_{ij}S_{ij}} \tag{3-28}$$

$$S_{ij} = \frac{1}{2}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) \tag{3-29}$$

When the terms k and ε are calculated, the turbulent viscosity may be calculated by Eq. 3-30, Eq. 3-31, Eq. 3-32, Eq. 3-33, and Eq. 3-34:

$$\upsilon_t = C_\mu \frac{k^2}{\varepsilon} \tag{3-30}$$

$$C_\mu = \frac{1}{A_0 + A_s \frac{kU^*}{\varepsilon}} \tag{3-31}$$

$$U^* \equiv \sqrt{S_{ij}S_{ij} + \widetilde{\Omega_{ij}}\widetilde{\Omega_{ij}}} \tag{3-32}$$

$$\widetilde{\Omega_{ij}} = \Omega_{ij} - 2\varepsilon_{ijk}\omega_k \tag{3-33}$$

$$\Omega_{ij} = \overline{\Omega_{ij}} - \varepsilon_{ijk}\omega_k \tag{3-34}$$

where, $\overline{\Omega_{ij}}$ is the mean rate of rotation observed in the moving reference frame with the angular velocity of ω_k. The rest of the parameters in the above formulation are given as Eq. 3-35, Eq. 3-36, and Eq. 3-37:

$$A_0 = 4.04, A_s = \sqrt{6}\cos\phi \tag{3-35}$$

$$\phi = \frac{1}{3}\cos^{-1}(\sqrt{6}W), W = \frac{S_{ij}S_{jk}S_{ki}}{\tilde{S}^3}, \tilde{S} = \sqrt{S_{ij}S_{ij}}, S_{ij} = \frac{1}{2}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) \tag{3-36}$$

$$C_1 = 1.44, C_2 = 1.9, \sigma_k = 1.0, \sigma_\varepsilon = 1.2 \tag{3-37}$$

The energy equation is used for calculating the rate of heat transfer as Eq. 3-38:

$$\frac{\partial \rho E}{\partial t} + \frac{\partial}{\partial x_i}[u_i(\rho E + p)] = \frac{\partial}{\partial x_j}\left(k_{eff}\frac{\partial T}{\partial x_j}\right) \tag{3-38}$$

in which, E is the total energy, p is pressure, T is temperature and k_{eff} is the effective thermal conductivity term given by Eq. 3-39:

$$k_{eff} = k + \frac{c_p \mu_t}{Pr_t} \tag{3-39}$$

where, c_p is the specific thermal capacity.

Then, the energy flow for the solid side of the problem (the pipe body) can be written as Eq. 3-40:

$$\frac{\partial}{\partial t}(\rho h) + \nabla.(\vec{v}\rho h) = \nabla.(k\nabla T) + S_h \tag{3-40}$$

where, h is sensible enthalpy and the second term on the left-hand side represents convective energy transfer due to rotational or translational motion of the solids. The terms on the right-hand side are the heat flux due to conduction and volumetric heat sources within the solid (S_h), respectively.

The value of heat transfer between the wall surface and the fluid for a unit area of the wall is calculated as Eq. 3-41:

$$q = h(T_f - T_w) \tag{3-41}$$

in which, h is the local convective heat transfer coefficient, T_f is the local fluid temperature and T_w is the local wall temperature. The heat transfer rate for the boundary of a solid cell of the pipe is calculated as Eq. 3-42:

$$q = \frac{k_s}{\Delta n}(T_w - T_s) \tag{3-42}$$

where, T_S is the thermal conductivity of the solid cell, Ts is the local temperature of the solid cell, and Δn is the distance of the center of the solid cell and the wall surface.

<u>Solution method and boundary conditions:</u> For simulation of the heat transfer process in this work, first, the computational space is discretized by octagonal 3-D elements in a non-uniform grid. According to the importance of the heat distribution along the radius of the pipe, the meshing has been done with very small elements. Figure 3-7 shows the mesh grid made on a cross section of the pipe.

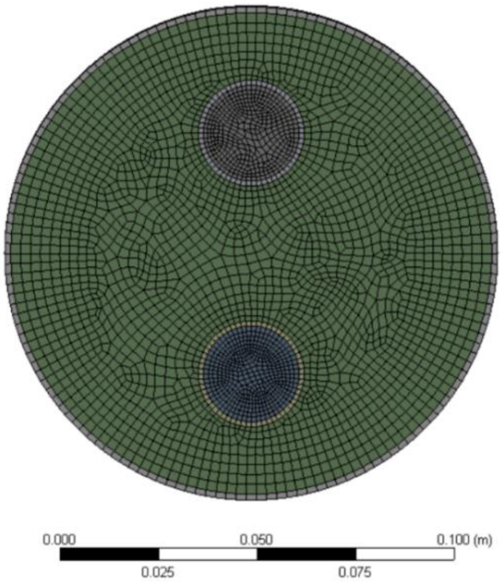

Figure 3-7 The mesh grid on the pipe cross section.

The boundary condition of the uniform flow for the inlets, and the boundary condition of static pressure for the outlets are used. The no-slip boundary condition for the walls is also considered here. Note that for the simulation of the flow behavior close to the wall, the standard wall function is used which is broadly used for industrial flows.

Due to the large geometry of the problem and the high cost of the computations, in this work, continuity and momentum equations along with turbulence and energy equations have specifically been solved for the computation of hydraulic parameters for a small length of the problem. For the rest of the computations, energy and continuity equations are used only. Since booster pumps are located on the transmission pipelines to compensate the pressure losses, the velocity of the fluid within the pipes has been considered to be constant.

This assumption eliminates the need for the momentum equation, and makes the effects of the turbulent fluctuations and secondary flows negligible.

The simulations are accomplished based on the finite volume method within the computational fluid dynamic model of ANSYS FLUENT.

3.3 The Case Study

This section presents information about the case study of the project as well as the characteristics considered for the twin-pipes for being simulated numerically.

3.3.1 The Location and Design of the Network

In this thesis, Urupema city in Santa Catarina state is chosen as the case study of the project in which the district heating system is going to be assumed to be developed. This small municipality is one of the coldest cities in Brazil which is located in the south region. Also, it was the coldest city of the country in 2013 by -6.3 ºC record. In fact, there is another motivation to select this city as a small one since this project is the start of such project in Brazil. Moreover, some projects such as a 78.2 MW wind farm have been evaluated to hire wind power and solar energy in this city. Therefore, these renewable energies can be considered as near-future sources to supply heat for the current work and it is another reason to choose this city [75, 76]. Figure 3-8 presents two views of the city.

Figure 3-8 Two views of the city of Urupema

This town has estimated 2487 habitants and around 353 km^2 area. Consequently, the demographic density is approximately 7 habitants/km^2. However, only 0.6 km^2 of this area is urban area and consecutively, the demographic density in urban area can be expressed as 2166 habitants/km^2 since near to 1300 people live in urban area. Altitude and pasture fields consists around 68% of area of the municipality, i.e. 240 km^2. Native forest predominance has an area around 99 km^2 and 28% of total area of city. In addition, 7.5 km^2 or 2% of municipal area belongs to agriculture and orchards [76, 77]. Figure 3-9 gives a helicopter view of the city.

Figure 3-9 The satellite map of Urupema

The altitude of the town is 1335 meters from the sea level. The town is located in a small valley surrounded by elevations of Morro das Torres hill with altitude of 1726 m. Although, the urban area is located between 1315 m near the CTG Nordeste Urupemense sports club, up to 1430 m on the road that it joins Rio Rufino region to the North. In the south entrance, the houses are in levels of 1350-60 m. The average altitude of Urupema is 1425 meters [77].

The altitude and geographical position in the south Santa Catarina plateau as it can be seen in Figure 3-10 cause that temperatures can exceed -10 °C in winter. The occurrence of snow in a few days of the year is common, with average of 3 to 5 days per year and fast and little in general. Strong snow with accumulation over 5-10 cm is occasional. The annual average temperature is near to 14 ºC and the annual average precipitation is around 1800 mm. However, temperatures can go below freezing between April and September.

Furthermore, absolute minimum temperatures are below 5 °C during all months of the year. The frost appears all year round most commonly between April and November. There was already a record of -1 °C in January 1994. Also, Morro das Torres hill is the best place to see the icicle in Brazil [75]. A landscape of this town in two different normal and snowy days is shown in Figure 3-11.

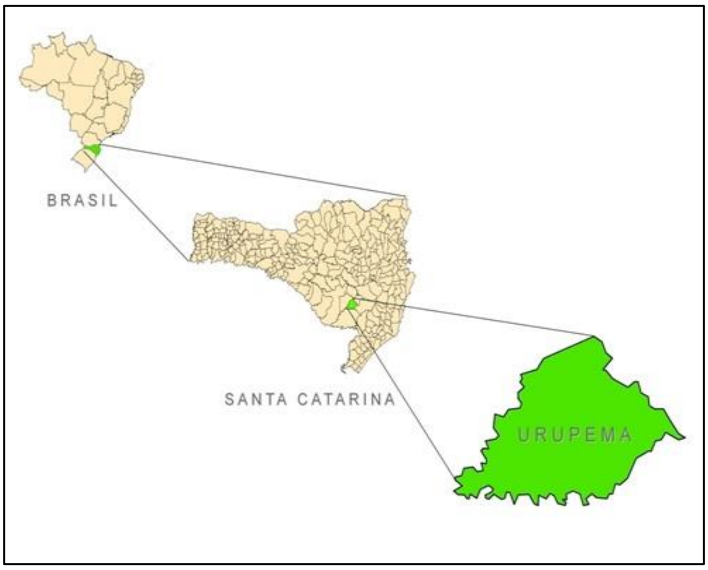

Figure 3-10 Geographical position of Urupema.

Figure 3-11 Snowy day in Urupema town

Together with São Joaquim and Bom Jardim da Serra in the same state, São José dos Ausentes in the neighboring state of Rio Grande do Sul, which are the coldest in the south, negative temperatures can occur from 35 to 45 days per year from March to November. Urupema has the lowest absolute minimum temperatures among them. Frost or snow happens from April to October on colder days. The old meteorological station of Urupema that currently is deactivated was one of the highest in Santa Catarina with a height of 1,419 m about 1 km north of the city's headquarters. Due to it is located below Morro das Torres hill in the preferred direction of the cold air flow and being almost 100 meters higher than the municipal seat, the meteorological station often showed smaller maximums than the center of Urupema. However, the lowest temperatures in Brazil in strong cold weather occur especially in the city where Epagri's weather station was installed in 2010. The lowest temperature recorded so far this season is -8.8 °C in the city and -20 to -30 °C at Morro das Torres in June 28th of 2011. Climatological data of Urupema from 2010 can be seen in Table 3-2 and Figure 3-11 [75].

Table 3-2 Climatological data of Urupema (Temperatures are in Celsius degree)

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year
Abs. Max. Temp.	31.1	32.3	28.5	27.5	25.2	23.5	25.7	27.8	28.8	27.5	28.3	30.5	32.3
Ave. Max. Temp.	24.5	24.5	22.1	20.0	16.2	14.9	15.2	17.3	18.0	19.6	21.6	23.5	19.8
Ave. Temp.	18.1	18.3	15.8	13.5	10.0	8.2	8.8	10.4	11.6	13.4	15.1	17.3	13.4
Ave. Min. Temp.	12.8	13.5	10.7	7.9	4.6	2.4	3.3	4.4	5.9	8.1	9.5	12.0	7.9
Abs. Min. Temp.	2.3	1.5	-1.7	-4.4	-6.8	-8.8	-7.5	-8.3	-6.3	-4.7	-1.8	0.1	-8.8

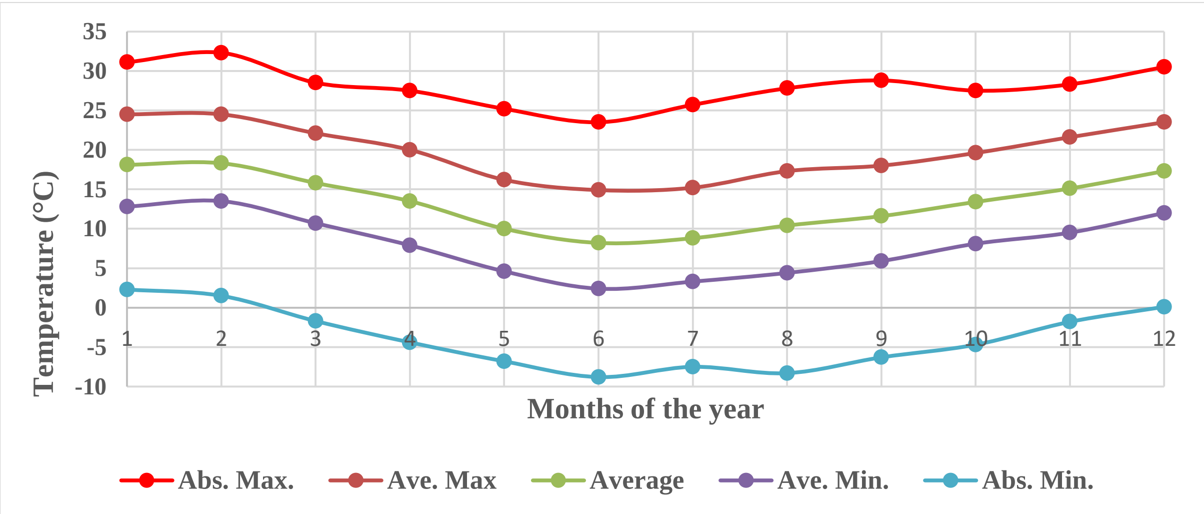

Figure 3-12 Climatological chart of Urupema city

The fairly cold climate along with the high share of renewable sources, such as solar thermal and electricity system, wind energy, etc., make this point an appropriate location to host the smart district heating system designs. A total of 100 detached houses in the medium-sized category (with 2-4 inhabitants) are considered to be covered by a district heating system.

Generally, it is very challenging to find a database of heat consumption, and this gets even more difficult when information by category of consumption, i.e. space heating and DHW use, is required. The heat databases include the hourly heat consumption information need to be refined to estimate what portion of this is used for space heating and what portion of the DHW demand coverage. In addition, such databases present the aggregated data instead of an individual building consumption. Therefore, it is not possible to drive an exact heat consumption profile for the buildings of the network. That is why, in this work, the heat consumption profiles of the individual buildings are created by calculating their space heating demand and randomization of a standard DHW draw-off pattern. This will be discussed comprehensively in the results section. In order to simplify the topology of the case study, the network shown in Figure 3-12 is considered to be supplied by district heating.

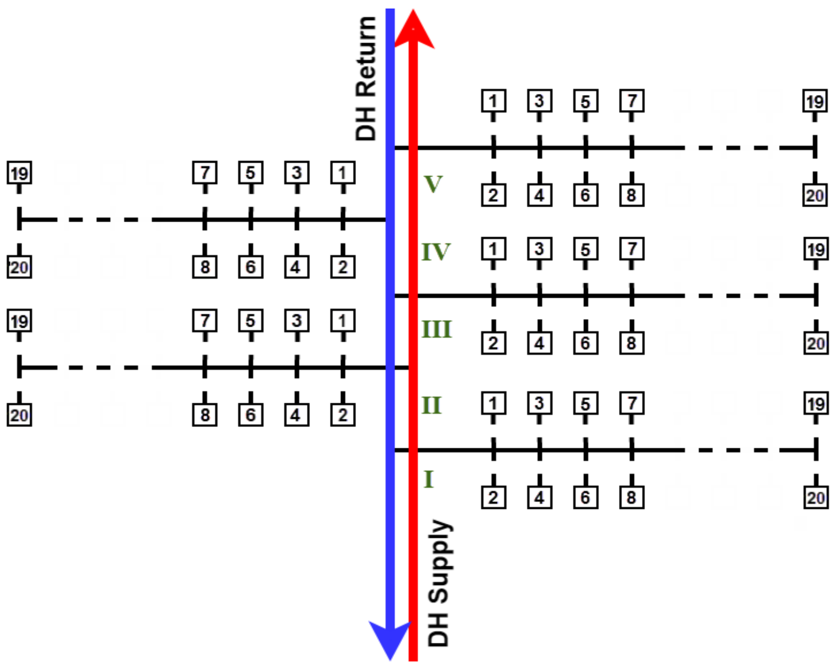

Figure 3-13 Topology of the buildings in the network.

3.3.2 Twin-Pipe Characteristics

According to the Figure 3-12, the network includes 5 streets, each with 20 detached-houses. The distance between the streets is 100 m, the distance of the houses in each street is 20 m and the houses are 10 m far from the street. The living area of each building is 150 m^2 on a 15 m × 10 m field.

Twin-pipes came into service when district heating technology jumped into its third generation design from the second generation concept. This concept comprised the extremely lower temperatures of supply and return, and a fundamental revision in the components of the system, including the pipeline structure [45]. Figure 3-13 shows the schematic of a piece of a twin-pipe with its various components marked on it. As Figure 3-13 shows, in a twin-pipe, both of the supply and return lines come into a single outer casing while an efficient insulation (usually polyurethane foam - PUR) prevents them exchanging heat to each other and the ambient.

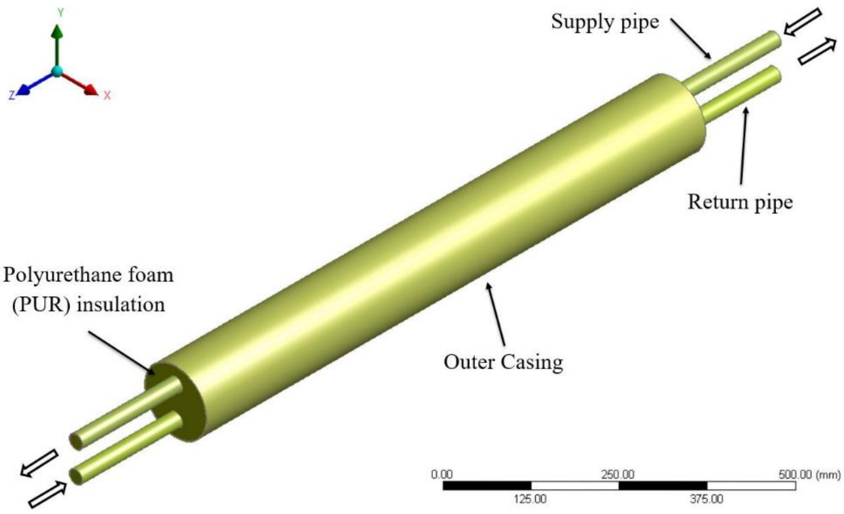

Figure 3-14 The schematic of a piece of a twin-pipe.

In this work, a typical twin-pipe is considered to fulfill the investigations. Table 3-3 presents information about the dimensions of different typical twin-pipes usually seen in various sections of a district heating pipeline including the transmission part, the street branch pipes, and the house connection pipes. As the focus of this study is on the transmission section of the pipeline, pipe type 5 is the one investigated in this work.

Table 3-3 Dimensions of various twin-pipes investigated [58].

	Pipe			**Casing**	
	Nominal Diameter (mm)	**Outer Diameter (mm)**	**Thickness (mm)**	**Outer Diameter (mm)**	**Thickness (mm)**
Type 1	20	26.9	2.6	125	3
Type 2	20	26.9	2.6	140	3
Type 3	65	76.1	2.9	225	3.5
Type 4	65	76.1	2.9	250	3.5
Type 5	**200**	**219.1**	**4.5**	**560**	**6**
Type 6	200	219.1	4.5	630	6.6

Table 3-4 presents further information about the characteristics of the pipes.

Table 3-4 Characteristics of twin-pipes investigated and district heating flow through the pipes [24].

Component	**Material**	**Density (kg/m^3)**	**Specific Heat (J/kg.K)**	**Thermal Conductivity (W/m.K)**	**Viscosity (kg/m.s)**
Heat Carrier	Water	982	4,136.5	0.65	0.001
Insulation	PUR	30	133	0.021	-
Pipe	Steel	8,030	502.5	16.27	-

Note that the thermal conductivity and density of water may change as the temperature varies; however, this change is so small that it is considered negligible in this study.

Table 3-5 presents information about the three different district heating systems under evaluation in this work.

Table 3-5 Characteristics of the three district heating schemes.

District Heating Type	**Supply Temperature (^oC)**	**Return Temperature (^oC)**	**Pressure (MPa)**
ULTDH	45 (all the time)	25 (all the time)	1.2
LTDH	55 (all the time)	30 (all the time)	1.2
Variable	70 (4 h a day) / 45 (20 h a day)	35 (4 h a day) / 25 (20 h a day)	1.2

4 Results and Discussions

In this section, the results of the simulations on the various configurations of district heating systems including the NUTDH system are discussed.

4.1 Results

For accomplishing the simulations, first, one needs to create rational demand profiles for the network. For DHW draw-off profile, the following Danish standard hot water draw-off pattern (detailed in Table 4-1), which is recommended for a medium-sized family, is used [79]. According to the table, there are several hand washing draw-offs (A) with a flow rate of 3 lit/min each, two shower draw-offs a day (B) with the flow rate of 6 lit/min each, and two further draw-offs with the flow rate of 4 lit/min (C and D).

Table 4-1 Standard DHW tapping pattern for a medium-sized dwelling

Time	**min**	00	05	30	01	15	30	45	00	30	30	30	45	45	30	30	30	00	15	30	00	30	15	30
	hr	7	7	7	8	8	8	8	9	9	10	11	11	12	14	15	16	18	18	18	19	20	21	21
Draw-off Type		A	B	A	A	A	A	A	A	A	A	A	A	C	A	A	A	A	A	A	A	D	A	B
Information		A: 0.105 kWh; 3 lit/min												B: 1.4 kWh; 6 lit/min										
		C: 0.315 kWh; 4 lit/min												D: 0.735 kWh; 4 lit/min										

The total daily draw-off of this pattern is 100 lit for each building. As this profile seems lighter than real-life water tapping, a correction factor of 25% is used to weight up the loads in each category. For making the DHW profile of the whole network, this profile is randomized for all of the buildings. This is simply done in MATLAB. Figure 4-1 shows the resultant DHW demand of the whole network over a sample day. In practice, a larger draw-off trend should be observed during early-morning and early-evening hours. Although this trend is not observed in Figure 4-1, the logical total daily tapping value and the fluctuating trend of the profile make the pattern an appropriate reference for the simulations.

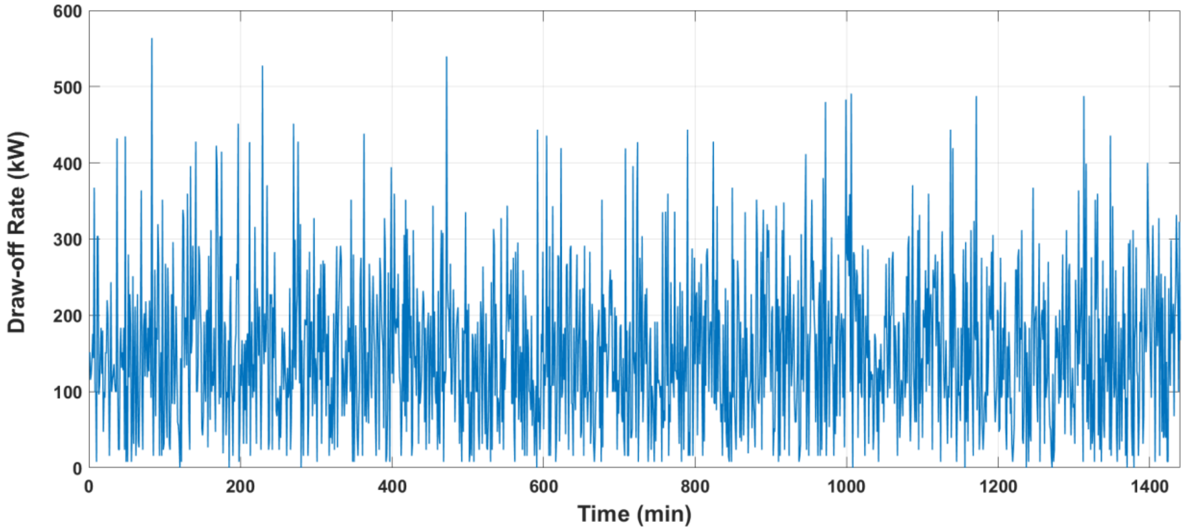

Figure 4-1 DHW draw-off profile of the entire network during a day.

The space heating demand of the network is calculated by considering several factors, such as the ambient temperature, the standard comfort temperature, the buildings stuck thermal performances, etc. Figure 4-2 illustrates the ambient temperature of Urupema in a monthly averaged format over the entire year 2015 [70].

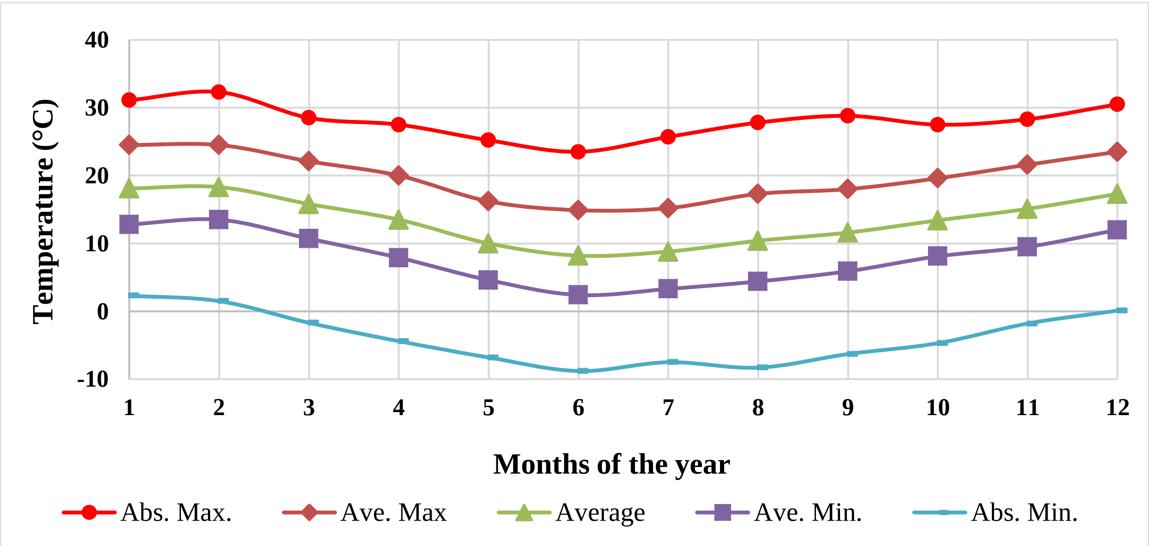

Figure 4-2 Average ambient temperature of Urupema city in 2015.

As an important factor on the space heating demand of the network, the solar irradiation in this area is also needed. Figure 4-3 presents information about the minutely measured solar energy irradiated on a horizontal surface with 1 m^2 area in Urupema during 2015 [70].

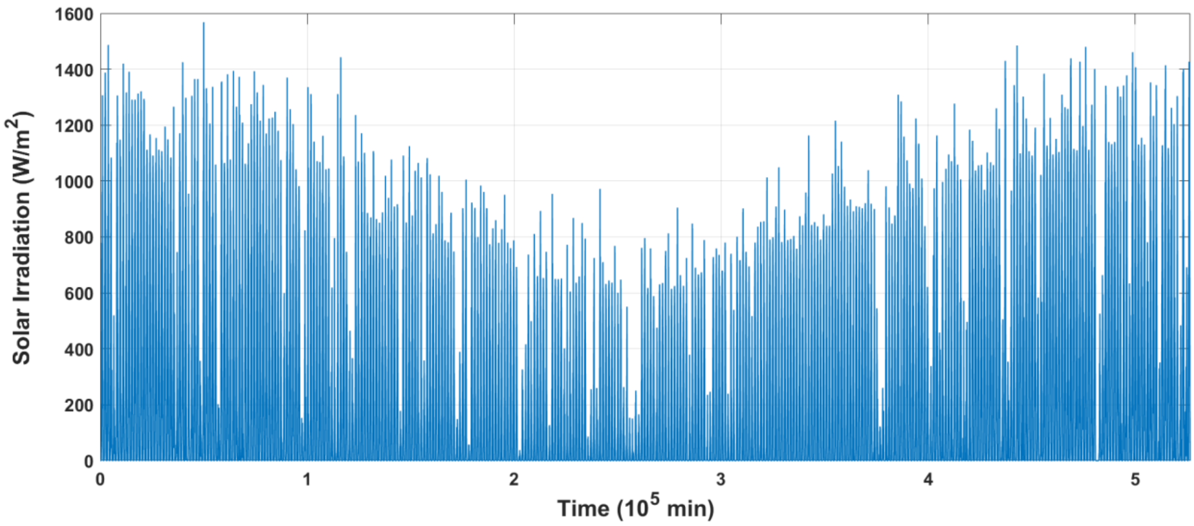

Figure 4-3 Solar irradiation availability in the case study during 2015.

For the buildings of the network, according to [71], an average overall heat loss factor of 0.93 W/m².K is considered. This is based on 2.32 W/m².K for windows (20% of the building shell surface), 0.625 W/m².K for walls (30%), 0.5 W/m².K for roofs (25%) and 0.625 W/m².K for the floor including cold bridges (25%). As all the buildings in the network are considered to be residential, the average hourly air exchange rate for ventilation is 0.5 m³/h per m³ of the heated building volume. The average daily internal gain of the building due to the inhabitants' metabolism effects is 2.3 °C [71].

In Brazil, the standard NR 17-Ergonomics from 1990 defines the acceptable thermal comfort conditions by defining the limits of effective temperature between 20 and 23 °C, air velocity is set to be less than 0.75 m/s and humidity should be above 40% [72]. In this work, the desired indoor temperature is set on 22 °C. Having said all these, one may calculate the space heating demand of each of the buildings, and subsequently, the entire network. Figure 4-4 presents information about the heating demand of each of the buildings over the year. Clearly, as the comfort temperature and the characteristics of the buildings are considered the same, similar heat demand profile is obtained for all of the end-users in the network.

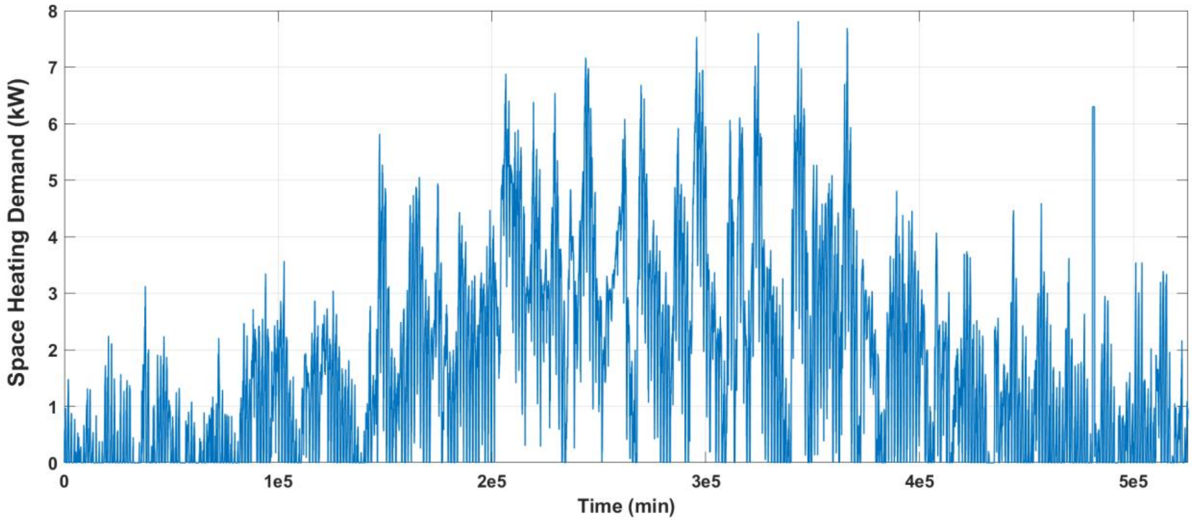

Figure 4-4 Space heating demand of each building in the network over the year.

Having the randomized draw-off profile of the network and the space heating demand of the buildings, one may calculate the instantaneous total demand (space heating + hot water) of the entire network for each of the considered district heating schemes. Figure 4-5 shows how different designs could result in the change of the district heating load. Figure 4-5 is plotted for three consecutive days of summer and three days in winter. The data is not presented for the entire year because this makes the graphs unreadable.

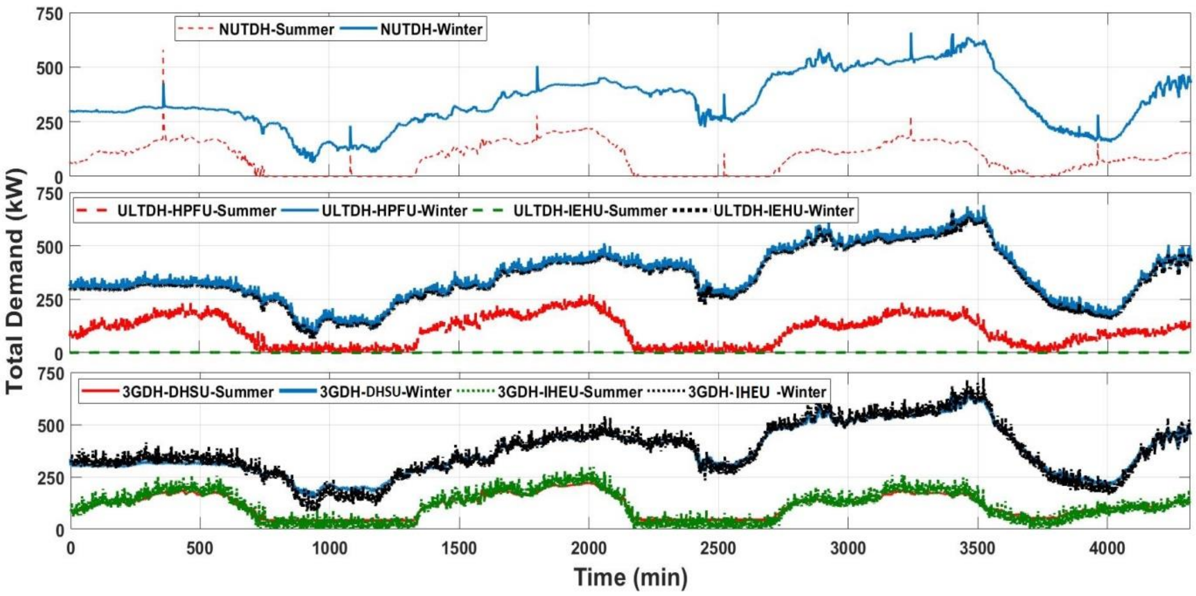

Figure 4-5 The heat demand of various district heating systems for the case study over sample days.

According to the Figure 4-5, the NUTDH system has a peak demand twice a day, i.e. when the heat storage units are charged. This decreases the load considerably during the rest of the day, even to a zero during the hot season. Between the two schemes of the ULTDH system, it is well observed how the IEHU would lead to a lower load because this system is for space heating only.

The main advantage of the heat pumps is enabling the ULTDH system to cover the DHW demand of the buildings, though it makes the substation be dependent on the electricity of the building. The heating duty of the ULTDH-HPFU system is calculated as Eq. 4-1:

$$\dot{Q}_{ult-hpf} = \dot{Q}_{sh} + \dot{Q}'_{dhw};\ \boldsymbol{where}\colon \dot{Q}'_{dhw} = \dot{Q}_{dhw} - \dot{Q}_{cond-hp} \qquad (4\text{-}1)$$

in which, $\dot{Q}_{ult-hpf}$ is the total heat load of a building when employing a ULTDH-HPFU scheme, $\dot{Q}'_{dhw}$ is the share of the district heating system in DHW preparation of the building, $\dot{Q}_{cond-hp}$ is the heat provided via the condenser of the heat pump for DHW supply. The electricity required to derive the heat pump is calculated by Eq. 4-2:

$$\dot{E}_{hp} = \frac{\dot{Q}_{cond-hp}}{\beta_{hp}} = \frac{\dot{Q}_{dhw} - \dot{Q}'_{dhw}}{\beta_{hp}} \qquad (4\text{-}2)$$

where, β_{hp} is the coefficient of performance of the heat pump, assumed to be equal to 3 in this work.

Regarding the 3GDH scheme, it can be seen how employing a local storage tank can help for peak shaving in the network, decreasing the design capacity of the network. This peak shaving effect can be much more sensible if the draw-off is based on a large-scale family rather than the medium-sized families considered in this work.

Having the profiles of the instantaneous heat demands, and taking into account the design supply and return temperatures, one could see the differences between the flow rates of district heating water through the main distribution pipelines in each of the five different schemes. Figure 4-6 gives information about this parameter. According to Figure 4-6, the flow rate in the 3GDH system is considerably lower than the other two designs. This is mainly because of the larger temperature difference between the supply and return lines in this scheme. In the ULTDH system, expectedly, the flow rate for the IEHU is smaller because it is only responsible for space heating.

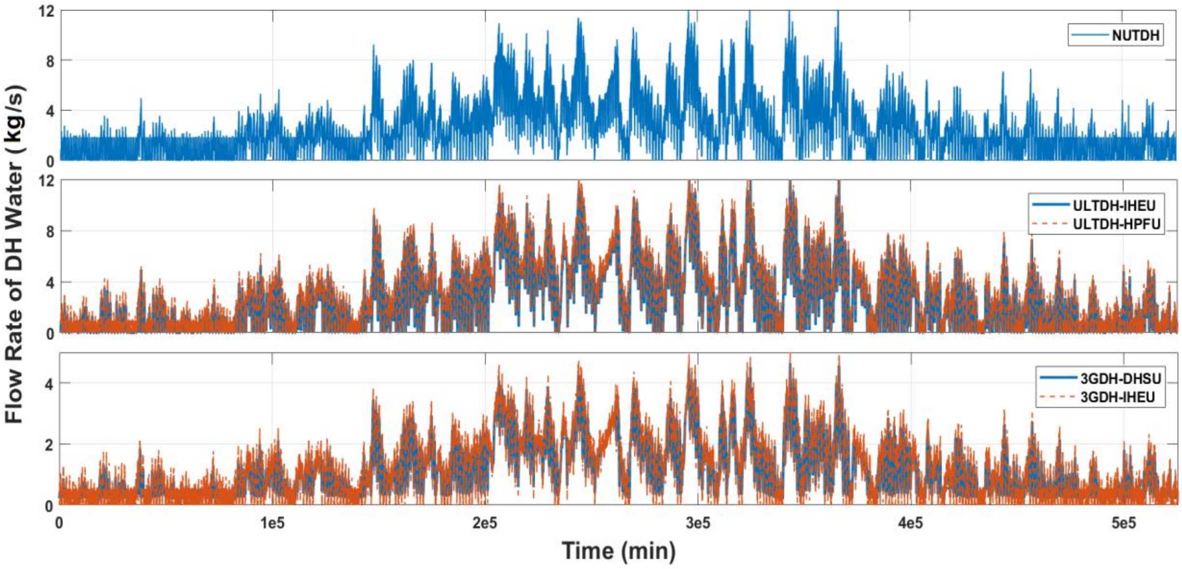

Figure 4-6 Flow rate of district heating water through the pipeline of each of the considered cases.

Having the demands and flow rate profiles, the first step for analyzing the district heating systems is sizing the pipeline of each of the considered schemes. Table 4-2 presents detailed information about the size of pipes in various cases for the case study. It should be mentioned that for both of the 3GDH-DHSU and the ULTDH-HPFU systems small house connection pipes are obtained, however, these values are adjusted to 10 mm as the smallest possible house connection pipes. Note that the obtained dimensions for each of the systems may be different from the realistic systems and the main reason for this is the lack of realistic data for the DHW draw-off of the network.

Table 4-2 Size of the pipes for each of the considered systems.

	3GDH		ULTDH		NUTDH	
	DHSU	IHEU	IEHU	HPFU	DHW Line	SH Line
House Connection (Ø, mm)	10	11	10	15	10	10
Street Branch, 0-200 m (Ø, mm)	24	27	40	42	17	40
Street Branch, 201-400 m (Ø, mm)	17	21	28	31	14	28
Main Pipe, Part I (Ø, mm)	54	56	89	91	60	
Main Pipe, Part II (Ø, mm)	49	51	79	81	54	
Main Pipe, Part III (Ø, mm)	42	44	69	71	46	
Main Pipe, Part IV (Ø, mm)	34	37	56	58	38	
Main Pipe, Part V (Ø, mm)	24	26	40	41	27	

Results and Discussions

As seen, a 3GDH system with a DHSU substation configuration results in the smallest pipe dimensions. The main reason for this is the peak shaving that the storage tanks provide. On the other hand, the ULTDH system, with both substation types, requires the largest dimensions of the pipes. This is due to the small temperature difference of the district heating medium in the supply and return lines, i.e. only 15 ^{o}C. The NUTDH system needs the second largest pipes in the network because of the sharp sudden increase in the heat load of the system when the high-temperature supply starts.

Having the dimensions of the pipelines and the flow rate of the water through the pipelines, one could make a comparison between the levels of pressure drop in each of the five scenarios. Naturally, when the pressure drop value is known, the amount of work required for the booster pumps to compensate the pressure losses may be calculated. Figure 4-7 presents information about the instantaneous rate of pressure loss over the whole network for the various cases. The pressure drop profiles are only presented for 10 days in February as the hottest and 10 days of July as the coldest periods of the year. As seen, both of the 3GDH schemes show the lowest rates of pressure drop with a maximum rate of 1.7 MPa, because of the smaller flow rate of water through the pipeline. The NUTDH presents the highest rate of pressure drop with a peak value of 3.9 MPa.

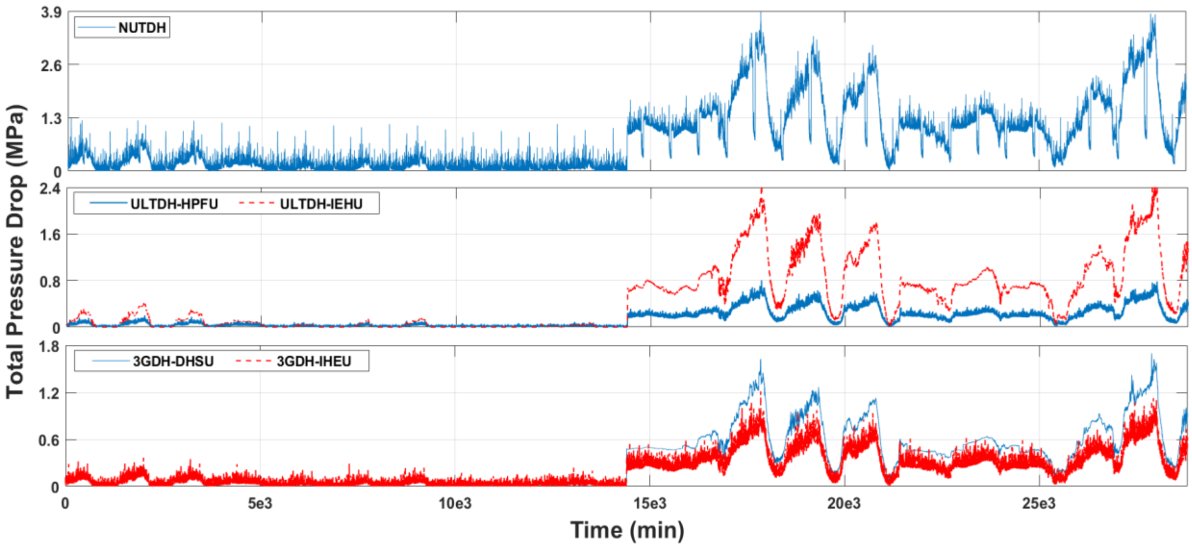

Figure 4-7 Pressure loss rate in the entire pipeline for various cases.

Figure 4-8 shows the rate of heat losses in the five considered scenarios.

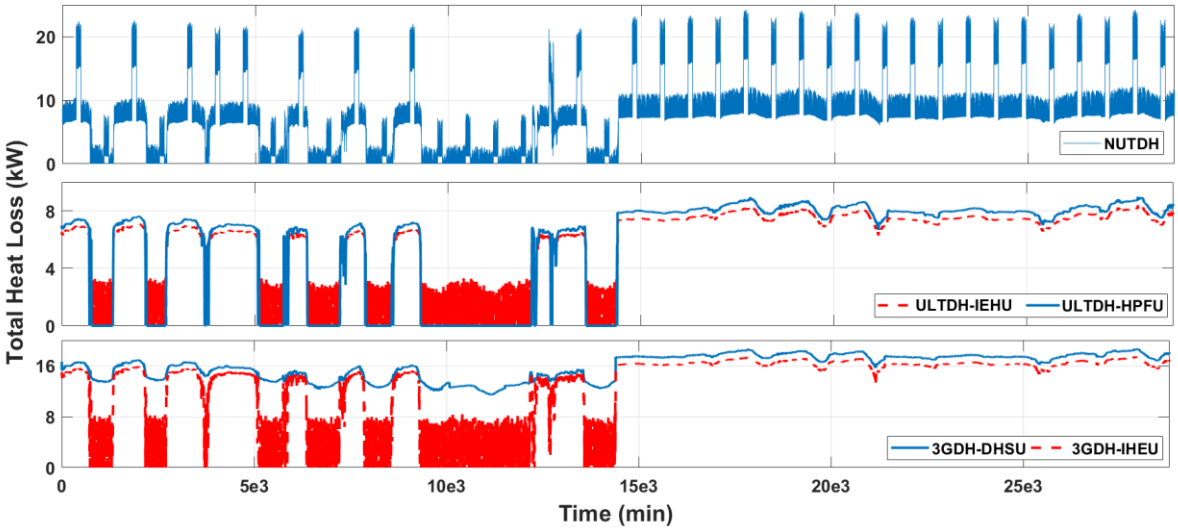

Figure 4-8 The rate of heat losses in the entire pipeline of the considered scenarios for an entire year.

As seen, expectedly, both of the ULTDH systems show the best performances among the considered cases. This is mainly due to its ultralow temperature of supply along the pipeline all the time. Among these two, the system with the electrical heater performs insignificantly better because of the lower heating duty and consequently lower mass flow rate and smaller dimension of the pipes. It can be seen that the NUTDH system performs far better than both of the 3GDH schemes and its heat loss rate is not considerably higher than the ULTDH cases. Among the 3GDH systems that show the poorest performances among the five scenarios, the system with IHEU substation shows a better performance because of the heat losses from the storage tanks in each substation.

Considering the presented information about the performance of the various systems, Table 4-3 gives statistics about the total annual heat loss from the transmission pipeline and the total annual pump work in each system. The table also presents the annual rate of heat loss, defined as the total annual amount of heat dissipated from the systems divided by that supplied to the network over the whole year. According to the table, and confirming the discussion above, it can be seen that the 3GDH-DHSU has the highest annual heat lost rate of 138.6 MWh and the low required pump work of 1.4 MWh. The 3GDH-IHEU system offers the lowest required amount of pump work with only 1.1 MWh while its annual heat loss is about the high value of 110 MWh. The high operating temperatures (that not only cause the highest rate of heat losses but also make the utilization of low-grade renewable energy and waste heat flows difficult) is the main reason why the 3GDH system is not compatible with the 4GDH system standards.

Table 4-3 The details of the overall performance of various district heating cases in an entire year.

DH Scheme	3GDH		ULTDH		NUTDH
Substation Configuration	**IHEU**	**DHSU**	**IEHU**	**HPFU**	
Total Annual Work of Pump (MWh)	1.1	1.4	1.2	1.4	6.9
Total Annual Heat Lost (MWh)	109.2	138.6	45.0	49.5	64.2
Rate of Heat Loss (%)	6.2	7.8	2.8	2.9	3.6
In-Building Electricity Use (MWh)	0	0	231.2	77.1	0
Rate of Demand Coverage (%)	100	100	75	83	100

On the other hand, the ULTDH-IHEU and ULTDH-HPFU systems offer the lowest rate of losses of 45 MWh and 49.5 MWh, and the low pump works of 1.2 MWh and 1.4 MWh, respectively. A ULTDH system should definitely present the lowest rate of loss because of its ultralow-temperature all the time. As such, its low rate of pump work is because this system is in off during many periods of the year. Although the ULTDH system, in both substation types, presents the lowest rates of losses, this technology is not compatible with the 4GDH system requirements. The first reason for this is that it does not provide the DHW of the end-users, and it needs an auxiliary heater for this purpose (i.e. the electrical heater and the heat pump). As seen in the last row of the table, the ULTDH schemes are the only technologies that do not cover all the heating demand of the network, i.e. 75% demand coverage by the IEHU and 83% coverage by the HPFU. The table shows that the ULTDH-IEHU and ULTDH-HPFU systems consume a total annual of 231.2 MWh and 77.1 MWh electricity in the buildings, respectively. This is in contrast with the main objective of district energy systems, i.e. to make the buildings independent of standalone energy systems. In addition, the previous studies show that the ULTDH-HPFU system does not offer a good cost-effectiveness, mainly due to the high cost of the heat pump in the substation [73]. Besides, the ULTDH systems require additional legionella prevention/elimination processes as the temperature range of 25-45 ^{o}C is the most appropriate condition for legionella growth and multiplication.

Finally, according Table 4-3, the NUTDH system offers the highest pump work of 6.9 MWh for a year. This is not considered as a drawback as long as this system can provide the very low rate of heat loss of 64.2 MWh, which is almost half the 3GDH systems loss. This is equal to the interesting annual heat loss rate of 3.6%. Note that these values are much smaller than real life district heating systems where, for example, a regular 3GDH presents about 25-30% of heat loss. This is because, in this work, the heat loss from the distribution pipes is not taken into account by considering the heat production chain just near to the network. For the transmission losses of different cases, the given loss rate values could be simply scaled with the same ratios to each other.

Considering the presented results, one may conclude that the proposed NUTDH system may be an appropriate solution for the 4GDH system. This can be concluded because:

- it offers a very low rate of loss,
- its very low average supply temperature facilitates the utilization of low-grade renewable technologies and waste heat flows.
- the risk of legionella is totally solved via the periodic thermal disinfections (high-temperature operations).

Hereafter, some of the key factors in the design and dimensioning of the NUTDH system are discussed. One of the key parameters in designing the NUTDH system for the case study is the size of the storage tanks for each neighborhood/street. Naturally, the larger the storage tanks are, the slower they are discharged. Meanwhile, a larger storage tank has a higher cost and a larger rate of loss. Technically, based on the Brazilian DHW standard, the minimum top-node temperature of the storage tank should be 60 ^{o}C. Considering the effectiveness factor of the heat exchangers, the storage tank volume that provides the minimum top-node temperature of 65 ^{o}C during the year is opted as the optimal tank volume in this work. Figure 4-9 shows the minimum top-node temperature observed among all the storage tanks in the network for various volumes. According to the table, a heat storage unit with 0.61 m^3 is the best choice for this system.

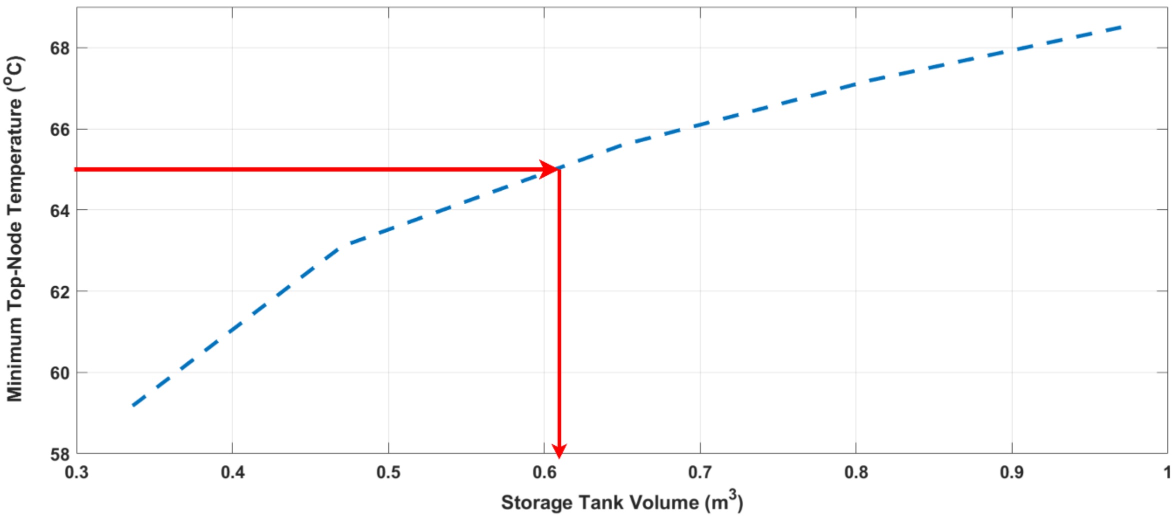

Figure 4-9 Dimensioning the storage tanks in the NUTDH system.

Figure 4-10 shows the supply temperature of the storage tanks in the network over the two-hour charging periods in two typical seasonal days, i.e. one in summer and another in winter. This is an index of showing how the temperature drops along the pipeline. As seen, the level of temperature drop is larger for the further storage tanks because of the longer distance that the fluid passes and due to the reduction of the mass flow rate after each street, which increases the rate of heat losses. In addition, it is observed that the temperature drop during the summer day is larger. This is reasonable because the mass flow rate through the pipes is smaller in the summer and this causes a larger temperature drop along the pipe. Overall, the level of temperature drop is not significant due to the small length of the pipeline.

The next step is to present the results of the simulations accomplished on the heat storage tanks. Before that, one should validate the model and the numerical method used for this purpose. For this, the numerical results are compared with the experimental results reported in [74] for a storage tank with the height of 1.93 m, a diameter of 1.16 m, and an insulation thickness of 0.05 m. The tank is initially at the uniform temperature of 20.5 ^{o}C and is charged with a hot water flow at 39 ^{o}C and specific flow rates. The detailed characterisics of the tank and the experiment conditions are given in the reference work [74].

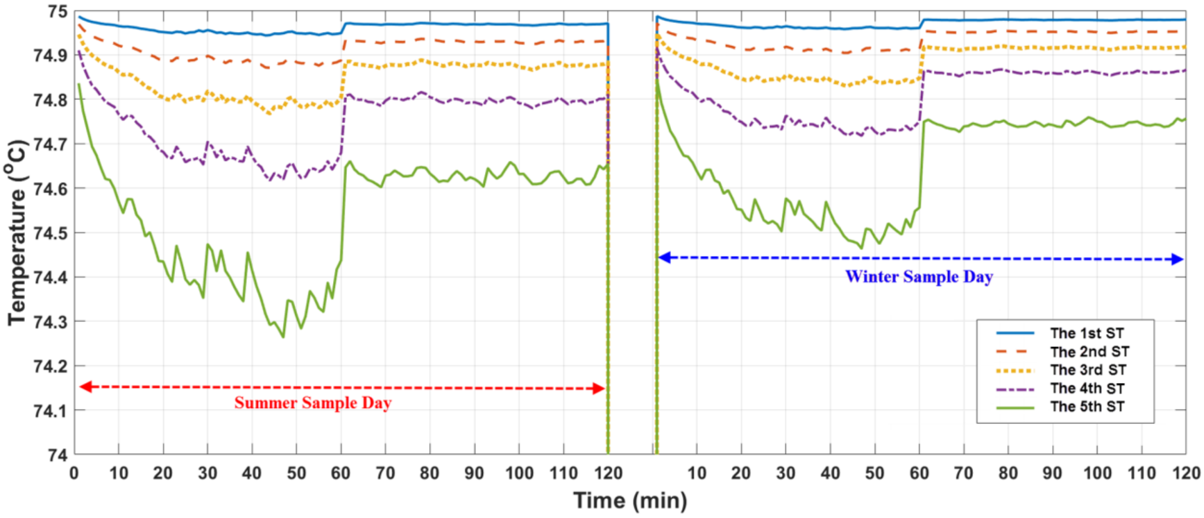

Figure 4-10 The supply temperature of the storage tanks in two different sample days, ST: storage tank.

Figure 4-11 makes a comparison of the numerical and experimental results for the given heat storage tank when hot water with a mass flow rate of 1364 kg/h is applied to charge the tank for 0.5 h, 1 h, and 1.5 h. As can be seen, there is a strong agreement between the layer temperatures predicted by the model and those reported by the experimental work. Therefore, the numerical model and method used for the stratified heat storage tanks can be reliably used in this work.

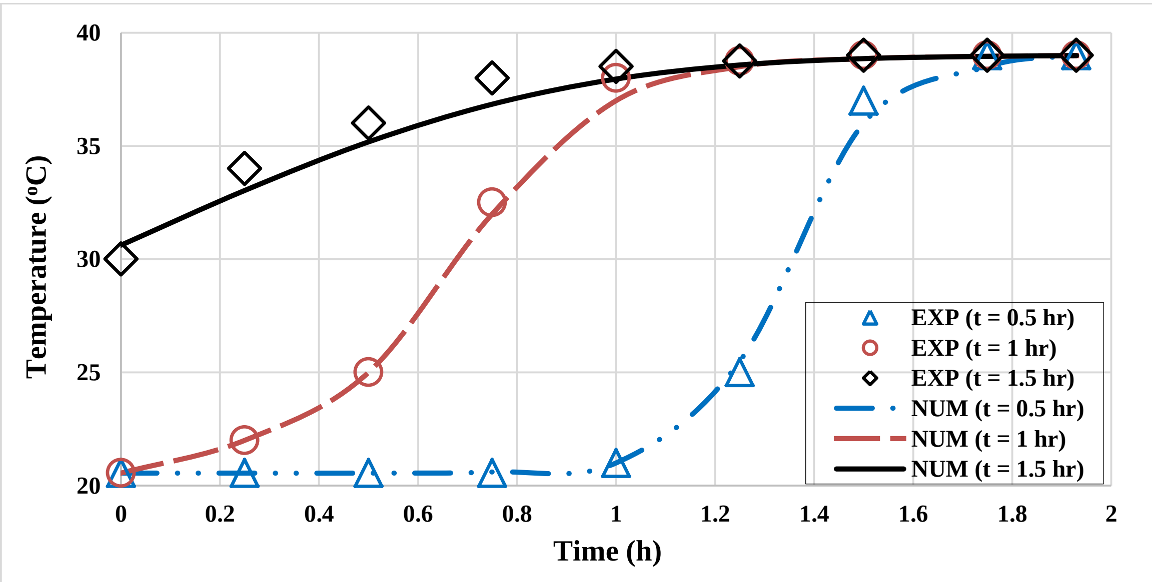

Figure 4-11 The Comparison of the numerical and experimental storage tank temperature distribution profiles.

Figure 4-12 shows the variation of the temperature of the storage tanks (at three levels of bottom, middle and top) over a sample winter day. Figure 4-12 shows how the storage tank is charged during the charging time and how it is discharged when the system is in low-temperature mode. As Figure 4-12 shows all the nodes approach the supply high temperature during the charging phases. It is seen that the temperature profile of the bottom node of the tanks fluctuates more sharply. This is because the hot water of the upper nodes is replaced by the colder yet warm water of their below nodes, whereas for the bottom node, it is replaced by the cold water of the district heating return line (at 35 ^{o}C).

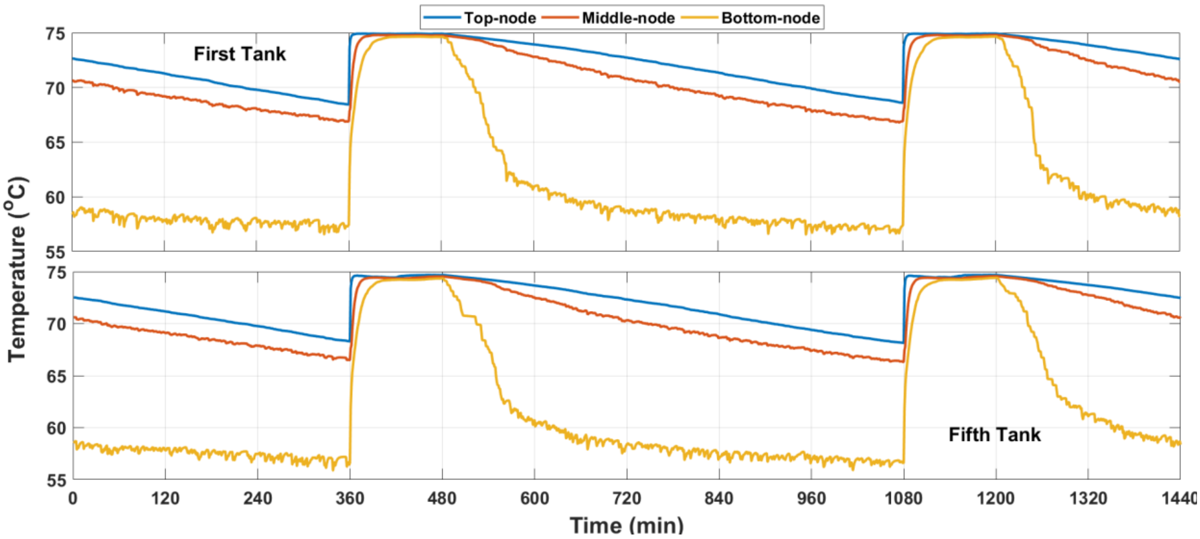

Figure 4-12 The temperature of the tanks in different nodes over a sample winter day.

As discussed, the NUTDH system has two supply lines after the heat storage units, one for space heating and the other for supplying the DHW demand of the users. Figure 4-13 and Figure 4-14 compare the supply temperatures of a few buildings (building 1 in street I, building 10 in street III and building 20 in street V) in their DHW and space heating lines, respectively. These graphs are also presented for one winter day and one summer day. Note that, the DHW flow with a temperature higher than the standard value is mixed with cold water to reach the desired temperature level.

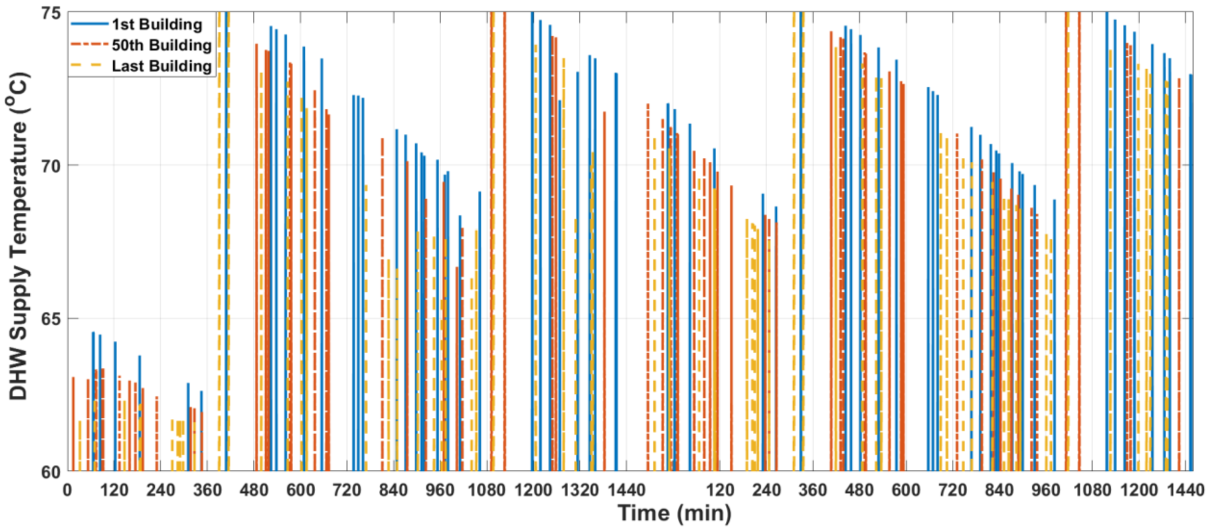

Figure 4-13 DHW supply temperature of a few buildings in the network.

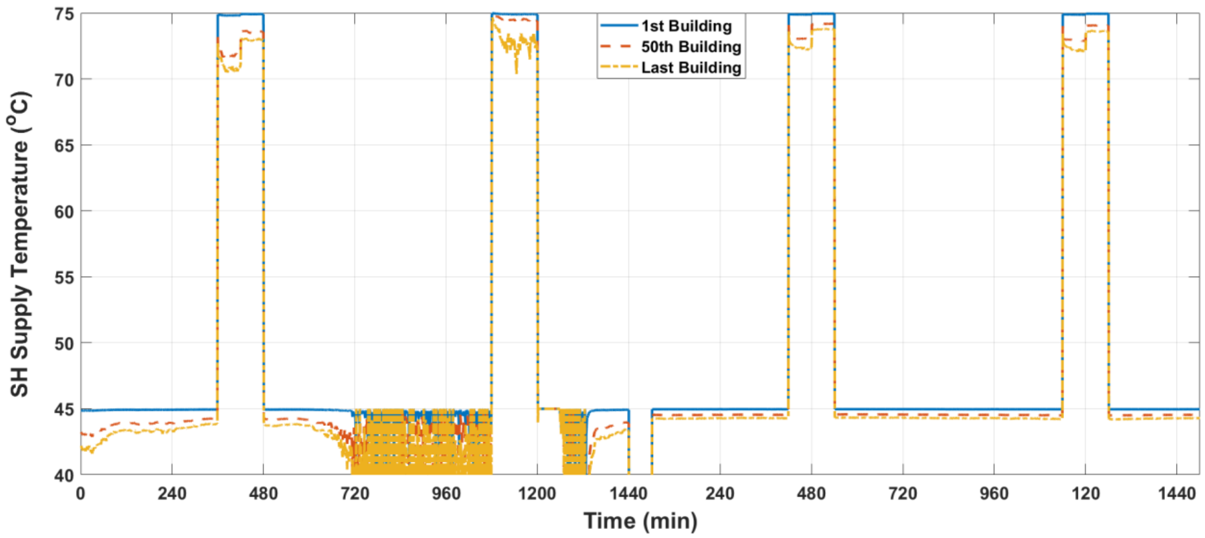

Figure 4-14 Space heating supply temperature of a few buildings in the network.

Hereafter, the results associated with the numerical modeling of the pipes in various scenarios is presented. Figure 4-15 presents the contours of temperature distribution in the radial direction of the insulations in each of the three cases. The legends of the contours give an indication of the temperatures that the colors represent for each case.

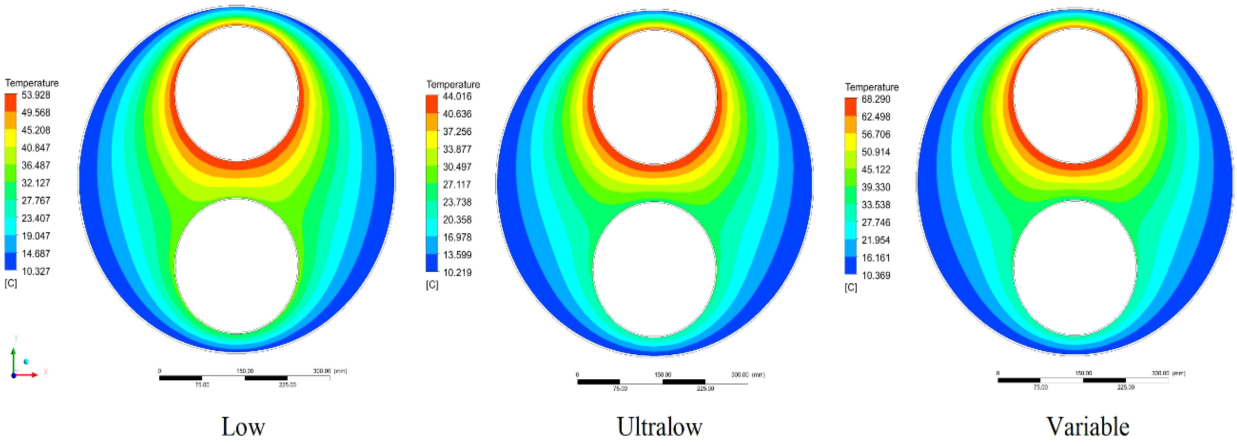

Figure 4-15 The contours of radial temperature distribution for the pipes through the insulations.

Figure 4-16 presents these contours for the working fluid itself through the supply line (upper panel) and the return lines (lower panel) of the pipes. As seen, the effect of the supply line temperature on the temperature of the return line in all the three cases is observable and this effect is smaller in the ultralow-temperature case.

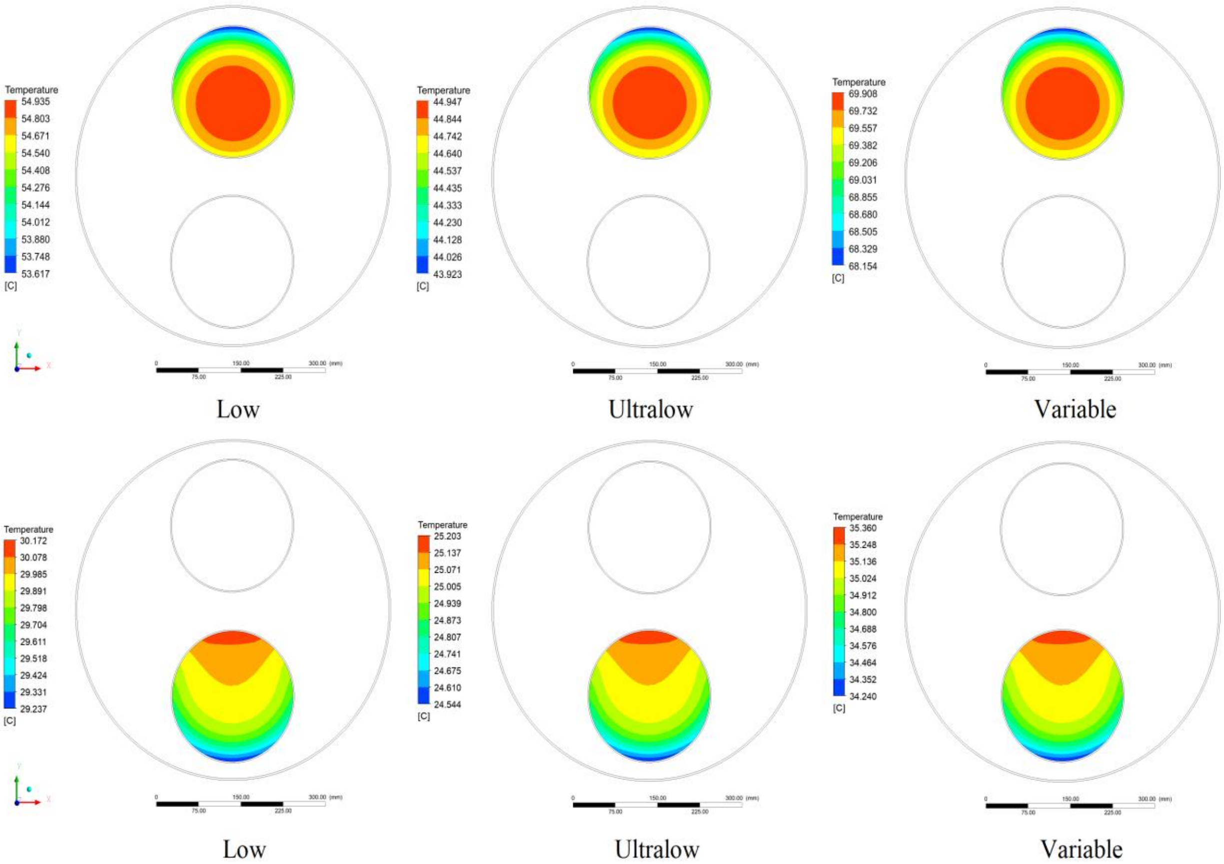

Figure 4-16 The contours of radial temperature distribution for the working fluid inside the pipes.

Results and Discussions

Figure 4-17 presents the profile of temperature drop in the axial direction along the supply and return lines of the pipeline (up to 10 km) for the three different scenarios. It should be mention that VTDH scenario refers to variable temperature district heating. For making these graphs, the same ambient temperature (5 ^{o}C) is applied for all the three cases. As such, in order to make a fair comparison, the same heat delivery rate (5 MW), instead of the same mass flow rate, is applied. Naturally, for delivering the same amount of heat to the end-users, the mass flow rate of different cases should be different (as a direct function of the temperature difference of the supply and return lines). Thus, the ULTDH system will result in the highest mass flow rate while the NUTDH system when working on the high-temperature supply mode makes the lowest mass flow rate. It is reminded that the NUTDH system has two operational modes. When it is on the low-temperature supply mode, it makes a profile just the same as the ULTDH case.

As seen, the highest level of temperature drop is for the NUTDH system while working on the high-temperature supply mode. It will be just below 1 $^{\circ}$C and 0.23 $^{\circ}$C for the return line. However, one should note that, based on the design of the NUTDH system, this only applies for a short period of time a day, e.g. 2 hours, and the rest of the day, the temperature drop profile will be just similar to the ULTDH system. In this case, the temperature drop values for the supply and return lines over a 10 km pipe will be 0.33 °C and 0.07 °C, respectively. The LTDH system presents the temperature drop values of 0.52 °C and 0.14 °C as moderate temperature drop values among the three cases.

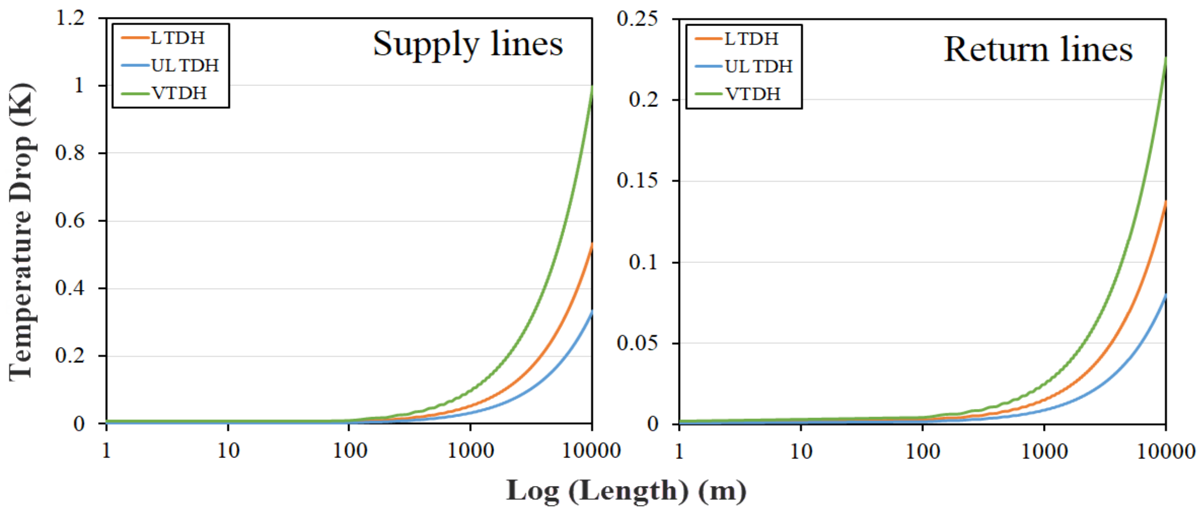

Figure 4-17 The temperature profile of the supply line of the pipes for different cases.

Figure 4-18 investigates the effect of Reynolds number on the level of total heat loss through the pipes in different cases. This includes the heat loss from the supply lines (the left panel), and the return lines (the right panel). As expected, regardless of the Reynolds number, for the same mass flow rate of the heat carrier, the NUTDH system while working in the high-temperature supply mode makes the higher rate of loss while the ULTDH system results in the lowest rate of heat loss. This is true for both of the supply and return lines. On the other hand, the increase in the Reynolds number makes a growth in the rate of loss. This is again expected because increasing the Reynolds number means the increase in the mass flow rate. However, this increment of the rate of heat loss is not linear and the pace of growth falls as the mass flow rate goes up. This fact also applies to both of the return and supply lines of all the three cases.

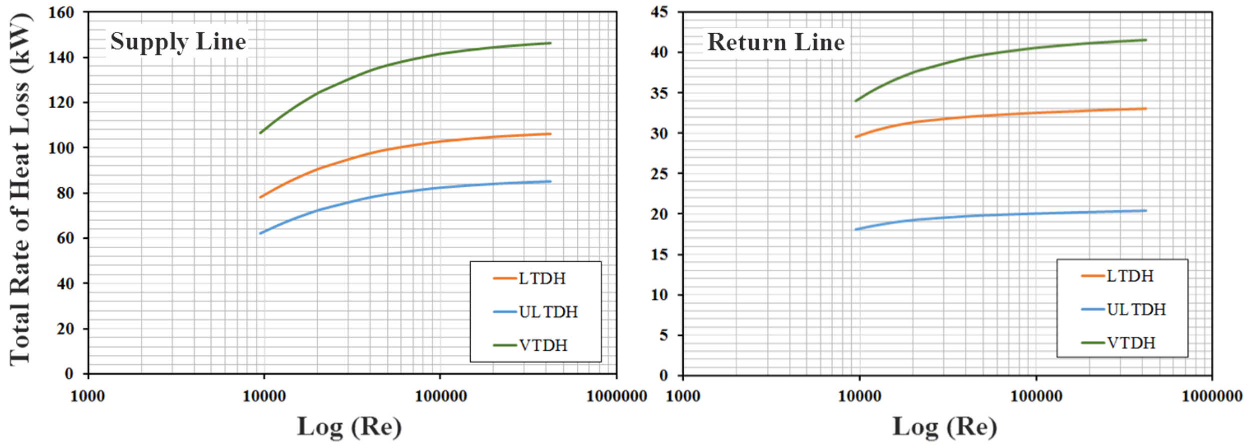

Figure 4-18 The effect of Reynolds number on the level of heat loss through the supply and return lines.

Figure 4-19 shows the total rate of heat loss from both of the supply and return lines along the 10 km pipelines. Figure 4-12, indeed, gives an indication of the summation of the losses reported in Figure 4-18 for each case. Naturally, the trends remain the same and the ULTDH system results in the lowest rate of loss while the LTDH system presents an average level of loss and the NUTDH system (when in high-temperature supply mode) gives the highest rate of total heat loss.

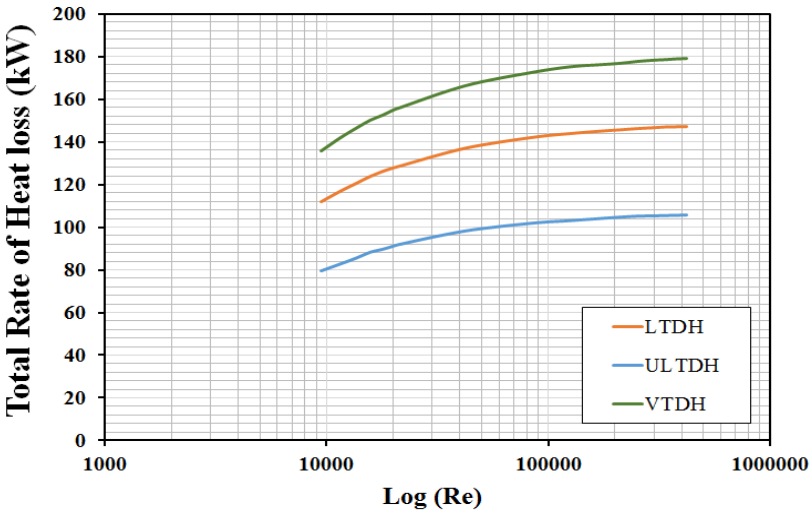

Figure 4-19 The effect of Reynolds number on the level of total rate of heat losses over the entire pipeline.

Figure 4-20 illustrates the effect of Reynolds number on the level of temperature drop for the supply and return lines over the entire pipeline for each of the three scenarios. Again, the NUTDH system while working at the high-temperature supply mode makes the largest value of temperature drop in the supply line while the ULTDH system makes the lowest value of temperature drop through that.

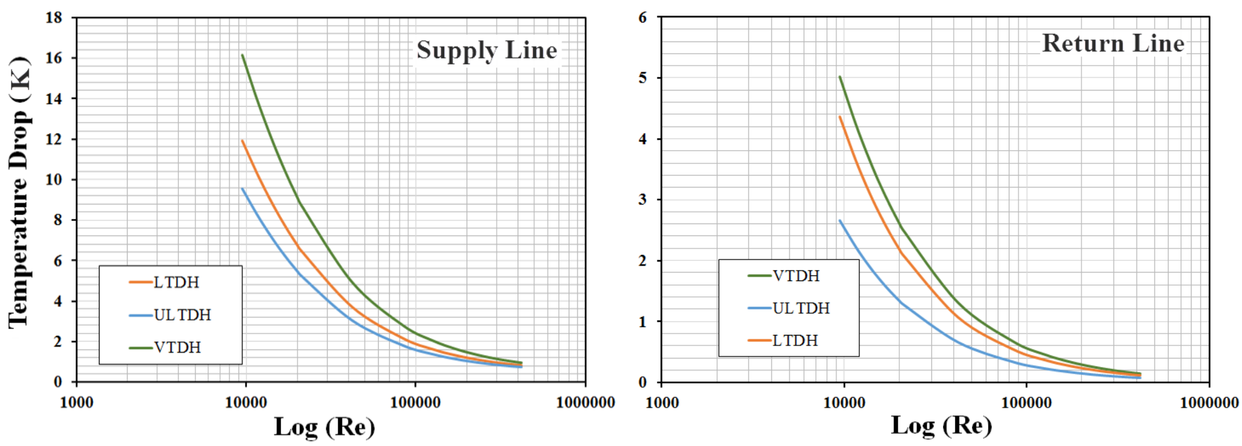

Figure 4-20 The effect of Reynolds number on the level of temperature drop over the entire pipeline

Figure 4-21 illustrates the profile of pressure for a unit meter of the pipeline for the above-mentioned three cases. Here again, to make a fair comparison, the same amount of heat delivery (rather than a similar mass flow rate) is applied to the pipes in different cases. The presented pressure drop profiles are related to the supply lines only. As the characteristics of the pipes and the mass flow rates are the same for the supply and return lines, and due to the fact that there is a small temperature difference between these lines making insignificant changes in the physical properties of water, almost the same pressure profiles are obtained for the return line. Note that the value of pressure drop increases proportionally as the length of the pipe increases. The pressure drop values are important because they can represent the amount of work required for running the booster pumps along the paths for compensating the pressure losses through the pipes.

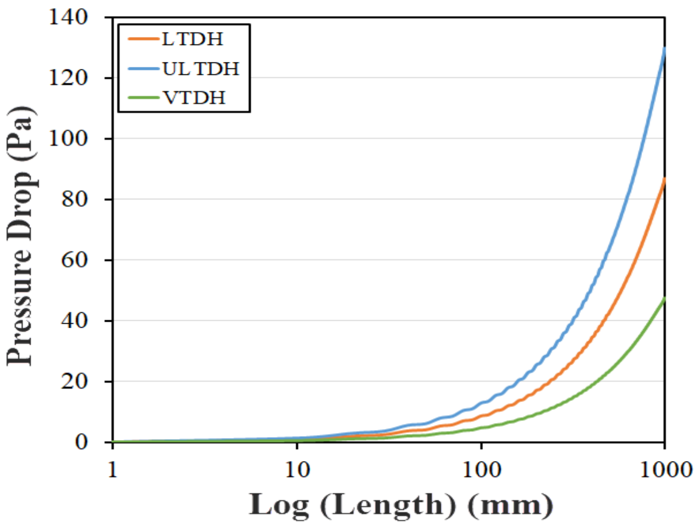

Figure 4-21 The level of pressure drop through the supply lines in each case.

Figure 4-22 shows the variation of Nusselt number for each of the three cases as a function of Reynolds number. As seen, there is not a significant difference between different cases, neither in the supply lines nor through the return lines.

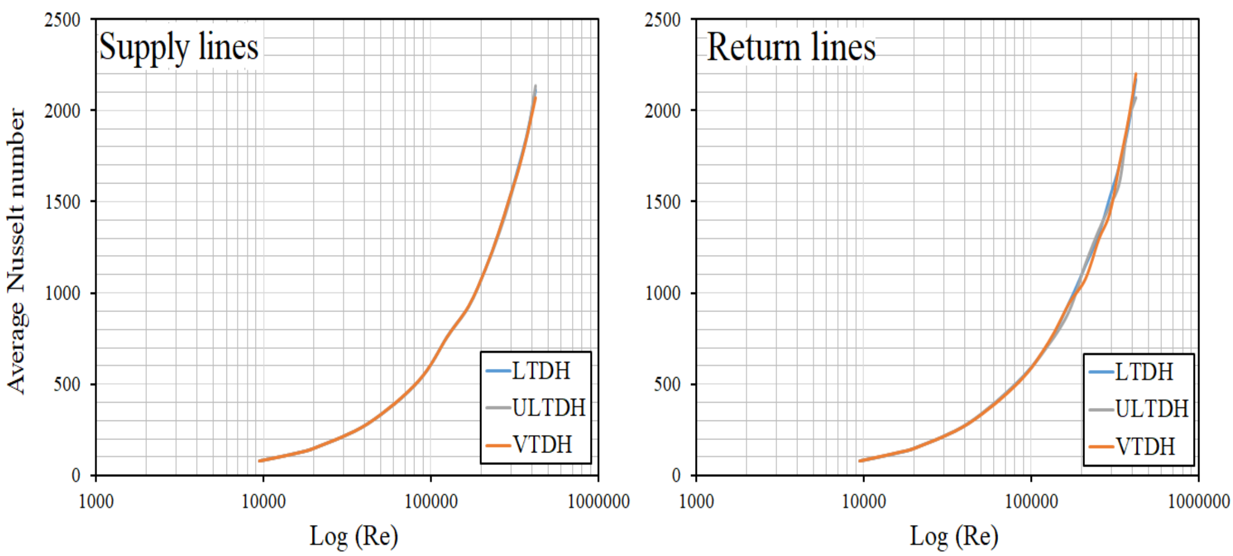

Figure 4-22 The effect of Reynolds number on Nusselt number through the pipes in each case.

Figure 4-23 presents the profile of the heat loss from the supply and return lines of each case as a function of the angle. The right panel in Figure 4-23 shows how the angle is considered, where the upper tube is the supply line and the lower tube is the return line. According to the figure, at angle 0°, where the supply line is closer to the surrounding soil, the loss is at its maximum level while when it approaches the angle 180° gets minimal. For the return line, the trend is revers because the supply line at angle 0° heats it and the losses get maximal when the angel goes toward 180^{o}. The noteworthy point is that the heat loss from the supply line to the return line is not considered as a loss for the whole pipe because it remains in the system.

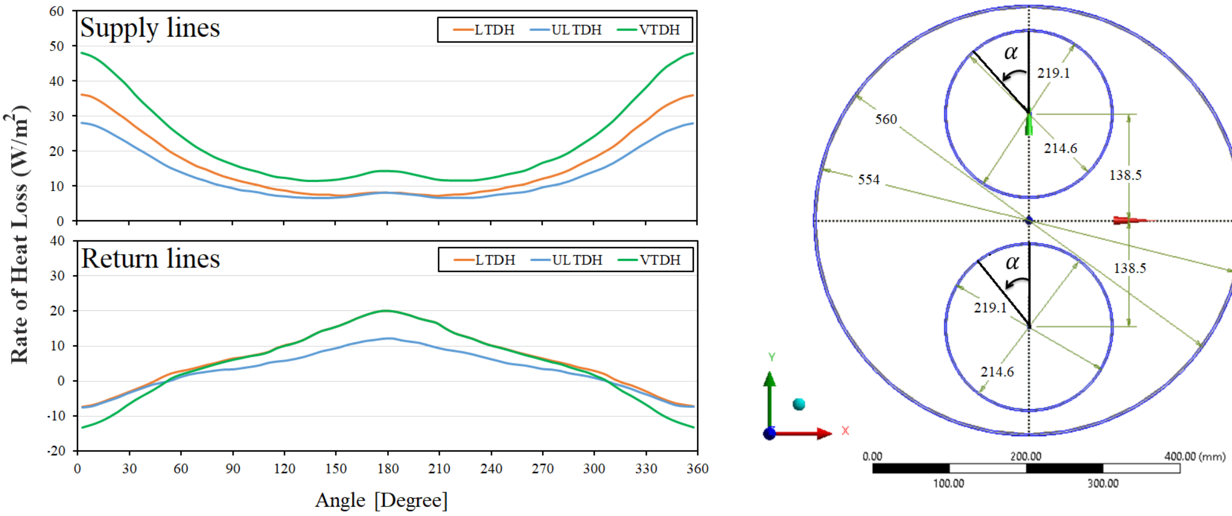

Figure 4-23 The rate of heat loss from the supply and return pipes as a function of angle in each case.

Figure 4-24 shows the effect of variation in the insulation thickness for each case for the supply and return lines and the total losses through the pipe (the lower panel). The considered range for the thickness of the insulation come in the form of the outer diameter of the casing just above the outer diameter of the pipes up to an outer diameter of 870 mm. According to Figure 4-24, although the effect of increasing the thickness of the insulation gets milder as the thickness grows, a significant reduction in the rate of loss is seen in the lower ranges of the outer diameter. For example, an increase of the outer diameter from 554 mm to 590 mm results in about 20%, 18% and 14% less rate of heat loss for the supply lines of the NUTDH system (from 150 kW to 123 kW), the LTDH system (from 110 kW to 90 kW) and ULTDH (from 85 kW to 72 kW), respectively. The rates of improvement for the return lines are even more impressive, where for the NUTDH, LTDH and ULTDH systems the saving rates of respectively 33%, 32%, and 30% are observed for the same amount of insulation strengthen. The heat loss rate values (before and after increasing the insulation thickness) for the whole pipe, including both supply and return lines, are 180 kW to 143 kW, 138 kW to 110 kW, and 105 kW to 83 kW, respectively. Since the main objective of lower operating temperatures in the district heating systems is getting a lower rate of loss, the reinforcement of the insulations in the transmission pipelines compared to the existing standard pipes could be highly helpful. Of course, this needs an optimization based on techno-economic considerations to see how much it would cost to make such a reinforcement of the insulation per meter of the pipe and how much benefit it would make. This, naturally, will be different for various district heating schemes.

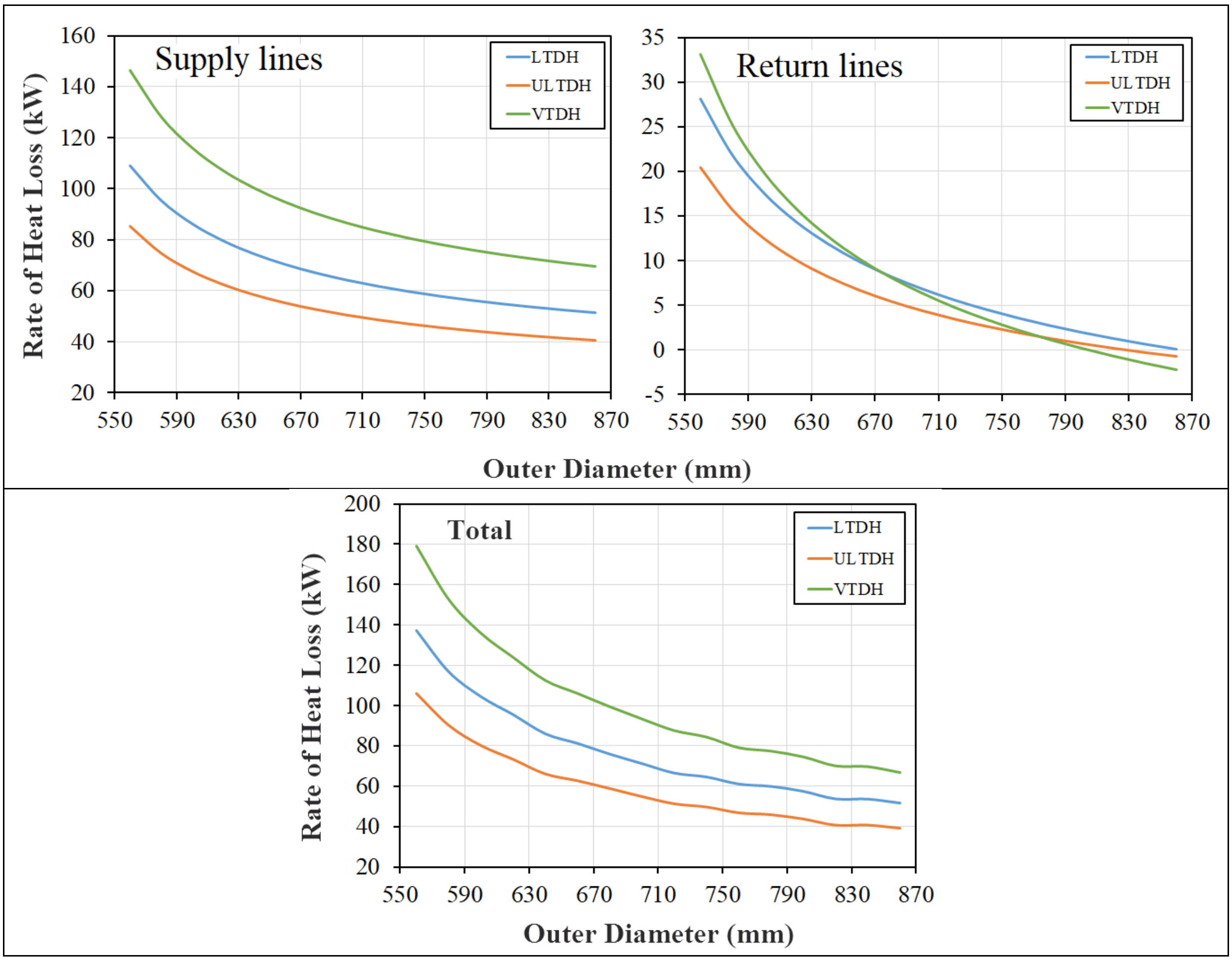

Figure 4-24 The effect of strengthening the insulation of pipes for various cases.

Figure 4-25 shows the variation of the outlet temperature of the supply and return of the NUTDH system over time. The main objective is to investigate the effect of thermal inertia of the pipe to see if this parameter allows the system to operate as it is expected for a NUTDH system. For this, it is assumed that the system is in its ultralow-temperature supply mode (45/25 ^{o}C), and then suddenly, the high-temperature supply mode starts (70/35 ^{o}C). Naturally, as the operation mode changes, due to the long length of the pipe (10 km) and the limited velocity of the flow through the pipes (considered about 1 m/s here), even if there is no external parameter affecting the flow, it would take time to this reaches to the outlet point (about 166 minutes). However, it can be seen that even if the effects of thermal inertia of the pipe is neglected, the external effects cause the outlet temperature to gradually approach the supply temperature. This even will take longer (almost double) when the effects of the thermal inertia of the pipe are taken into account. A similar trend is observed for the return line as well. Therefore, it is clearly seen that a variable temperature supply could not be effectively taken for a district heating pipeline without considering the expected delay in reaching the desired temperature at the end of the pipeline. This means that the timing which is set for the operation of the NUTDH system should be based on the calculated delays and the demand of the networks to charge the decentralized heat storage units.

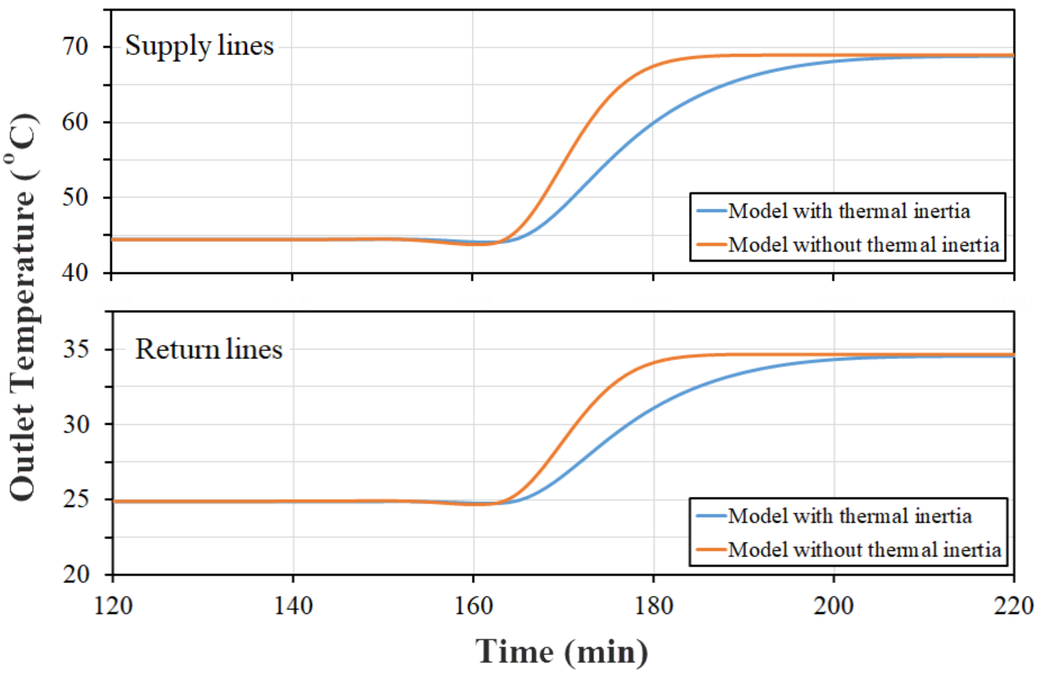

Figure 4-25 The effect of the thermal inertia on the outlet temperature of the NUTDH system.

Figure 4-26 compares the trend of variation of the outlet temperature of a NUTDH system when the length of the pipeline is 10 km and 1 km, respectively. For this, a complete high-temperature, ultralow-temperature supply process, based on the given definition for this system, is accomplished. That is, the pipeline is first at 45 $^{\circ}$C, and then suddenly two hours of 70 $^{\circ}$C supply, followed by 2 hours of 45 $^{\circ}$C supply, is applied.

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According to Figure 4-26, the outlet temperature of the long pipe comes with a big delay while the small length of the pipe for the second case makes the desired outlet temperature achievable after a short delay. This result is a favorable outlet temperature profile out of the regular high-temperature and ultralow-temperature supply procedures for the NUTDH system with a short length of 1 km. This means that the effect of thermal inertia of the pipe on the distance of the heat storage units to the end-users is negligible. In addition, the figure proves that in case of decentralized heat production, i.e. neighborhood scale heat production for a district heating network, the utilization of a NUTDH scheme would make much more sense and requires a simpler control method.

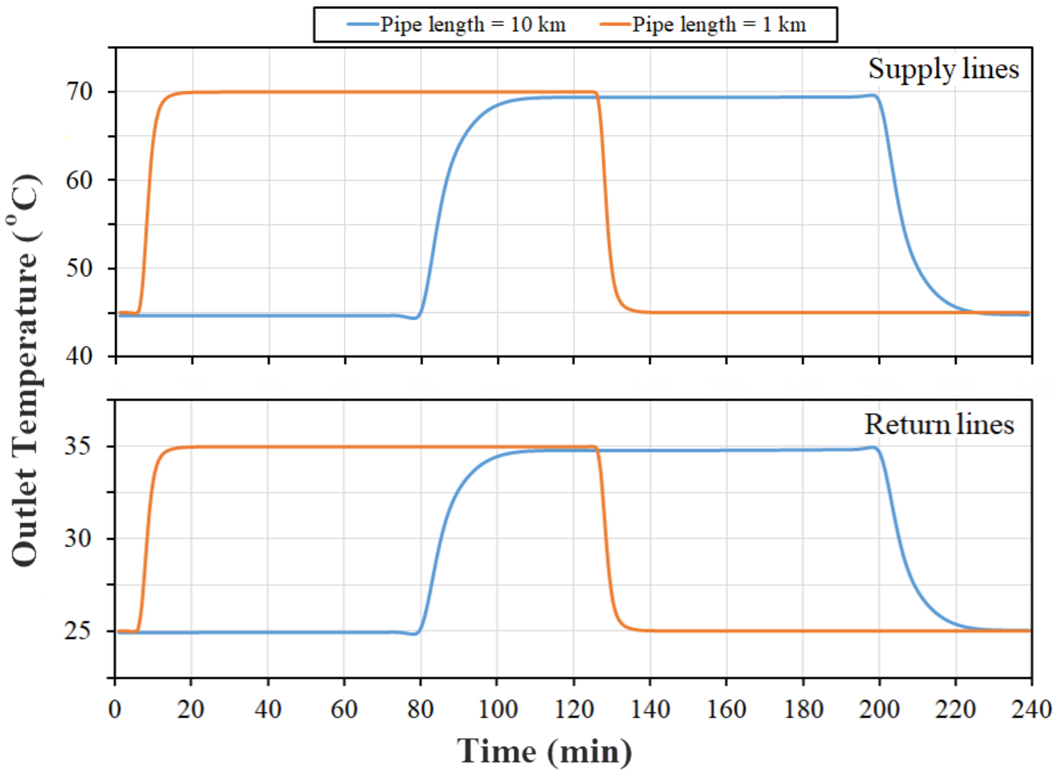

Figure 4-26 The trend of the outlet temperature in a NUTDH system with different pipe lengths.

4.2 Final Considerations

The results of the modeled district heating system showed the feasibility of the implementation of such advanced system as a promising system in the near future to integrate with the sustainable smart energy system. For this reason, all countries should take steps toward this kind of efficient and economic energy systems to solve energy crisis and use of renewable energy sources as much as possible. In this research, a small city in the south of

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Brazil was investigated as a pilot, however, the experience of this research can be used for modeling other cold cities and regions to improve energy matrix in Brazil. Therefore, it can be suggested to plan a roadmap in the country to develop the latest generation of district heating system.

Various possible district heating scenarios that can be practically implemented by today were investigated and compared together. Finally, the innovative proposed system that works with the variable and non-uniform temperature showed a better overall energy performance in comparison to other feasible systems which work with low and ultra-low temperatures. In addition, the results showed the compatibility and feasibility of the existing twin pipes technology with the proposed system. This is very important since twin pipes are more efficient thermally and economically with a lower heat loss rate.

Next, different aspects of the proposed system such as the heat storage size, the temperature distribution, the effect of Reynolds number, the pipeline length, the insulation of the pipes, and the thermal inertia were studied.

In fact, the proposed model has all main necessary features to meet 4GDH requirements. These features comprise low temperature distribution, low heat loss, higher thermal efficiency, integration with other energy sectors, and the capability of use of clean, low-grade and renewable energies to reach a sustainable energy system. Although, some technical and organizational issues should be considered for the implementation of this new concept. The effect of thermal inertia in pipes and consequently, the delay in reaching high supply temperatures is very important to charge the decentralized heat storages. Finding an efficient strategy for the charging process of the heat storage units, the operation of the flow control valves, and obtaining an optimum size for the storage tank are other key factors in this system. The insulation of the pipes and techno-economic calculations to achieve the best reinforcement of the insulation should be investigated. And finally, the institutional frameworks, strategic energy plans, and increase of the public awareness and commitment to establish this new technology are crucially required.

5 Conclusions

In this thesis, the technical feasibility of using district heating systems in a case study in the southern part of Brazil was investigated. For this, the novel concept of NUTDH system was proposed and thermodynamically analyzed. In addition, the thermo-hydraulically compatibility of the currently in use pipes in the existing district heating systems with the proposed district heating scheme was assessed. In addition of the thermodynamic feasibility of a 4GDH system in Brazil, another important objective of this work is to introduce the new heat supply technology that strongly meets the 4GDH systems criteria, i.e. a very low heat loss rate; low supply/return temperatures so that low-grade renewable technologies and waste heat sources may integrate, etc.

In methodology, a cold and small city that will have access to renewable energy sources in near future was chosen to show the feasibility of the proposed approach. Then, geographical and climatological data including population of 300 people, 100 detached houses in the medium-sized category along with weather temperatures over the year was studied. Next, thermodynamic modeling of pipelines and substations was investigated. As discussed, the substations can comprise the storage tank and plate heat exchangers. Finally, twin pipe compatibility with the proposed system was verified only for transmission lines.

The NUTDH system was precisely designed and sized in all the components and its performance was analyzed for the case study. The results were compared with those obtained for the performance of other competitive district heating schemes, i.e. ULTDH and 3GDH systems with different substation configurations. The results showed that the system might sufficiently provide the required heat of the whole network for both space heating and DHW uses. It was proved that the NUTDH system, with a total annual loss of about 64 MWh, is extremely more efficient than the 3GDH in both substation configurations with the annual heat losses of about 109 MWh and 138 MWh. The NUTDH loss is slightly higher than the ULTDH schemes with the annual heat losses of about 45 MWh and 50 MWh. Even though the loss of the proposed system is higher than the ULTDH schemes, it is still superior because not only the ULTDH schemes do not cover the DHW demand of the end-users and require in-building electricity supply, which is a serious drawback, but also they need additional legionella prevention measures. In addition, the ULTDH-HPFU system suffers from the too high cost of capital of the substation. Overall, the NUTDH system showed a high feasibility for being a dominant district heating scheme in the future, though it still needs more investigations in various techno-economic aspects to address the technical and practical gaps before being broadly implemented.

One of the most important aspects for further investigations is the compatibility of the pipes with the proposed NUTDH system as the pipeline is the most important part of a district heating system. Therefore, as the second phase of this project, the thermal hydraulic performance the pipes in presence of variable temperature supply, like that of the NUTDH system, was numerically studied. Here also, the different district heating scenarios were considered in the numerical simulations. After presenting the general thermal and hydraulic performance assessment results of such pipes for all the three considered cases, i.e. the ULTDH, the LTDH and the NUTDH systems, the effect of changing the insulation thickness and the thermal inertia of the pipe on the performance of each of the cases was investigated.

The results show that, expectedly, the ULTDH system shows the lowest rate of loss and temperature drop along the pipe, where the NUTDH system results in a higher rate of loss and temperature drop when working on a high-temperature supply mode. The results show that the twin-pipe (as they are) may fit the ULTDH and LTDH systems while there seems to be space for strengthening the insulation of the pipes for a further reduction of the rate of losses in a cost-effective way. This, however, requires an economic trade-off to see how much would be gained for a certain amount of reinforcement of the insulation and how much this will increase the cost of the pipes.

On the other hand, it was shown that the concept NUTDH system requires the consideration of the effect of thermal inertia of the pipe for an accurate high-low temperature supply control method so as not leave the end users remain with an uncovered demand or the storage tanks being fully discharged. This problem, however, is not that serious for a district heating with shorter transmission pipeline or those bringing the dynamic supply temperature to the pipes close to the end-users (below 1 km distance for example). Although the thermal inertia of the pipe affects the practically achievable supply temperature of the end-users in this system as well, the effect is so small that the system can reliably be employed by the regular existing twin-pipes.

Conclusions

The technical feasibility of use of district heating systems in a case study in the southern part of Brazil was studied and it was confirmed by the results. The novel concept of NUTDH system was proposed and thermodynamically analyzed. This analysis showed this concept strongly meets the 4GDH systems criteria. Besides, the thermo-hydraulically compatibility of the currently in use twin-pipes in the existing district heating systems with the proposed district heating scheme was assessed. Actually, all of predetermined objectives of this research were achieved.

Exergy analysis of the system to explore exergy losses and inefficiencies of it and improving the energy efficiency of the system can be considered as an important step to continue this research. The optimization of the charging process of the heat storage units and the operation strategy of the flow control valves, can be suggested as the next step of this research project. The thermo-economic analysis of the insulation of pipes is another important subject to study.

The cost estimation of such projects and the optimization of different parameters of the district heating system such as pipeline sizing, costs, and use of different heat product units can be studied in this area. The analysis and the optimization of the heat supply chain, the use of triple-pipe concept specifically for substations, and developing this research to bigger cold cities even in other countries, also, can be considered as suggested projects in the future along this thesis project.

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