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Programme of Research, Development and Demonstration on District Heating and Cooling

Optimization of District Heating Systems by Maximizing Building Heating System Temperature Difference

Chris Snoek¹, Libing Yang¹ Tom Onno² Svend Frederiksen³ Hans Korsman⁴

¹CANMET Energy Technology Centre, Natural Resources Canada – Ottawa, Canada
 ²Gagest Developments Ltd. – Ottawa, Canada
 ³JFR Engineering I/S – Copenhagen, Denmark
 ⁴Nuon Technisch Bedrijf – Duiven, The Netherlands

Introduction

The International Energy Agency (IEA) was established in 1974 in order to strengthen the cooperation between member countries. As an element of the International Energy Programme, the participating countries undertake co-operative actions in energy research, development and demonstration.

District Heating offers excellent opportunities for achieving the twin goals of saving energy and reducing environmental pollution. Its is an extremely flexible technology which can make use of any fuel including the utilisation of waste energy, renewables and, most significantly, the application of combined heat and power (CHP). It is by means of these integrated solutions that very substantial progress towards environmental targets, such as those emerging from the Kyoto commitment, can be made.

For more information about this Implementing Agreement please check our Internet site <u>www.iea-dhc.org/</u>

Annex VI

In May 1999 Annex VI started. The countries that participated were: Canada, Denmark, Finland, Germany, Korea, The Netherlands, Norway, Sweden, United Kingdom, United States of America.

Title project	ISBN	Registration number
Simple Models for Operational Optimisation	90 5748 021 2	S1
Optimisation of a DH System by Maximising Building System Temperatures Differences	90 5748 022 0	S2
District Heating Network Operation	90 5748 023 9	S3
Pipe Laying in Combination with Horizontal Drilling Methods	90 5748 024 7	S4
Optimisation of Cool Thermal Storage and Distribution	90 5748 025 5	S5
District Heating and Cooling Building Handbook	90 5748 026 3	S6
Optimised District Heating Systems Using Remote Heat Meter Communication and Control	90 5748 027 1	S7
Absorption Refrigeration with Thermal (ice) Storage	90 5748 028 X	S8
Promotion and Recognition of DHC/CHP benefits in Greenhouse Gas Policy and Trading Programs	90-5748-029-8	S9

The following projects were carried out in Annex VI:

Benefits of Membership

Membership of this implementing agreement fosters sharing of knowledge and current best practice from many countries including those where:

- DHC is already a mature industry
- DHC is well established but refurbishment is a key issue
- DHC is not well established.

Membership proves invaluable in enhancing the quality of support given under national programmes. The final materials from the research are tangible examples, but other benefits include the cross-fertilisation of ideas which has resulted not only in shared knowledge but also opportunities for further collaboration.

Participant countries benefit through the active participation in the programme of their own consultants and research organizations. Each of the projects is supported by a team of Experts, one from each participant country. The sharing of knowledge is a two-way process, and there are known examples of the expert him/herself learning about new techniques and applying them in their own organization.

Information

General information about the IEA Programme District Heating and Cooling, including the integration of CHP can be obtained from:

IEA Secretariat Mr. Hans Nilsson 9 Rue de la Federation F-75139 Paris, Cedex 15 FRANCE +33-1-405 767 21 Telephone: Fax: +33-1-405 767 49 E-mail: hans.nilsson@iea.org or The Operating Agent NOVEM Ms. Marijke Wobben P.O. Box 17 NL-6130 AA SITTARD The Netherlands Telephone: +31-46-4202322 Fax: +31-46-4528260 E-mail: m.wobben@novem.nl

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This paper does not necessarily fully reflect the views of each of the individual participant countries of the Implementing Agreement on District Heating and Cooling, including the integration of CHP.

Summary

An effective district heating (DH) system has two primary distinguishing features: a low supply temperature and a high temperature difference between the supply and return (Δ T). A low supply temperature results in an increased overall efficiency if combined heat and power (CHP) and/or waste heat is utilized, while a high Δ T results in low flow in the DH system.

In this study, the principle of cascading loads in building heating systems was used to increase the temperature difference between the supply and return.

To thoroughly evaluate the thermodynamic and economic performance of different building systems, all three main DH components (heat production plant, distribution network and building heating system) were considered as an integrated system. In order to identify the optimum design of each of the building heating systems, a series of case studies, based on notional groups of buildings, were performed and comparative thermodynamic and economic analyses produced. These case studies comprised combinations of:

- four building types: large multi-functional building (sport facility), single-family homes, multi-family home blocks and small office buildings;
- several district heating substation configurations with cascading space heating loads and/or domestic hot water loads for each building type;
- two climates, Amsterdam and Toronto, for single-family homes, multi-family home blocks and small office buildings; a third climate, Sudbury (Ontario, Canada), for large multi-functional buildings; and
- a combined cycle gas turbine CHP plant with natural gas-fired peaking boilers as a heat production source.

Computer models for the entire DH systems were developed using Simulink[®] software and the heating systems were simulated for an entire year. The results are presented in a series of graphs and tables for each case analyzed. The graphs and tables show the DH temperature differences (flow weighted ΔT), DH water flow, the DH system operation costs as well as revenue generation.

The results showed that for all cases examined, the DH ΔT increased by cascading of the heating loads. However, the magnitude of the improvement in ΔT varied for the different types of buildings.

For the large multi-functional building, there were eight thermal loads requiring different supply temperature levels. This building provided greater opportunities for maximizing ΔT than were present in the other building types. The eight loads (at the design outdoor temperature of -30° C) were:

1)	Pool ventilation air handler - Air Handler 3	230 kW
2)	Fin-tube convectors for perimeter heating	225 kW
3)	Three ventilation air handlers – Air Handler 1	
	(shut off at nighttime)	210 kW
4)	Glycol-based floor heating above parking garage	150 kW
5)	Fan-coil heaters (shut off at nighttime)	120 kW
6)	Floor heating in daycare	45 kW
7)	Pool water heater (constant)	30 kW
8)	Domestic hot water (DHW) for showers, etc.	430 kW
	Grand Total	1440 kW



Figure 1: Case 2 connection scheme for the large multi-functional building

Three district heating substation connection schemes were studied for the large multi-functional building:

Connection scheme 1 (Reference case, Case 1): all heating subsystems were connected in parallel. Domestic hot water was heated in 2-stages: pre-heater and after-heater.

Connection scheme 2 (Case 2 as shown in Figure 1): heating subsystems were cascaded in two levels. Domestic hot water was heated by a pre-heater and after-heater.

Connection scheme 3 (Case 3): heating subsystems were cascaded in two levels. The connection was similar to the Case 2 system except the glycol heating system was placed in the second level to keep the heat demand ratio between the first level and the second level more balanced during both the daytime and nighttime operation.

The simulation results (see Table 1) showed that the cascaded systems (Case 2 and Case 3) have higher DH water temperature difference in all seasons. Overall, the cascaded system improved the Δ T by more than 5°C for Case 3 and 4°C for Case 2. Due to the higher Δ T, the cascaded systems resulted in a lower system flow. The flow reduction was 7.8% and 6% for Case 3 and Case 2 respectively.

Table 1: Seasonal flow-weighted average district heating temperature difference (ΔT) and average system flows for the large multi-functional building in Sudbury

	Sum	nmer	Fa	all	Wir	nter	Spi	ring	Full	Year
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔT		ΔΤ		ΔT		ΔT	
Case1	61.24	-	63.25	-	59.58	-	61.91	-	61.16	-
Case2	63.28	2.04	68.24	4.99	63.19	3.61	66.70	4.79	65.15	3.99
Case3	63.30	2.06	69.41	6.16	64.87	5.29	68.16	6.25	66.46	5.30
	Flow	%	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)		(l/s)	
Case1	0.891	-	1.582	-	3.109	-	1.927	-	1.871	-
Case2	0.862	3.24	1.469	7.16	2.933	5.65	1.791	7.04	1.758	6.06
Case3	0.862	3.25	1.444	8.71	2.858	8.08	1.753	9.00	1.724	7.84
F.W.T. Δ T: Flow-weighted Average Δ T, (°C).										
D.I.: Degree (°C) Improvement relative to Case 1.										
%: Percer	%: Percentage of system flow reduction relative to Case 1.									

The results shown in Table 1 indicated that a significant performance improvement occurred in the fall and spring. In these seasons, a 4.5°C increase in ΔT and more than 7% in system flow reduction for Case 2 was achieved. Case 3 resulted in a 6°C increase in ΔT and approximately 9% in system flow reduction. In the winter, the ΔT improvement was about 3.6°C - 5.3°C and flow reduction was 5.6 - 8%. In the summer, the cascaded systems had smaller performance improvement compared to the other seasons; 2°C increase in ΔT and over 3% in system flow reduction. This was because, in the summer, there were either no or very low loads in the heating subsystems and the effects of cascading were less significant.

It should be noted that the size of all components was kept identical in all three cases. This means that the ΔT of the parallel system would have been smaller than the values shown in Table 1 if the components had been sized for their corresponding supply temperatures at design conditions. In other words, the increase of ΔT and reduction of system flow due to cascading would have been larger. The increase in ΔT is therefore due to cascading <u>only</u>.

The results shown in Table 1 also indicated, except in the summer, that cascaded system Case 3 had better performance compared to Case 2. By placing the glycol heating system in the second level, Case 3 achieved a more balanced power ratio between the first and second level, both during daytime and nighttime. This means that less return water from the first level bypassed the second level in the Case 3 system. Consequently, Case 3 resulted in higher ΔT and lower system flow rates compared to the Case 2 system.

For other types of heating plants, e.g., incineration plants, industrial, and waste heat plants, the economic benefits of cascading might have been larger than observed with the extraction/condensation steam turbine.

For single-family homes, the increase in ΔT , resulting from cascading, was almost insignificant due to the ventilation load (fan-coil load, which operated at the lower temperature level) being small compared to the radiator load. The largest ΔT improvement and system flow reduction occurred in the winter.

For multi-family home blocks, it was found that the increase in Δ T was quite significant by cascading the domestic hot water (DHW) load with the space heating loads. Mixing of DHW recirculation water with DHW, from the pre-heater, also improved the system performance compared to the parallel system where DHW re-circulation water was mixed with the cold municipal water. The largest Δ T improvement (4.3°C for both Toronto and Amsterdam) and system flow reduction (8% for Toronto and 5.8% for Amsterdam) occurred in the summer. The yearly average Δ T increase was 1.2°C for Toronto and 1.4°C for Amsterdam. The yearly average flow reduction was 2.4% for Toronto and 2.5% for Amsterdam. For small office buildings, when the fan-coil to radiator power ratio (at design conditions) was about 0.5, the system performance was improved quite significantly by cascading these two heating loads. The largest Δ T improvement (3°C for Toronto and 1.6°C for Amsterdam) and

system flow reduction (5.8% for Toronto and 3.4% for Amsterdam) occurred in the winter. Flow is

limited during winter, making the DH system capacity expansion possible. Overall, the yearly ΔT increase was 1.7°C for Toronto and 1.1°C for Amsterdam. The yearly average flow reduction was 3.4% for Toronto and 2.2% for Amsterdam.

Systems with different fan-coil to radiator power ratio (FC/RAD=1 & 2) at design conditions were also studied for the small office building cases. The results showed that if the power ratio of the fan-coil to radiator increased, the effects of the cascading were reduced. The Δ T improvement between cases was relatively small for FC/RAD ratios of 1 and 2, although the return temperature decreased significantly.

Due to relatively lower DH return temperatures, the cascaded systems resulted in lower network heat losses, lower pumping energy demand, higher net electricity production and a higher revenue compared to the parallel systems.

It can be concluded from the results of this study, that the overall improvement in ΔT , resulting from cascading different thermal loads, depended on the following three factors:

- required temperature level of the different thermal loads;
- magnitude of these loads; and
- time-of-day usage patterns of the loads.

It is therefore important to keep the above three factors in mind when designing a cascading system.

A system with thermal loads with different temperature levels provides the potential for cascading to maximize ΔT . Heating subsystems requiring a high temperature should be placed in the first level where the systems are supplied by high temperature water. Heating subsystems requiring lower temperature levels can be cascaded to a second level where the return water from the first level can be used. The systems studied in this project were cascaded in two levels. Multi-level (more than two levels) cascading may result in even greater improvement, however more complex control systems may be required.

To achieve maximum improvement in ΔT , the thermal loads in a cascaded system should be arranged in a way that the load ratio between the levels is balanced throughout the year. Usually, the magnitude of the total loads in the upper level should be higher than those in the lower level in order to avoid the use of high temperature water in the second level as much as possible. If the ratio between the upper and lower level is either too high or too low, return water from the first level will bypass the second or high temperature water will be required to supply the second level. Either way, the ΔT decreases.

The time-of-day usage patterns of the different loads should also be considered while designing a cascaded system. Some thermal loads are required in heating season only, such as space heating. Other thermal loads are required throughout the year. Some loads are required only during daytime or nighttime. To achieve the maximum yearly improvement in ΔT , the thermal loads in a cascaded system should be arranged in a way that the load ratio between different levels is balanced as well as possible, both at daytime and nighttime conditions. In practice, this may be difficult to achieve. However, by paying attention to these principles at the design stage, systems will operate much more economically.

The benefits of cascading building heating systems can be realized in both maritime and continental climates. Cascading has its greatest effect when the loads are required for a major portion of the year. Cascading designs that favour wintertime flow reductions facilitate system expansion, since system flow is limiting in the winter.

The district heating substation is considered to be the vital part of the building heating system. In the past, IEA work has been conducted on these stations by Volla, Frederiksen et.al. (1996). In this work, which was partly based upon earlier work done at the Lund Institute of Technology, a graphical representation was used, which in a clear way demonstrated achievable supply and return temperatures for various connection schemes. In conjunction with a detailed analysis, this representation not only permits the designer to predict the performance of specified substations, but can also serve as an intelligent guide when looking for configurations and capacities which are optimal according to specified criteria. In the current project, graphical analyses were carried out for various substations used in the case studies. Results achieved with graphical analyses can thus complement and confirm simulation results.

Chapter 12 of Part II, 'Graphical and Mathematical Analysis of Cascaded Substations' analyzes space heating and domestic hot water connection schemes. For various types of connection schemes with low temperature radiator heating, increased cooling of primary water, due to cascading is indicated, in particular when the domestic hot water provision is equipped with recirculation to compensate for heat losses. It is found that a type of 3-stage connection scheme, termed 'Russian 3-stage' has the potential for particularly low return temperatures. This connection scheme requires rather complicated control equipment and may therefore be practical only in big substations, e.g. in hotels.

Under the idealized assumption of infinitely large heat exchanger surface areas, a table summarizes the analytical expressions derived for the dimensionless primary return temperature for parallel 2-stage and Russian 3-stage (without and with re-circulation of domestic hot water) substations. The analytical derivations confirm the results found in graphical analyses. Connection schemes incorporating domestic hot water storage instead of instantaneous heating were analyzed, mainly on a qualitative basis. It was found that from a thermodynamic point of view, a combination of cascading and hot water storage could be advantageous, even though it may result in rather complicated schemes.

Chapter 13 of Part II deals with cascading of radiator and fan-coil space heating. It is shown that the gain in primary return temperature is very dependent on the size of the fan-coil heat exchanger. With a low-temperature radiator system, this heat exchanger must be rather big for a substantial temperature gain to materialize. With a smaller heat exchanger size, the gain obtained by cascading may even be negative, i.e. simple parallel connection of the two heat exchangers for building heating will be better than cascading.

The effects of various parameter variations were also analyzed. A composite diagram illustrates the optimization of the secondary forward temperature of a cascaded substation to minimize primary return temperature. Also, an example shows that it can be thermodynamically advantageous to let the inducted air temperature fluctuate, instead of keeping this temperature constant by the control of bypasses. This finding is cautioned by the observation that for human comfort reasons, inducted air temperature cannot be allowed to vary within too wide limits.

An additional benefit of the simulation studies was the development of algorithms that greatly speeded up the process, enabling a full year of heating system operation to be simulated in a few minutes. These simulations use two-dimensional table lookup, with the reference table data derived from either fundamental principles or the manufacturer's date.

There is potential for further speedup, leading to the possibility of the simulation programs being used for system design optimization at the routine engineering level.

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1 Introduction

An effective district heating (DH) system has two primary distinguishing features: a low supply temperature and high temperature difference between the supply and return (Δ T). Decreasing the temperature levels in DH systems reduces the heat losses, gives better performance potential of combined heat and power (CHP) plants and provides the possibility for the utilization of waste heat. A high temperature difference between the supply and return also results in lower flows, which means lower pumping energy required for the DH system.

The principle of cascading of loads in building heating systems can be used to increase the temperature difference between the supply and return, ΔT (Snoek, 1999).

The vast majority of present day buildings have multiple heating loads. Dwellings have both domestic hot water (DHW) and space heating needs. Modern, highly insulated and airtight homes have ventilation requirements as well. Commercial buildings have heating and ventilation requirements. The space heating load can be delivered by different systems, for example, radiators, fin-tubes, under-floor heating as well as ventilation/re-circulation air heating combinations.

In addition to these more common buildings, there are several types of buildings with thermal loads requiring different supply temperatures. These buildings can provide even greater opportunities for maximizing the ΔT . Examples of buildings with diverse thermal loads are sport facilities (hot water for showers, lap pool and whirlpool heating, space and ventilation heating), laundries (hot water and space heating), greenhouses, dairies, etc.

The connection of these loads to district heating systems can be made in a variety of ways. The simplest way is to connect all loads in parallel. However, a better heating system can be constructed by cascading system or a system combining series and parallel connections. The latter, a more complex connection method, ensures that the maximum amount of energy is extracted from the DH water before it is returned. A well designed and optimized process therefore, results in a system that can accommodate the lowest supply temperatures, with the lowest possible return temperatures and system flows.

Different building heating system configurations, which have similar performance at peak load (design load), can have significant characteristics at partial load (Van der Meulen 1991). The more statistical data that is available on system performance under different operating conditions, the more accurately energy recoverable from the waste heat of CHP plants and a back-up boiler operation can be predicted. Therefore, long-term time-series simulations are necessary in order to carry out system performance and economic studies of different building heating systems (Snoek 1997 and Onno 1998).

The main objectives of this project are:

- Perform long-term time-series (a typical year) simulations/optimizations of building heating systems connected to a DH system, including heat production plant, distribution network, substation and building heating system; and
- provide guidelines for the effective cascading of different loads of a building heating system to maximize the building temperature difference.

2 Methodology and Definition of Case Studies

In order to identify the optimum design of different building heating systems, it was decided to carry out a series of case studies based on notional groups of buildings. Computer simulation tools were developed to perform the case studies.

A DH system consists of three main components: the heat production plant, the heat distribution system and the buildings to be supplied, and their heating systems. To thoroughly evaluate the thermodynamic and economic performance of different building heating systems connected to a DH system, all three components were taken into consideration as an integrated system.

The following sections describe each of the DH components and define case studies which were agreed to by the Experts Group.

2.1 The Buildings and Building Heating Systems

In this study, it was decided to investigate various building heating system (including substation) designs suitable for:

- large multi-functional buildings with diverse thermal loads;
- single-family homes in medium and high density areas; and
- small office buildings.

The above three types of buildings were considered suitable for connection to a DH system. As mentioned in the Introduction, these buildings have multiple heating loads which provide opportunities for cascading.

Examples of buildings with diverse thermal loads are sport facilities, commercial laundry facilities, greenhouses, etc. These buildings can provide opportunities for maximizing the ΔT by cascading loads. After careful consideration, an athletic facility in Sudbury, Ontario, Canada, was selected to represent the large multi-functional building group. There were many loads in the athletic facility, such as hot water for showers, lap pool and whirlpool heating, space and ventilation heating. In order to determine the profile of these loads, it was decided to monitor the building heating system at different times over a one year period.

After discussions with the Experts Group, an additional building type was considered in this study: multi-family homes (a block of single-family homes) using a centralized DH substation. The space heating demand and domestic hot water usage profile of this type of building would be very similar to apartment buildings. There is more potential for cascading the space heating and DHW heating in the substation for these types of homes than that for single-family homes. The characteristics of the single-family home was used for the multi-family homes as well.

The designs and information relating to the four selected building types are described in detail in Chapter 3.

It was agreed to by the Experts Group to select two to three different DH substation and building heating system connection schemes for the above four types of buildings. That is:

- three connection schemes for large multi-functional buildings;
- two connection schemes for single-family homes;
- two connection schemes for multi-family homes; and
- three connection schemes for small office buildings.

The connection schemes and designs of the district heating substations and building heating systems selected for the case studies are described in Chapter 4.

2.2 Heat Production Plant and Distribution Network

There is a very wide range of possible heat supply plants. For example, the heat supply plant can consist of:

- Boilers, conventional or condensing
- Single cycle gas-turbine CHP plant
- Combined cycle gas turbine CHP plant
- Steam turbine CHP plant
- Reciprocating engine CHP plant
- Industrial waste-fire boilers
- Incineration plant

Of the above plant options, there are two broad categories: those where the efficiency of the plant is strongly dependent on the DH operation temperatures and those where the efficiency of the plant is less influenced by the temperatures. The combined cycle gas turbine (CCGT) CHP plant and steam turbine CHP plant are among the first category.

The main objective of this project was to compare the performance of different building systems by cascading of heating loads. By considering the objective of this project and in order to limit the number of case studies to manageable proportions, it was decided to select a combined cycle gas turbine CHP plant for the case studies.

In reality, the structure of the DH distribution network is complicated as it may involve hundreds of pipelines with different diameters, heat loss coefficients, flow rates, etc. It obviously would require an enormous amount of effort to build a computer model and a long time to perform these time-series simulations.

Based on previous studies (B¿hm 1981, Frederiksen 1982 and Zhao 1995), it can be concluded that any complex DH network system can be simplified to a simple network (even with one pair of pipes). It was therefore decided to use a single pair of supply and return pipes, a single consumer and a single bypass to describe a complex DH network.

More detailed information on the heat production plant and distribution network used in the case studies are described in Chapter 5.

2.3 Climates

The space heating demand profile depends on the local climate, the building construction material and the times the building is occupied. The most important influence is the climate. It was decided to examine two different climate types: a continental climate such as found in central and Northern Europe and North America, and a maritime climate such as found in Western Europe. The two selected climate data were used in the case studies for single-family homes, multi-family homes and the small office buildings.

An athletic facility in Sudbury, Ontario, Canada, was selected to represent large multi-functional buildings. The climate where the athletic facility was located was used for the case studies of this building.

The detailed climatic information that was used in the case studies is described in Chapter 6.

2.4 Case Studies Analyzed

Having selected the types of buildings, DH substations and building heating systems, climate, heat distribution systems and CHP plant, it was possible to put together a series of case studies to be analyzed in this study. Combining these resulted in 17 case studies as listed in Table 2.1.

Table 2.1: Case studies analyzed

Building Type	Climate	CHP Plant	Number of DH Substation and Building Heating System Connection Schemes
Large Multi-functional Buildings	Sudbury	Combined	
		Cycle	3
Single-family Homes	Maritime	Combined	
		Cycle	2
Single-family Homes	Continental	Combined	
		Cycle	2
Multi-family Homes	Maritime	Combined	
		Cycle	2
Multi-family Homes	Continental	Combined	
		Cycle	2
Small Office Buildings	Maritime	Combined	
		Cycle	3
Small Office Buildings	Continental	Combined	
		Cycle	3

3 Buildings for Case Studies

3.1 Large Multi-functional Buildings

In the year 2000, a district heating system was under construction in Sudbury, Ontario. At the same time, a new YMCA/multi-functional building was under construction. This type of facility has a requirement for heat sources at different temperature levels. These heating loads were of interest from the point of view of potential system optimization by cascading of loads. An agreement was reached between the YMCA and NRCan to add measurement points during construction to identify the characteristics of the different loads, in terms of both magnitude and time of day. This YMCA building was selected for case studies for large multi-functional buildings.

3.1.1 General Building Characteristics

The building has 9,300 square metres and consists of four main areas:

- YMCA gymnasium and swimming pools
- Hospital rehabilitation wing
- Older adult education and social activities
- Infant daycare

The first area has the largest amount of floor space.

The south facing window area is approximately 100 square metres, and therefore the effect of solar gain was quite small, relative to the other thermal loads. Solar gain, therefore, was not considered in the simulations.

3.1.2 Building Ventilation Systems

A number of air handlers and fan-coils were used to ventilate the building. With the exception of one air handler, the systems mix fresh ventilation air with re-circulated building air. Under cold weather conditions, the fresh air ratio was in the range of 20 to 30%, with the low value being used at the coldest conditions. This minimized the risk of freeze-up of the coil.

The exception to the mixed re-circulation air system is the change room air supply. A heat pipe heat recovery unit is used with 100% makeup of fresh air.

3.1.3 Methodology to Establish Loads

During construction, fittings were added to critical locations in the heating system for 10 insertion probes to measure supply/return temperatures for different loads. These probes were also equipped with self-heated elements to obtain an indication of the magnitude and dynamics of the water flow.

The main building heat meter was equipped with instrumentation to record energy and water volume. The outdoor temperature was also recorded.

In the case of some loads, it was not possible to use insertion probes and pipe surface temperatures were measured under the insulation. It was also possible to extract useful data from the building's monitoring system.

Data were recorded for a number of months, including both winter and summer conditions.

The four main facilities operate at different time schedules:

	Weekdays	Weekends
YMCA	05:00 - 21:45	07:30 - 20:45
Hospital rehab.	07:30 - 18:00	Not occupied
Older adult centre	06:30 - 18:00	09:30 - 15:00
Infant daycare	06:00 - 23:00	Not occupied

When the building is not occupied, the ventilation systems are turned off, with the exception of the pool ventilation system. The perimeter heating system is maintained at its normal operating temperature. As a result, the building has very little temperature variation from occupied to non-occupied conditions.

In the case of domestic hot water for showers, etc., the load is at a very low level at night. During the night period, the hot water load consists of cleaning staff usage, supply of fresh water to the pools and maintaining the temperature of the re-circulation loop.

Since the major heat load of the building is in the YMCA section (probably 70 to 80% of the total), with a usage pattern that is not significantly different on the weekend, it was decided to simplify the analysis by assuming that the overall building facility operates on an identical sevenday cycle.

3.1.5 Day/night Total Loads Versus Outdoor Temperature

The heat meter energy data were used to compute total instantaneous power levels. Since the power is very dynamic in nature, it was filtered with a 30 minute running average.

To obtain an overview of the total day/night loads, the data were sorted into two groups - 02:00 to 04:00 and 14:00 to 16:00. The mid-afternoon period was selected since the building heat loss would be reasonably stable at that time. Also, the DHW load is at one of the lower points during this time, making it easier to compare the day/night conditions.

Over 3000 data samples of total power level were processed. Since the data are not uniformly distributed against the outdoor temperatures, the samples were preprocessed by binning them into intervals of 2°C and computing the averages for these intervals before fitting a graph to the results.

For the mid-afternoon data, 80 kW was subtracted from the data to partially eliminate the effects of DHW and allow a comparison of day and night loads under similar conditions. The results are shown in Figure 3.1. The 80 kW value is approximately the afternoon DHW load under summer conditions. By subtracting this value, the total power for both day and night conditions during warm weather converges to the same value. This residual value is composed mainly of the loads to heat the pool water, to ventilate the air in the pool and maintain the pool air temperature close to 30°C.

The night load is seen to be considerably lower than the day load. It should be noted that the night load has a slight downward curvature at a lower outdoor temperature. This is mostly due to the lower values of fresh air ventilation with the extremely low temperatures. The same effect is not evident in the day load. This is probably due to the DHW load being somewhat higher in cold weather, with more people using the facilities in the winter. Only a constant 80kW has been subtracted from the day load, while the DHW load in the winter during the 14:00 to 16:00 period would typically be greater than 140 kW.

In order to simplify the analysis, a straight line fit has been used in both cases.

This gives the following peak loads, extrapolated to -30°C:

```
02:00 to 04:00 - 680 kW
14:00 to 16:00 - 1010 kW
```



Figure 3.1: Total building power under day/night conditions versus outdoor temperature

The effect of ventilation on the building load is quite significant, resulting in a power reduction of more than 30% during unoccupied periods. This shows the potential for efficiency improvements in energy use by application of heat recovery to the building ventilation systems. However, this potential improvement is not part of the present study.

3.1.6 Building Loads

Even though loads such as building ventilation can have a nonlinear component, it was assumed that this could be neglected and the load simulated with a linear function. Examination of the other loads indicated that most of them could also be approximated by a linear function without significant errors.

The loads will differ in their zero power intercept as a function of outdoor temperature. For example, the pool ventilation system will have an intercept at about 30°C, due to the requirement for higher air temperature.

There are two exceptions to the assumptions of linear functions for heat loads:

- The glycol floor heating system above the parking garage is turned off at an outdoor temperature above +4°C.
- The swimming pool water heating requirement will be close to constant, assuming that the ground temperature below the pool does not change significantly through the year.

It was not possible to directly measure the power level in each heating subsystem. In some cases, it was not possible to insert a probe in the line at the correct point. In other cases, the flow indicating probes appeared to give higher than expected values, well above the level at which it had been possible to calibrate the probes.

However, most of the loads could be established to reasonable accuracy by using a combination of NRCan instrumentation and data from the building monitoring system. This was true for the air handler systems, which constituted close to half the building heating load. By subtracting the known loads from the total and using information regarding loads that were turned off at night, it was possible to estimate a detailed breakdown of the loads. This breakdown is shown in Figure 3.2.



Figure 3.2: Breakdown of heating subsystem loads versus outdoor temperature

The loads are shown in order of magnitude, with the load stated at an outdoor temperature of -30°C.

1)	Pool ventilation air handler	230 kW
2)	Fin-tube convectors	225 kW
3)	Three ventilation air handlers (total)	210 kW
4)	Glycol-based floor heating system	
	above parking garage	150 kW
5)	Fan-coil heaters (total)	120 kW
6)	Floor heating in day care	45 kW
7)	Pool water heater (constant)	30 kW
	Total	1010 kW

The internal gain was implicitly included in the load curves shown in Figure 3.2.

3.1.7 Domestic Hot Water Load

The domestic hot water system is the highest instantaneous peak load, due to its highly dynamic nature. It can be over 400 kW, though usually it is in the range of 250 to 300 kW. Figure 3.3 shows its variability through a typical week, with the data filtered using a running average of 30 minutes.



Figure 3.3: Typical DHW load profiles with 30 minute filtering

Day 41 is a Sunday, with a shorter period of activity. The Saturday load is similar to a typical weekday, though it appears somewhat higher in level. The assumption is that the average of the Saturday and Sunday loads are sufficiently close to a typical weekday so that the same reference load profile could be used for all days of the week.

To select a reference DHW load profile, it was necessary to apply heavier filtering to eliminate random fluctuations. A 60-minute running average was applied and the load profile of Figure 3.4 was selected as being most representative of the data.



Figure 3.4: Reference DHW daily profile with 60 minute filtering

The filtered data shows the typical peaks and valleys, with four peaks representing the active periods: - before work - noontime - before and after supper.

The actual DHW load profile for the same day is shown in Figure 3.5, and is much more dynamic. This was the reference day used for the simulations.

The typical number of people using the YMCA facility in February 2002 during a weekday was about 800. During the summer, it is expected to be somewhat less.



Figure 3.5: Reference day for DHW simulations

3.2 Single-family Homes

A single two-story house was considered representative of living conditions that can be found in many countries. The front side of the house is shown in Figure 3.6.



Figure 3.6: Front side of the single-family house

On the ground floor level, the house contains a living room connected to an open kitchen and three smaller rooms for bathing, sleeping and storage respectively. The rooms on the first floor can be reached from the entrance hall, via the stairs to a landing. The main dimensions and an indication of the heating system can be found in Figure 3.7.

The construction of the outer walls of the house and the thermal properties of its main parts are shown in Figure 3.8. The outer and inner layers of the outer walls are made of brick. The outer layer contains small openings, in order to keep the cavity dry by natural ventilation.

Each house has a DH substation. The house was heated by radiators. The radiators were provided with a thermostatic valve for room temperature control. A mechanical ventilation system with a heat recovery unit was used for house ventilation. The ventilation air flow was approximately 0.5 air changes per hour.

The heating system was able to meet the design temperature of 20° C in the various rooms in the house. The building heat loss was 14.5 kW (at an outdoor temperature of -10° C and a room temperature of 20° C) for Amsterdam, the Netherlands and 18.5 kW (at an outdoor temperature of -18° C and a room temperature of 20° C) for Toronto, Canada.



Figure 3.7: Schematic plan of the single-family house.



Figure 3.8: Construction of the outer walls and thermal properties of the main parts of the single-family house

3.3 Multi-family Homes

It was decided to use a block of single-family homes described in the previous section to represent multi-family homes. Unlike the single-family homes, the multi-family homes use a centralized DH substation.

3.4 Small Office Buildings

A "Nuon" office building in the Netherlands was selected for the small office building case studies. The office was built as a rectangular block, with a ground floor, a first and a second floor. The utility room was placed on the roof and contained the air handling unit. The ground floor was built directly on a slab, without cellar or crawl space.

The building was 48m long by 14.4m wide. The total height was 10.5m (see Figure 3.9). The total office room floor area was $1729m^2$. The floor area of the corridors, stairs, washrooms, etc. was 414 m^2 .

All parts of the building were provided with a false ceiling, except the stair cases. Height of the plenums above the false ceiling was about 35cm.

The windows of the building consisted of plain double glazing, without any coating, in aluminum frames. The U-value of the glazing was $1.8 \text{ W/m}^2 \text{ K}$. The solar transmissivity was not known, but can be assumed to be 0.7. On the outside, the windows were provided with a solar protection system. Other U-values can be found in Table 3.1.



Figure 3.9: Main dimensions of the office building

Table 3.1: Thermal properties of main parts of the office building

Part	U
	(W/m²K)
Outer walls	0.40
Double glazing windows	1.80
Inner wall 1	2.40
Inner wall 2	0.50
Inner door	3.00
Ground floor	0.40
First and second floor	2.80
Roof	0.40

Radiators were used for perimeter heating. An air handing unit with a heat recovery unit was used for building ventilation heating. The heating system was able to meet the design room temperature of 20°C. The ventilation air outlet temperature was designed for 25°C. Domestic hot water was not provided by the district heating system.

Steam humidifying was activated if the relative humidity in the office rooms was less then 40%. This might occur during the cold winter period.

Fresh air entered the air handling unit at a rate of $10,000m^3/hr$. The air entered the duct system with a temperature of $20^{\circ}C$. After the duct system, the air entered the corridors and office rooms via grates in the false ceilings.

The total building heat losses at an outdoor temperature of -10° C (Amsterdam, the Netherlands) were 225kW, of which the transmission heat loss was 108 kW and ventilation heat loss was 117kW. At an outdoor temperature of -18° C (Toronto, Canada), the total building heat losses were 280kW, of which the transmission heat loss was 137kW and ventilation heat loss was 143kW.

4 District Heating Substations and Building Heating Systems for Case Studies

4.1 Substations and Building Heating Systems for the Large Multi-Functional Building

As described in Chapter 3, there are eight heating loads in the Sudbury YMCA building. Based on the required loads and temperature levels, three configurations were selected for the case studies. They are shown in Figures 4.1 to 4.3. DH primary and secondary flows are shown in the figures. For clarity, the tertiary water/air flows and controls are not shown.

Figure 4.1 shows the connection scheme for the Reference System, Case 1. In this system, space heating, ventilation heating and pool heating are connected in parallel. Air Handler 1 is the combination of the three ventilation air handlers as described in Chapter 3. Air Handler 3 is for the pool ventilation. Control valve Va2 adjusts the DH primary flow rate to the heat exchanger HE2 to control the secondary forward temperature of the system. This secondary forward temperature varies according to the outside temperatures.

DHW is provided by 2-stage heating. All or part of the primary return water from the heat exchanger HE2 is further cooled by incoming cold municipal water before being returned to the district heating network. Three-way control valve Va3 is used to bypass excess water from the space heating system to prevent the DHW from becoming overheated.



Figure 4.1: Case 1 connection scheme for the large multi-functional building

For Case 2, the space heating system has been arranged as shown in Figure 4.2. The glycol heating system, Air Handler 3, the fin-tubes and fan-coils comprise the first level. Air Handler 1, pool water heating and floor heating systems (which require lower temperatures) were placed in the second level.



Figure 4.2: Case 2 connection scheme for the large multi-functional building

The return water (secondary side) from the first level is further cooled in the second level. High temperature secondary water will be drawn (through valve Va5) to the second level if the return water from the first level cannot meet the heat demands. During cold winter days, excess return water from the first level may bypass the second level through valve (Va4). If the first level exit temperature from an individual heating component in the first level is too low to support heating the second level, the return water will bypass the second level through bypass valve Bp1, Bp2, etc. The bypassed water (if there is any) is mixed with the return water from the second level and is returned back to the heat exchanger HE2.

The Case 3 connection scheme is shown in Figure 4.3. In this configuration, the glycol heating is moved to the second level. There are two reasons for this move:

- As the glycol heating loop is used to heat the floor above the garage, the required glycol water temperature is not very high.
- The pool and floor heating loads are quite low while the Air Handler 1 is turned off during the night. Therefore, the second level heating loads are considerably lower than the first level during the night for the Case 2 configuration. It was expected that Case 2 could be improved by moving the glycol heating to the second level. By doing so, it was expected that the nighttime heat demand ratio between the first level and second level would improve.

The control of this system was similar to that of Case 2.



Figure 4.3: Case 3 connection scheme for the large multi-functional building

4.2 Substations and Building Heating Systems for Single-family Homes

In the case of single-family homes, the simulations focused on the effects of cascading radiator and fan-coil space heating systems in the substations. Two cases were studied: a first case where the two systems for space heating were connected in parallel; and a second case where they were connected in series.



Figure 4.4: Case 1 connection scheme for single-family homes

Figure 4.4 shows the connection scheme of Case 1, which can be viewed as a quite conventional type of connection scheme, except for the type of fan coil control, as will be explained below. Incoming district heating water from the forward line, DHF, is cooled in the substation and returned to the network return line, DHR. In the substation, the primary water flow is divided into two parallel flows that are being cooled separately in heat exchangers HE1 and HE2. Control valves Va1 and Va2 continuously adjust each of these two flows according to demands on the secondary side of these heat exchangers.

Sometimes, hot water is being drawn directly from a district heating network, instead of being produced as heated municipal water. Also, space heating direct connection is sometimes used instead of indirect connection via a heat exchanger. In fact, such more simplified solutions in some countries are particularly common in single-family homes, where the requirement to reduce first costs is strong. On the other hand, schemes with heat exchangers are not uncommon either. Which solution to choose is a much debated theme that is outside the scope of this report. The choice made here was to use heat exchangers.

Another basic choice of technology which is implicit in the Case 1 scheme and all other schemes of this report is the choice of instantaneous domestic hot water heating without hot water storage. Many types of storage solutions have been shown to result in poor primary water cooling. On the other hand, there are more modern solutions available in which a storage tank is being charged from an external heat exchanger, solutions which are capable of providing good cooling of primary water. Storage solutions were not used in this study.

Valve Val is shown to control outgoing hot water temperature in a simple feedback loop. This thermostatic control is sometimes made more sophisticated by combining the feedback with a feed forward loop in such a way that variations in hot water flow rate will affect the position of the valve directly. This is in addition to the indirect influence given by the thermostatic feedback. Such a sophistication will speed up the control and can be used to reduce dynamic variations in hot water temperature.

Valve Va2, which controls the flow rate to heat exchanger HE2, is a feedback loop governed by controller C. Both the temperature of the secondary water flow going out from HE2 and the outside air temperature are monitored and transmitted to the controller. This unit is programmed to adjust the set point of the outgoing secondary temperature to changes in heat load, so that at lower outside air temperature, the outgoing water temperature goes up. The desired temperature - temperature functional relationship can be programmed into the controller as a certain load curve.

To some extent, this curve can be selected freely. However, as has been shown in a previous IEA study (Volla et al. 1996), for static load conditions, at each outside air temperature, a certain choice of outgoing, secondary water temperature will produce the lowest primary return temperature to the district heating network. The present study will build on this result. Calculations were made to derive optimal load curves for Case 1 of Figure 4.4 and for the other schemes to be presented below.

For this load curve to function in an optimal way, it is essential that radiators are equipped with thermostatic control. That is, the radiator heat rate (to keep a certain constant indoor air temperature at varying heat load) is determined by the radiator thermostats, not only by the secondary forward water temperature. Thus, it becomes possible to select this temperature to optimize the primary return temperature. Since an optimal load curve implies that the forward temperature increases at increasing heat load, the load curve gives an approximate control of the radiator heat rate, while the thermostats provide final/fine control of the heat rate.

A pump (Pu) maintains water circulation in the space heating circuit. By variable speed control, the differential pressure of the circuit is kept constant at heat exchanger HE2.

Water flow to the fan-coil (FC) is controlled by a thermostatic control loop, by which the position of Valve Va3 is adjusted in a thermostatic feedback loop to maintain a constant temperature of air distributed from the fan-coil.

Fresh air drawn into the fan-coil is preheated in a heat recovery unit, HR. This saves energy and provides freeze protection to the fan coil. Theoretically, without heat recovery, a substantially lower primary return temperature (occasionally even below the freezing point in some winter load conditions) could be achieved at the expense of a bigger energy consumption and a need for special precautions to avoid freezing.

The type of fan-coil load control chosen here can be termed water flow control, as opposed to the type of control mostly adopted in current ventilation practice in a number of countries: temperature control. In the latter case, the water circuit at the fan-coil is equipped with a pump. Return water is admixed to the water flow supplied to the fan-coil, the ratio of the two flows being adjusted to provide a thermostatic control of the distributed air. By such temperature control, one can prevent occasional low water flow rate in the fan-coil, which represents a risk of freezing. On the other hand, temperature control represents a thermodynamic loss, in the sense of the Second Law of Thermodynamics due to the mixing of flows of different temperatures, which will result in exergy losses and a somewhat higher primary return temperature. Therefore, flow control is preferred here.

For practical application of the results of the present investigation, the risk of freezing should be considered carefully, especially for applications in severe winter climates. One possibility could be to add an optional temperature control in such way that flow control is normally adopted to achieve the lowest possible return temperature. The temperature control function will only then come into operation if either a selected water temperature or a selected cold air temperature falls below a certain value.



Figure 4.5: Case 2 connection scheme for single-family homes

Figure 4.5 shows Case 2 of the single-family home simulation. Here, radiator and fan-coil heating units are connected in series instead of being connected in parallel. Bypasses BP1 and BP2 are provided to compensate for load - temperature mismatching between radiators and fan-coil. Thus, if the water flow and the return temperature leaving the radiator circuit are too low for the desired air temperature from the fan-coil to be maintained, water of higher temperature will be directly supplied to the fan coil through bypass BP1. Likewise, when the air temperature distributed from the fan-coil becomes too high, thermostatic bypass control BP2 will lead to an appropriate water flow in bypass of the fan-coil.

4.3 Substations and Building Heating Systems for Multi-Family Homes

Figure 4.6 shows Case 1 of the simulation of systems for multi-family homes. As in the scheme of Figure 4.4, the hot water heating circuit, fan-coil and radiators are all connected in parallel. The only difference is that in Figure 4.6, the domestic hot water circuit has been supplemented by a recirculation line, HC.



Figure 4.6: Case 1 connection scheme for multi-family homes

Such hot water circulation is commonly used in modern, multi-family buildings to help maintain the temperature of hot water above a certain level and to compensate for heat losses in the hot water distribution pipes. Without hot water re-circulation, it may take longer for water drawn off from a tap to attain a reasonably high temperature, especially if the tap is installed a long distance from the substation.

In Case 2, shown in Figure 4.7, heating of domestic hot water has been divided into two stages. Both on the primary side (return flow from space heating heat exchanger) and on the secondary side (hot water circulation return), flows are mixed between pre-heater (PH) and after-heater (AH). Such two-stage heating is rather common in some countries, e.g. Sweden.

In two-stage heating, all primary water is being cooled by incoming cold municipal water before being returned to the district heating network. This implies a lower return temperature compared to parallel connection Case 1, where only primary water leaving the hot water heat exchanger meets cold municipal water.



Figure 4.7: Case 2 connection scheme for multi-family homes

4.4 Substations and Building Heating Systems for Small Office Buildings

For small office buildings, it is assumed in the simulations that there is no provision for domestic hot water consumption. In practice, there usually will be some small hot water consumption, for instance for hand washing. Often, however, there will be no baths or showers in office buildings, so the hot water load will be small.

Compared to family homes, offices often require bigger ventilation systems to accommodate more people occupying rooms during working hours, computers, copy machines, etc.

Ventilation air re-circulation in family homes or in office buildings is sometimes adopted to reduce energy consumption. The small office building systems evaluated in this study operated with and without air re-circulation.

Figure 4.8 shows a simple, parallel connection scheme, similar to previous parallel connection schemes.



Figure 4.8: Case 1 connection scheme for small office buildings.

Figure 4.9 shows a serial connection with bypasses, similar to previous schemes with serial connection of radiator and fan-coil space heating.

Case 2a of Figure 4.10 is a simplified version of Case 2, omitting by-passes in serial connection, allowing the distribution temperature of air leaving the fan-coil to fluctuate freely. This case is interesting, since omission of either one or both bypasses will reduce first costs. One might identify cases of applications of Case 2a where distributed air temperature will only vary moderately. In particular, somewhat higher air temperatures than ideally prescribed can be regarded as acceptable.



Figure 4.9: Case 2 connection scheme for small office buildings



Figure 4.10: Case 2a connection scheme for small office buildings
5.1 Heat Production Plant

As stated in Chapter 2, a combined cycle gas turbine was chosen as the heat production plant for all the case studies carried out in this project.

A combined cycle gas turbine CHP plant can take two principal forms: using a back-pressure steam turbine or an extraction/condensation steam turbine. The back-pressure steam turbine has the advantage, due to its simplicity, of being cheaper in installation than an extraction/condensation steam turbine. However, the operation of the back-pressure units is limited by the existence of a minimum heat demand that might not exist all the time. The extraction/condensation cycle presents a thermodynamically more advantageous way of achieving the flexibility required by a DH system, since during low heat demand periods it can produce electricity with maximum efficiency. In this way, the generating system can be kept operating regardless of the heat demand. In this study, a single extraction/condensation cycle was used.

In most CHP/DH schemes, the CHP plant only supplies part of the load. A peak and stand-by plant consisting of boilers supply the remaining load. In this study, a natural gas-fired peaking boiler was used to cover the peak loads.

A small CHP plant was chosen for this study. The small size implied that individual plants had no real effect in the electric power network load and did not take part in regulating the total power production. Another characteristic was that the main concern for the plant operator was to fulfill the heat demand. The capacity of the plant used in the study was assumed to be in the range of 20 to 35 MW_e. The choice made here was quite arbitrary. The electrical efficiency would be higher if a larger capacity was chosen. However, the capacity of a CHP plant does not affect the comparison of different building heating systems.

Figure 5.1 shows the flowchart of a combined cycle gas turbine plant with a single extraction/condensation steam turbine used in the studies.

With other kinds of heating plants, such as waste incineration plants, industrial waste heat plants, etc., the economic benefits of lowering the return temperature might have been greater than observed with the extraction/condensation steam turbine.



Figure 5.1: Flowchart of a combined cycle gas turbine CHP plant with a single extraction/condensation steam turbine

Air at a temperature of T_{g1} and pressure of P_{g1} was compressed in a compressor to a temperature of T_{g2} and pressure of P_{g2} . The compressed air and natural gas were injected into a combustion chamber and combusted. The exhaust gas was then expanded in a gas turbine. The expanded gas entered a heat recovery steam generator (HRSG), where part of the heat in the exhaust gas was recovered to produce steam. In order to further utilize the heat in the exhaust gas, an economizer was used to decrease the stack gas temperature by heating the DH return water.

The steam generated in the HRSG was used to produce additional electricity with a steam turbine. Part of the steam was extracted from the steam turbine to heat the DH water in a condenser. The remaining steam was further expanded in the steam turbine to the condenser pressure level.

The district heating return water temperature was first heated in the economizer and then in the DH condenser by the steam extracted from the steam turbine. If the DH water temperature leaving the condenser was lower than the required level, it was further heated by a peaking boiler.

It should be noted that no attempts were made here to perform an optimal design of the CHP plant as it was beyond the scope of this study.

5.2 Distribution Network

In reality, district heating distribution networks are very complicated. If there are over several hundred or thousand consumers connected to a DH network, it will require an enormous amount of effort to build up a mathematical model. Therefore, it was desirable to have a simplified network to describe the basic dynamics of the complex system.

A model with a single pair of supply and return pipes, a single consumer and a single bypass was developed by Bøhm (1988) for steady state analyses of DH networks. Also, Frederiksen (1982) applied a steady state model of a single pair of pipes.

The simplified model, referred to as an equivalent model, can be generated by gradually reducing the topological complexity of the original network (Hansson 1990, Zhao 1995). During this reduction, the relevant model parameters of the network are transformed in such a way that the dynamic behavior of the equivalent network will resemble the original one. The simplified DH network model includes the description of the critical points (the most unfavorable consumers) of the network and the aggregated sub-network models. The time delay of the equivalent network is equal to the average time delay of all consumers weighted by the consumption or flow of each consumer. The heat loss of the equivalent network is equal to the heat loss of the real network.

Zhao (1995) verified the aggregated model by comparing simulation results of several equivalent networks to their original network. The DH system had 535 consumers and 1,079 branches (consisting of pre-insulated pipes without loops). The original system and its equivalent systems with 500, 200, 100, 50, 25, 12, 6, 5, 4, 3, 2 and 1 branch were modeled. The results showed that the number of branches in the equivalent network could be reduced to 10 branches without affecting the accuracy. However, even for the equivalent network with 1 branch, the standard deviation of return temperature was 0.7°C. The maximum absolute difference in return temperature was approximately 2°C.

Based on these previous studies, it can be concluded that any complex DH network system can be simplified to a simple network (even with one pair of pipes). It was therefore decided to use a single pair of supply and return pipes, a single consumer and a single bypass (as shown in Figure 5.2) to describe a complex DH network.



Figure 5.2: A simplified district heating network

It should be pointed out, for simplicity and time-saving reasons, that the length of the pair of pipes used in the case studies was predetermined to present an unknown complex network. It was not derived from a known complex network.

Bypasses usually result in high return temperature. However, in order to keep the supply temperature at the building at a reasonable level at times when there are no or very low building heating loads, a bypass was used in the network model.

6.1 Climate Data Used in Case Studies for Multi-functional Buildings

An athletic facility in Sudbury, Canada was chosen to represent a large multi-functional building. Sudbury weather data therefore was used in the simulations. The Typical Meteorological Year hourly weather data of Sudbury was obtained from the WATSUN Simulation Laboratory of the University of Waterloo, Canada.

The weather files contain Typical Meteorological Year hourly weather data such as dry bulb temperature, dew point temperature, wind speed, wind direction, solar radiation, etc. They were created by concatenating twelve Typical Meteorological Months selected from long term series data. The normal and realistic temperature and radiation variability is therefore included in the data.

Figure 6.1 shows the outside temperature variations over a typical year for Sudbury. To avoid showing the high fluctuations of the hourly and daily data in the graphs, the raw data was averaged over a one week period. The simulations were performed with the hourly data. Solar gain was not considered in the case studies for the large multifunction building. The design outdoor temperature and number of degree-days for Sudbury are shown in Table 6.1.



Figure 6.1: Weekly averaged outdoor temperature in Sudbury

6.2 Climate Data Used in Case Studies for Homes and Small Office Buildings

To compare the performance of building systems in different climate regions, it was decided to consider a typical western maritime climate and a typical continental climate. As described previously, a home and a small office building built in the Netherlands were selected as building examples for the case studies. A city in Holland was considered to represent the maritime climate. Amsterdam weather data was available and was selected for the case studies.

Toronto weather data was selected to represent a continental climate. This selection was based on the studies performed in the District Heating and Cooling Annex V project "Optimization of Operating Temperature and an Appraisal of the Benefits of Low Temperature District Heating" (Woods 1999). It was found that Toronto's weather was more representative of the majority of continental European cities as well as cities in the northern parts of the United States and Canada after comparing it to the meteorological data from Helsinki. The Amsterdam weather data was obtained from the American Society of Heating and Air Conditioning Engineers (ASHRAE) "International Weather for Energy Calculations (IWEC Weather Files)" (ASHRAE 2001). The Toronto data was obtained from the WATSUN Simulation Laboratory of the University of Waterloo, Canada.

Figures 6.2 and 6.3 show the temperature variation over a typical year in Amsterdam, the Netherlands and Toronto, Canada respectively. To avoid showing the high fluctuations of the hourly and daily temperatures in the graphs, the raw data were averaged over a week period. However, the simulations were performed with the hourly data.



Figure 6.2: Weekly averaged outdoor temperature in Amsterdam



Figure 6.3: Weekly averaged outdoor temperature in Toronto

Solar gains were considered in the case studies for single-family and multi-family homes. Figure 6.4 shows the solar gains achieved by the home located in Amsterdam, while Figure 6.5 shows the

solar gains achieved by the home located in Toronto. The solar gains shown in the figures were calculated based on the solar radiation obtained from the Typical Meteorological Year weather data as well as the orientation and layout of the house. The solar gains shown in the graphs are also one week average.



Figure 6.4: Weekly averaged solar gain in Amsterdam



Figure 6.5: Weekly averaged solar gain in Toronto

The design outdoor temperatures and number of degree-days ($18^{\circ}C$ base) for the two locations are shown in Table 6.1.

Table 6.1: Design outdoor temperatures and degree-days for Amsterdam, Toronto and Sudbury

City	Design Outdoor Temp.	Degree Days
	(°C)	
Amsterdam	-10	2834
Toronto	-18	4082
Sudbury	-28	5447

7 Summary of Simulation Models and Assumptions

7.1 Summary of Simulation Models

To perform the simulations of district heating systems, hardware components that have a thermodynamic impact on the DH system have to be available in the form of numerical models. Many models were developed under the IEA Annex V project "Optimization of Operating Temperatures and an Appraisal of the Benefits of Low Temperature District Heating" (Woods, 1999). A combined cycle gas turbine CHP plant model and a simplified distribution network model were developed for this study.

All these dynamic models used in the simulations were developed through the use of the Simulink[®] software by The Math Works Inc. The following sections summarize the simulation models used in this study.

7.1.1 Building Model

In order to accurately simulate a complete district heating system, the building model must realistically respond to steady state and dynamic effects of weather conditions and building occupancy. Since the DH system simulation studies are aimed at discovering the effects of changes to the heating systems while keeping the building envelope constant, high absolute accuracy of any given building model is not necessary. For this reason, the building model developed by Onno (Onno, 1998 and Woods, 1999) was adopted in the simulations.

Figure 7.1 shows the diagram of the simplified building model. A number of simplifications were incorporated to speed up the simulation process. The primary emphasis of the building model was to include all transfer functions that have a significant influence on the characteristics of the various buildings that make a typical community. A detailed model description and verification can be found in Onno (1998) and Woods (1999).



Figure 7.1: Simplified building model

It should be noted that the building model included two thermal masses. However, if the room temperature set-point was kept constant, these different thermal masses did not affect the simulation to any significant extent.

7.1.2 Models for District Heating Substations and Building Heating Systems

A district heating substation is connected to many components. For the DH substations that are described in Chapter 4, hardware components that have thermodynamic impacts are heat exchangers, fan-coils, radiators, fin-tubes and control valves (including controllers and temperature sensors).

The simpler components such as temperature sensors, controllers and control valves were specified directly by a simple transfer function or Simulink[®] "built-in" block. The more complex components such as radiators, heat exchangers and fan-coils were developed based on general thermodynamic relations. The outputs of models for the more complex components were verified by data obtained from a DH heat transfer station at a school, from laboratory experiments, or verified by manufacturers' data. A detailed model description for these components can be found in Woods (1999).

The simulation models of the district heating substations were developed by combining component models together according to their actual configurations.

7.1.3 Heat Production Plant Model

As described in Chapter 5, it was decided to use a combined cycle gas turbine CHP plant with a natural gas-fired peaking boiler as the heat production source. Simulink[®] models for each component shown in the flowchart Figure 5.1 were developed based on the general thermodynamic principles and heat transfer theory. These models are described in detail in Appendix A.

To verify the combined cycle gas turbine CHP plant model, the results from the Simulink[®] model were compared to those from the Simsci PRO/II of the Simulation Science Inc.

Simsci PRO/II is a simulation software used for steady state simulation of refinery processes, chemical processes, batch processes, etc. Besides models for special components used in these processing industries, PRO/II contains models for general components, such as compressors, expanders (turbine), reactors and simple heat exchangers. With these standard block models, it was possible to build a simulation system for the combined cycle gas turbine CHP plant.

The comparison results are also described in Appendix A. The results showed that the deviations between the two models were relatively small and the accuracy of the Simulink[®] model was acceptable.

7.1.4 Distribution Network Model

The network model was used to calculate heat losses from the DH pipes and required pumping power for transporting the DH water. As described in Chapter 5, an equivalent network with a pair of supply and return pipes was used to represent a complex network. Therefore, the distribution network model was simplified to calculate the heat losses and pumping power from the two straight pipes.

It was assumed that the supply pipe and return pipe had the same diameter, equivalent length and equivalent heat loss coefficient. The pressure drops in the supply and return lines were also assumed to be the same. The required electrical pumping energy was calculated as a function of flow, pressure drop and overall pump efficiency. Detailed model descriptions can be found in Appendix B.

7.2 Assumptions Used in the Simulations

In order to perform the simulations of district heating systems, many input data needed to be defined and many assumptions needed to be determined. Also, some simplifications were needed to speed up the simulation process. These input data, simplifications and assumptions (for simplicity, all of these will be called assumptions in the following) used in the simulations are described in the sections below.

7.2.1 General Assumptions

Some assumptions applied to the dynamic simulations, common to all building types, are listed below.

District Heating Supply Temperatures

For case studies of the large multi-functional building (located in Sudbury), it was assumed that the district heating supply temperature varied from 120°C to 90°C at the heat production plant, as a function of outdoor temperature. This supply temperature was similar to what is being used in the Sudbury DH system now.

The supply temperature profile is shown in Figure 7.2. The supply temperature increased linearly from the 90°C base temperature when the outdoor temperature dropped below 15°C. As the outdoor temperature dropped, the system supply temperature change was rate-limited to no more than 2° C per hour. Conversely, as the outdoor temperature increased, the supply temperature could not decrease by more than 0.5° C per hour.

For the case studies of single and multi-family homes and small office buildings, it was assumed that the district heating supply temperature varied from 80°C to 95°C at the heat production plant, as a function of outdoor temperature. Lower DH supply temperatures were assumed for Amsterdam and Toronto due to the milder climate in these two locations compared to Sudbury. These assumed supply temperature variations are quite common in most European countries.

The supply temperature profiles are shown in Figure 7.3 and 7.4 for Amsterdam and Toronto respectively. The supply temperature increased linearly from the 80°C base temperature when the outdoor temperature dropped below 5°C in Amsterdam and below 0°C in Toronto. As the outdoor temperature dropped or increased, the rate-limits assumed for the Sudbury climate were used.



Figure 7.2: Variation of DH supply temperature at the heat production plant with outdoor air temperature for Sudbury



Figure 7.3: Variation of DH supply temperature at the heat production plant with outdoor air temperature for Amsterdam



Figure 7.4: Variation of DH supply temperature at the heat production plant with outdoor air temperature for Toronto

Secondary Supply Temperatures to Space Heating System

As described in Chapter 4, all space heating systems were indirectly connected to the district heating system. The secondary supply temperature to the space heating systems was 88°C at design condition for the large multi-functional buildings and was 70°C at design condition for other building types studied (i.e. for single-family homes, multi-family homes and small office buildings). This secondary supply temperature was optimized as a function of outdoor temperatures, to obtain the lowest DH return temperature from the space heating system.

The secondary supply temperature optimization was done by setting the secondary supply temperature to a value that resulted in the lowest DH return temperature in a climatic steady load condition. This means that at a given outdoor temperature the space heating load was the same even though the load situation will be, in general, dynamic. This also implies that the effect of solar and internal gains on the heating load was not considered in these optimizations. This secondary supply temperature optimization process was very similar to the one carried out in a previous IEA study (Volla 1996).

Variables Considered in the Building Simulations

The selection of variables to simulate each building type was carefully considered. Although the most realistic system would simulate all variables for each building type, such an approach would result in overly complex and lengthy simulations. Table 7.1 summarizes the variables which were taken into account for each of the different case studies.

	Forced	Ventilation Heat			
	Ventilation	Recovery	DHW	Solar Gain	Internal Gain
Large Multi-functional					
Buildings	\checkmark		\checkmark		
Single-family Homes	\checkmark	\checkmark	\checkmark	\checkmark	
Multi-family Homes	\checkmark	\checkmark	\checkmark	\checkmark	
Small Office Buildings	\checkmark	\checkmark			\checkmark

Table 7.1: Variables used in building heating system simulations

The south facing window area of the large multi-functional building was approximately 100 m^2 . Therefore, the effects of the solar gain is quite small relative to the other thermal loads of the system. The solar gain was therefore not considered in the simulations for this building type.

Solar gain was not included in the small office building simulations either. It was considered that the solar gain in the winter was relatively low as the office building is facing the east.

Space Heating Systems

As the connection schemes shown in Chapter 4 indicated, radiator heating systems were used for building perimeters. Fan-coil based hot air distribution systems were used for ventilation in single-family homes, multi-family homes and small office buildings.

Ventilation heat recovery systems were assumed present in the above three types of buildings. For simplicity, the ventilation heat recovery efficiency was selected at a constant 60%, although in reality, it varies depending on the intake fresh air temperature and load. The ventilation distribution hot air temperature was selected at 25°C or higher. This ventilation outlet temperature was somewhat higher than that of some European countries, but it was considered quite common in North American practice.

The building interior temperature was kept at 20°C constant.

For the large multi-functional buildings, as described in Chapter 3, there were seven space and ventilation subsystems. Assumptions used in these different heating systems will be described in Section 7.2.2.

Domestic Hot Water System

The domestic hot water supply temperature was controlled at 60°C. This relatively high DHW temperature was used to avoid the possibility of Legionella bacterium growth.

The cold municipal water supply temperature was varied, depending on the season, between 5° C and 12° C in Amsterdam as well as in Toronto, and between 3° C and 10° C in Sudbury.

Heat Production Plant

A combined cycle gas turbine CHP plant with a single extraction/condensation steam turbine was used as the heat production source. The CHP plant was designed to cover 40% of the DH peak load. A natural gas-fired boiler was used to provide peak heat demand.

The capacity of the CHP plant was in the range of 20-35 MWe. The choice made here was quite arbitrary. The electrical efficiency would be higher if a larger capacity was chosen. However, the capacity of the CHP plant does not affect the comparison of different building heating systems.

The fuel rate to the combustion chamber was assumed constant. It was assumed that the feed water flow to the heat recovery steam generator (HRSG) was constant as well. This feed water flow rate was derived based on the criteria that the minimum flow at the steam turbine condensation tail was about 10%. It should be noted that the optimization of the CHP plant design and operation was beyond the scope of this study.

It was assumed that the pressure in the HRSG was constant (20 bar) and the outlet steam temperature from the generator was constant (300°C) as well.

The extraction steam pressure was assumed at 3 bar for the large multi-functional buildings and 1.5 bar for the remaining three building types. The higher extraction pressure was used for the large multi-functional building due to its requirement of higher DH supply temperatures. The steam exit pressure at the steam turbine condensation tail was assumed at 0.05 bar.

The condensed water leaving the DH condenser was assumed to be 5°C higher than the district heating water inlet temperature.

Distribution Network

It was assumed that the supply pipe and return pipe have the same diameter, equivalent length and equivalent heat loss coefficient. The pressure drops in the supply and return lines were also assumed to be the same.

A bypass was used to keep the DH supply temperature at the substations above 70°C in cases where there was either no or very low heating loads and consequent low flows.

Simulations

All simulations, except for small office buildings, were performed for an entire year starting from January 1.

For small office buildings, as there were no DH loads in the summer, the simulations were performed for a typical heating season: starting on September 1 and ending on April 30.

7.2.2 Assumptions Used in the Simulations of Large Multi-functional Buildings

Case Studies

As described in Chapter 4, three district heating connection schemes were selected. The connection scheme shown in Fig 4.1 is called Case 1. The cascaded connection schemes shown in Figure 4.2 and Figure 4.3 are named Case 2 and Case 3 respectively.

Building Heating Subsystems

Besides the DHW heating system, the large multi-functional building has seven heating subsystems:

- 1) Pool ventilation air handler (Air Handler 3)
- 2) Fin-tube convectors
- 3) Three ventilation air handlers (Air Handler 1)
- 4) Glycol-based floor heating system above parking garage
- 5) Fan-coil heaters
- 6) Floor heating in daycare
- 7) Pool water heater

The design loads and temperatures for the above subsystems are summarized in Table 7.2. After discussions with the Experts Group, it was decided to keep the component sizes the same for all case studies. The main reason for this decision was to provide a comparison between all cases resulting only from the effect of cascading. For the same reason, the CHP plant capacity and the distribution network sizes were kept the same for all cases.

Table 7.2: Design heating loads and temperatures for different subsystems in large multifunctional buildings

		Design Temperatu	Design Temperatures		
	Design Load (kW)	Hot Side	Cold Side		
		(°C)	(°C)		
Air Handler 3 (Pool Ventilation)	230	88/82	12*/35		
Fin-tube Convectors (total)	225 (87 units)	88/75	20 (room temp.)		
Air Handler 1 (total)	210 (2 units)	40/8	6**/18		
Heat Exchanger for Glycol Floor Heating					
System	150	60/38	35/50		
Fan-coil Heaters (total)	120 (12 units)	88/56	10***/20		
Floor Heating	45	40/30	-		
Pool Water Heater (constant)	30	40/35	27/30		
(Total)	(1010)				
Notes:					
* : 30% of fresh outdoor air and 70	0% of re-circulation air, co	nstant air flow rate.			
** : 25% of fresh outdoor air and 7	5% of re-circulation air, co	nstant air flow rate.			
*** · 20% of fresh outdoor air and 8	0% of re-circulation air co	nstant air flow rate			

The design values listed above were based on the Sudbury YMCA heating system.

One of the three ventilation air handlers used in the Sudbury heating system covers 50% of the load and the other two air handlers cover the remaining loads. For simplicity, two identical air handlers were used in the simulations. Similar to the actual Sudbury heating system, the air handlers were oversized. Also, the air handlers were shut down at 10PM and turned on at 5AM the next morning.

Different fin-tube and fan-coil sizes were used in the Sudbury heating system. To simplify the simulations, it was also assumed that the heating loads were supplied by a number of identical fin-tubes and fan-coils. As with the ventilation air handlers, the fan-coils were turned off at 10PM and turned on at 5AM the next morning.

It should be mentioned that the heat exchanger for the glycol floor heating system used in Sudbury was undersized. In order to make the simulations more appropriate, the heat exchanger size was modified.

For the floor heating in the daycare, the water circulation flow in the under floor pipe was constant and the return water temperature from the hot side was controlled at 30°C.

The load for pool water heating was 30 kW and was constant over the year. The heat exchanger was considerably oversized in order to heat the pool water to the required temperature level in a short period of time. The return water temperature from the hot side was controlled at 35°C.

The size of the heat exchanger, HE2 (shown in figures 4.1 through 4.3), was selected based on the following design temperatures:

Primary side:	118°C / 75°C
Secondary side:	70°C / 88°C

The building design load was assumed at 1010 kW. The heat exchanger used in the simulations was oversized by 20%.

The heating system of the large multi-functional building is quite complex since there are so many subsystems involved. In order to simplify and increase the speed of the simulations, it was decided to develop a set of characteristic curves for each individual component except for heat exchanger HE2. The water flow rate and return temperature curves (as a function of power output and supply temperature) were generated by using the complex component model developed under the IEA Annex V project (c.f. Wood (1999) and section 7.1.2). Figures 7.5 and 7.6 show an example of the characteristic curves for a fin-tube with a length of 2.5 meters. The return temperature and flow rate were obtained by using Simulink[®] two-dimensional table look-up block.



Figure 7.5: Characteristic curves of a fin-tube – return temperatures

Domestic Hot Water System

The domestic hot water load profile, as shown in Figure 3.5, was used in the simulations. The monitoring data at the Sudbury YMCA building showed that the DHW consumption was somewhat less during the summer. In the simulations, the DHW power levels were assumed to gradually reduce to 70% of the winter values in the summer period (July and August). The DHW re-circulation flow was assumed to be constant (0.48 l/s). The circulation line had an assumed heat loss of 10 kW and a resulting temperature drop around the loop of 5°C.



Figure 7.6: Characteristic curves of a fin-tube – flow rates

The DHW pre-heater and after-heater have the same size. Their size was determined according to the following design temperatures when there were no space heating loads:

Primary side:	70°C / 25°C
Secondary side:	3°C / 60°C

The design DHW load was assumed at 430 kW including the DHW re-circulation loss.

Number of Multi-functional Buildings Connected to the DH System

For simplicity, it was assumed that there were 45 identical large multi-functional buildings, i.e. 45 DH substations, connected to the CHP plant. This number was derived by assuming the CHP plant capacity to be in the range of 20-35 MW_e . As stated earlier, the CHP plant covered about 40% of the DH peak load.

In reality, a DH system usually supplies an area with mixed types of buildings. However, the emphasis of this study was to compare the thermodynamic performance of different DH substations and building heating systems. It was decided that the above simplification would still result in valid comparisons between different systems.

7.2.3 Assumptions Used in the Simulations of Single-Family Homes

Case Studies

As described in Chapter 4, two district heating substation connection schemes were selected. The parallel connection (Figure 4.4) was defined as the reference case (Case 1). The cascaded system (Figure 4.5) was named Case 2.

Building Heating Load

Table 7.3 summarizes the design heating load and heat loss factors for the corresponding design outdoor temperatures in Amsterdam and Toronto.

	Amsterdam	Toronto
Design Outdoor Temperature (°C)	-10	-18
Design Heating Load (kW)	14.5 (total)	18.5 (total)
Heat Recovery Unit (kW)	1.8	2.3
Fan-coil (kW)	1.7	2.0
Radiator (kW)	11.0	14.2
Heat Loss Factor (kW/°C)	0.483	0.483

Table 7.3: Design heating loads and heat loss factors for single-family homes in Amsterdam and Toronto

As described in the previous section, the room temperature was selected at 20°C and ventilation air temperature was assumed at 25°C. The efficiency of the heat recovery unit was selected at 60% constant. The ventilation air flow rate was assumed to be constant at 0.083 m³/s, which is approximately 0.5 air change per hour for both climates.

Domestic Hot Water Consumption

The DHW consumption was assumed at 238 l/day/house. The DHW use profile shown in Figure 7.7 was used in previous work for the IEA DH&C Annex V, (see Woods 1999). The peak domestic hot water load was 19 kW. This is an average peak power of a typical single-family house, with some diversification included.



Figure 7.7: Domestic hot water consumption profile for one single-family home

Design Temperatures for Heating Components

Similar to the large multi-functional buildings, it was decided to keep the component sizes the same for all case studies. This means that the size for the heat exchangers, radiators, fan-coils and control valves were identical for Case 1 and Case 2. The main reason for this decision was that it provided the possibility to compare the differences between the two cases, resulting only from the effect of cascading. For the same reason, the CHP plant capacity and the distribution network sizes were kept the same for Case 1 and Case 2.

Table 7.4 summarizes the design temperatures for the heating components used in the single-family homes.

Table 7.4: Design temperatures for different heating components in single- family homes

	Primary Side (°C)	Secondary Side (°C)
Heat Exchanger for Space Heating	90/40 (Amsterdam)	35*/70 (Amsterdam)
	90/40 (Toronto)	35*/70 (Toronto)
Radiators	70/40	20 (room temperature)
Fan-coils	40/30 (Amsterdam)	8**/25 (Amsterdam)
	40/30 (Toronto)	4.8**/25 (Toronto)
Heat Exchanger for DHW Heating	70/25	5/60
Notes:		

*: This is the mixed secondary inlet (return) temperature from the radiator and fan-coil systems.

**: This is the outlet air temperature from the heat recovery system assuming 60% ventilation heat recovery efficiency.

The fan-coil to radiator power ratio at design conditions was approximately 0.15. The heat provided by the radiators was considerably higher than that provided by the fan-coils. It was therefore decided to design the fan-coils at 40/30°C for the primary side (water side) to fully utilize the return water from the radiator system. Please refer to Table 7.3 for the design load for the radiators and fan-coils.

The design load for the space heating heat exchanger was 12.7 kW for Amsterdam and 14.2 kW for Toronto. The design load for the DHW heat exchanger was 19 kW. The heat exchangers were oversized by 20%.

Number of Homes Connected to the DH system

It was assumed that the DH system was located in a suburban residential area. For simplicity reasons, it was assumed that there were 1800 identical single-family homes, i.e. 1800 DH substations, connected to the CHP plant both for Amsterdam and Toronto. This number was derived by assuming the CHP plant capacity to be in the range of 20-35 MW_e . As described previously, the CHP plant covered about 40% of the DH peak load.

In reality, a DH system usually supplies an area with mixed types of buildings. However, the emphasis of this study was to compare the thermodynamic performance of different DH substations and building heating systems. It was decided that the above simplification would result in useful comparisons between different systems.

It should be noted that the assumed daily DHW consumption profile was the same for all houses. The effect of the DHW diversification factor on the heat production plant and distribution network was not considered. This assumption simplified the simulations, but resulted in very peaked load duration curve at the plant. However, it was considered that this simplification would not affect the comparison of different DH substation connection schemes.

7.2.4 Assumptions Used in the Simulations of Multi-family Homes

Case Studies

As described in Chapter 4, two district heating substation connection schemes were selected. The parallel connection (see Figure 4.6 for system configuration) was defined as the reference case (Case 1). The two-stage system (see Figure 4.7 for system configuration) was named Case 2.

In the two-stage system, the domestic hot water heating was cascaded in two stages. The radiator and fan-coil systems were in parallel connection in both cases.

Building Heating Load

It was assumed that 20 identical single-family homes were connected to one district heating substation.

Each single-family home had the same building heat loss factor as described in Table 7.3. The room temperature was selected at 20°C and ventilation air temperature was assumed at 25°C. Each house had a heat recovery unit with an average efficiency of 60%. The ventilation air flow rate was assumed as constant of 0.083 m³/s which equals approximately 0.5 air changes per hour.

Domestic Hot Water Consumption

The DHW consumption of the 20 houses was assumed at 4160 l/day, i.e. 208 l/day/house. This assumption value was derived from a demand curve for a 24-unit apartment building, taken from Yang (1994). The DHW consumption diversification in the 20 houses has been taken into account. Figure 7.8 shows the daily DHW profile for the 20 single-family homes. The peak domestic hot water load was 3.5 kW/house. This is an average peak power of a typical multi-family unit.



Figure 7.8: Domestic hot water consumption profile for multi-family homes (20 houses)

Domestic Hot Water Re-circulation

The DHW re-circulation water flow was assumed to be constant (0.12 l/s). The circulation line has an assumed average heat loss of 200 watts per house and a resulting temperature drop around the loop of 8° C.

Design Temperatures for Heating Components

As described previously, it was decided to keep the component sizes the same for all case studies. This means that the size for the heat exchangers, radiators, fan-coils and control valves were identical for Case 1 and Case 2. The main reason for this decision was to provide a comparison between the two cases resulting only from the effect of cascading. For the same reason, the CHP plant capacity and distribution network sizes were kept the same for all cases.

The design temperatures for the heating components are summarized in Table 7.5.

The design load for the space heating exchanger was 234 kW for Amsterdam and 324 kW for Toronto. The design load for the DHW heat exchanger was 74 kW including the heat loss of the re-circulation pipes. Similar to the single-family homes, the heat exchangers were oversized by 20%.

Table 7.5: Design temperatures for different heating components in multi-family homes

	Primary Side (°C)	Secondary Side (°C)			
Heat Exchanger for Space Heating	90/44 (Amsterdam)	39*/70 (Amsterdam)			
	90/44 (Toronto)	39*/70 (Toronto)			
Radiators	70/40	20 (room temperature)			
Fan-coils	70/30 (Amsterdam)	8**/25 (Amsterdam)			
	70/30 (Toronto)	4.8**/25 (Toronto)			
Heat Exchanger for DHW Heating (parallel)	70/25	5/60			
Heat Exchangers for DHW Heating (2-stage and					
3-stage)	70/25	5/60			
Notes:					
* : This is the mixed secondary inlet (return)) temperature from the radiators and fa	an-coils.			
** : This is the outlet air temperature from the	e heat recovery system assuming 60%	6 ventilation heat recovery			
efficiency.					
*** : It was assumed that the pre-heater and after-heater have the same size. Their size was determined according to					
the above design temperatures when th	ere was no space heating loads.				

Number of Homes Connected to the DH system

It was assumed that the DH system was located in a suburban residential area. There were 3600 homes, i.e. 180 DH substations, connected to the CHP plant both in Amsterdam and Toronto. This number was derived by assuming the CHP plant capacity to be in the range between 20-35 MW_e.

It should be noted that the number of homes was doubled compared to the case for the singlefamily homes. This was mainly because the DHW load profile used in the simulations, which included the diversification factor of the 20 homes (for one substation), was considerably lower compared to the one for the single-family homes. It should be noted that the number of homes does not reflect reality. However, the simplified assumptions did not affect the comparisons between the different substation configurations, since the comparison were made for the same type of building.

7.2.5 Assumptions Used in the Simulations of Small Office Buildings

Case Studies

As described in Chapter 4, three district heating substation connection schemes were selected. The parallel connection (Figure 4.8) was defined as the reference case named Case 1. For Case 2, the radiator and the fan-coil were cascaded, as shown in Figure 4.9.

Members of the Experts Group had suggested the addition of a case study (Case 2a, see Figure 4.10). In this case, as described in Section 4.4, the radiator system and fan-coil system was in pure series connection. That is, the ventilation air was heated in the fan-coils by the radiator return water and there were no control valves for the fan-coil system. Obviously, this system would result in the lowest DH return temperature compared to Case 1 and Case 2 as no secondary high temperature water was ever drawn by the fan-coil system. However, with this configuration, the fan-coil outlet air and room temperature may not be able to reach the required set-points at all times. The main reason to add this case study was to see if the ventilation outlet air and room temperature variations were in an acceptable range due to the elimination of these control valves.

Building Heating Load

As described in Chapter 3, the "Nuon" building in the Netherlands was selected as an example building for small office building case studies. The original design data was used in the case studies. Table 7.6 summarizes the design heating load and heat loss factors for the corresponding design outdoor temperatures in Amsterdam and Toronto. The room temperature was assumed at 20°C and ventilation air temperature was assumed at 25°C. The fresh air flow rate was constant at 2.78 m³/s and heat recovery efficiency was 60%.

Table 7.6: Design heating loads and heat loss factors for small office buildings in Amsterdam and Toronto

	Amsterdam	Toronto
Design Outdoor Temperature (°C)	-10	-18
Design Heating Load (kW)	225 (total)	280 (total)
Heat Recovery Unit (kW)	60	76
Fan-coil (kW)	57	67
Radiator (kW)	108	137
Heat Loss Factor (kW/°C)	7.50	7.37

The heat loss factors were slightly different for buildings in Amsterdam and Toronto. The heat loss factors (kW/°C) were calculated based on the difference of the room temperature (20°C) and design outdoor temperature. However, the ventilation outlet air temperature was assumed at 25° C which resulted in the difference of the building heat loss factor in Amsterdam and Toronto.

In can be determined from the above table that the fan-coil to radiator power ratio (FC/RAD ratio) at design condition was approximately 0.52 in Amsterdam and 0.49 in Toronto. The original design had no ventilation air re-circulation.

Members of the Experts Group suggested the addition of several case studies to evaluate the effectiveness of cascading at different FC/RAD ratios. In order to keep the building heating load the same as in the original case, it was decided to add ventilation air re-circulation. The fresh air intake flow was kept the same as in the original case. With different air re-circulation rates, the FC/RAD ratio could be varied.

To achieve FC/RAD ratios of 1 and 2, the fresh air was mixed with re-circulation air. This raised the total air flow through the fan-coil by 50% to 300%. For instance, in the case of FC/RAD of 2 in Toronto, the re-circulated air was 25% of the total flow through the fan coil. It was decided that with advanced air diffusion devices, increased air flow rates would not cause any significant discomfort to the building occupants.

The ventilation fresh air and re-circulation air flow were constant in the simulations.

Internal gains of 30 kW from 8am to 6pm were used in the simulations. For simplicity, this internal gain was included in the simulations for weekends as well.

Design Temperatures for Heating Components

As mentioned earlier, it was decided to keep the component sizes the same for all case studies. This means that the size for the heat exchangers, radiators, fan-coils and control valves were identical between Cases 1, 2 and 2a. The main reason for this decision was to compare the difference between the two cases resulting only from the effect of cascading. For the same reason, the CHP plant capacity and distribution network sizes were maintained identical for all cases.

Table 7.7 summarizes the design temperatures for the heating components used in the small office buildings.

It should be noted that it was decided to use the same design temperatures for the fan-coils when FC/RAD=1 and 2 although lower outlet water temperature (i.e. enlarging the fan-coil size) could be selected to keep the use of secondary supply water (high temperature water) at a minimum.

Number of Offices Connected to the DH system

It was assumed that the DH system was located in a downtown commercial area. There were 350 small offices, i.e. 350 DH substations, connected to the CHP plant both for Amsterdam and Toronto climates. This number was derived by assuming the CHP plant capacity to be in the range between 20-35 MW_e. As stated earlier, the CHP plant covered about 40% of the DH peak load.

Table 7.7: Design temperatures for different heating components in small office buildings

	Primary Side (°C)	Secondary Side (°C)			
Heat Exchanger for Space Heating	90/29 (Amsterdam)	24*/70 (Amsterdam)			
	90/30 (Toronto)	25*/70 (Toronto)			
Radiators	70/40	20 (room temperature)			
Fan-coils	40/24* (Amsterdam)	T _{air} **/25 (Amsterdam)			
	40/25* (Toronto)	T _{air} **/25 (Toronto)			
Notes:					
* : These temperatures were obtained by as	ssuming the fan-coils using only the ra	diator return water, based on the			
original design (i.e. no air re-circulation)	. This means the radiators and fan-coi	Is were in series connection at			
design condition.					
** : T_{air} was the mixed temperature of outlet	: T _{air} was the mixed temperature of outlet air from the ventilation heat recovery unit and re-circulation air. Its value				
depended on the air re-circulation rate.	T _{air} was 8°C for Amsterdam and 4.8C f	or Toronto when there was no air			
re-circulation (assuming 60% ventilation	heat recovery efficiency)				

In reality, as stated in previous sections, a DH system usually supplies an area with mixed types of buildings. However, the emphasis of this study was to compare the thermodynamic performance of different DH substations and building heating systems. It was decided that the above simplification would still result in valid comparisons between different systems.

8 Case Study Results for Large Multi-functional Buildings

During the simulations, the DH supply and return temperatures, as well as the DH flow rate at the substations and at the heat production plant, were captured and recorded every ten minutes. The raw data resulting from the simulations were then processed to obtain hourly-based data.

The simulation results are illustrated in time-series format for a typical winter week in Figure 8.1. The seasonal and yearly averaged data, such as flow-weighted average ΔT and system flows of the three studied connection schemes are shown in Table 8.1. Yearly performance data, such as DH water consumption, heat losses, pumping energy consumption and electricity production results, are shown in Table 8.2. The cost performance data are illustrated in Table 8.3 and Figure 8.2.

Comparisons between the parallel connection (Case 1) and cascaded connections (Case 2 and Case 3, see Figure 4.2 and 4.3 for their configuration) are made based on the simulation results.

8.1 Simulation Results

Figure 8.1 shows the hourly DH supply and return temperatures as well as DH flow rate at a substation for a typical winter week. The outdoor temperature is also illustrated in the figure.

It can be seen that the cascaded system, Cases 2 and 3, resulted in lower DH return temperatures and flow compared to the parallel system (Case 1). Case 3 had the best performance compared to the other two systems.

The significant ΔT improvement and system flow reduction occurred during the daytime when DHW was used. The return water from the first level was further cooled by the second level subsystems and then cooled in the DHW pre-heater.

During the nighttime, ΔT improvement was smaller compared to that of during the daytime. This was because DHW was not used during the nighttime. When heating loads were high at night, the ΔT improvement, due to cascading, was not significant (see hour 72 and 142 in Figure 8.2). This was because the return water temperatures and flow from the first level were quite high. Therefore, some of the water had to bypass the second level heating subsystems which resulted in small improvements in ΔT .

8.2 Seasonal and Yearly Averaged Performance Data

Table 8.1 summarizes the seasonal and yearly flow-weighted average ΔT and total system flow (45 buildings/ substations) for the large multi-functional building case studies. The ΔT improvement in °C and percentage of flow reduction relative to Case 1 are also shown in the table.

The four seasons were defined in the following way:

Summer	—	June, July and August
Fall	-	September, October and November
Winter	-	December, January and February
Spring	-	March, April and May

It can be seen from Table 8.1 that the cascaded systems (Cases 2 and 3) had higher DH water ΔT during all seasons. Overall, the cascaded system improved the ΔT by approximately 4°C for Case 2 and 5°C for Case 3. Due to the higher ΔT , the cascaded systems resulted in low system flows. The flow reduction of Case 2 was 6% and 7.8% for Case 3.



Figure 8.1: District heating supply and return temperature and flow rate at a substation for a large multi-functional building during a week in February

The results shown indicated that significant performance improvement occurred in the fall and spring. In these two seasons, there was over 4.5° C increase in Δ T and more than 7% in system flow reduction for Case 2, and over 6°C increase in Δ T and approximately 9% in system flow reduction for Case 3. In the winter, the Δ T improvement was about 3.6°C for Case 2 and 5.3°C for Case 3. The average system flow reduction was 5.7% for Case 2 and 8% for Case 3. Flow in a DH system is limiting in the winter. Therefore, Case 3 makes a larger than 8% increase in system capacity possible. In the summer, the cascaded systems had smaller performance improvement compared to the other seasons. In the summer, there were no or very low loads in the heating subsystems and effects of the cascading were less significant.

Table 8.1: Seasonal flow-weighted average district heating temperature difference (ΔT) *and average system flows for large multi-functional buildings*

	Summer		Fall		Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔΤ		ΔT		ΔT		ΔT	
Case1	61.24	-	63.25	-	59.58	-	61.91	-	61.16	-
Case2	63.28	2.04	68.24	4.99	63.19	3.61	66.70	4.79	65.15	3.99
Case3	63.30	2.06	69.41	6.16	64.87	5.29	68.16	6.25	66.46	5.30
	Flow	%	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)		(l/s)	
Case1	40.1	-	71.2	-	139.9	-	86.7	-	84.2	-
Case2	38.8	3.24	66.1	7.16	132.0	5.65	80.6	7.04	79.1	6.06
Case3	38.8	3.25	65.0	8.71	128.6	8.08	78.9	9.00	77.6	7.84
F.W.T. Δ1	F.W.T. ΔT: Flow-Weighted Average ΔT, (°C).									
	D.L. Dograd (°C) Improvement relative to Case 1									

D.I.: Degree (°C) Improvement relative to Case I.

%: Percentage of system flow reduction relative to Case1.

It should be mentioned that the size of all components was kept the same in all three cases. This means that some of the components, such as Air Handler 1, pool water heating etc., were oversized if they were placed in the parallel connection (Case 1). The ΔT of the parallel system would have been smaller than the values shown in Table 8.1 if these components had been sized for the corresponding supply temperature at design condition. In other words, the improvement of ΔT and reduction of system flow due to cascading would be higher.

The results shown in Table 8.1 also indicate, except in the summer, that cascaded system Case 3 had better performance compared to Case 2. The connection scheme difference between the two cases was that the glycol heating subsystem was placed in the second level instead of the first level in Case 3. As described in Chapters 3 and 4, the ventilation Air Handler 1 (which was placed in the second level in Cases 2 and 3) was shut off at night. The glycol heating system, however, was operated continuously when the outside temperature was below 4°C. By placing the glycol heating system in the second level, Case 3 resulted in a more balanced power ratio between the first and second level, during both daytime and nighttime. This means less return water from the first level bypassed the second level for the Case 3 system. Consequently, Case 3 resulted in higher Δ T and lower system flow rate compared to the Case 2 system.

In the summer, Cases 2 and 3 had approximately the same average temperature difference in DH supply and return temperatures. The reason for this was that the glycol heating system was not in use in the summer as the outside temperature was usually well above 4°C. Table 8.2 summarizes the yearly average performance data for the three cases. The data shown in the table are yearly average flow-weighted DH temperature differences at the substations and yearly DH water consumption, building thermal energy consumption, fuel consumption, DH network heat loss, pumping power as well as electricity production. Comparisons between the cascaded systems (Cases 2 and 3) and the parallel system (Case 1) are also shown in the table.

From Table 8.2 it can be seen that Case 3 has a Δ T increase of 5.3°C while Case 2 has a Δ T increase of 4°C compared to Case 1. Due to the higher Δ T, the cascaded systems resulted in lower DH water consumption and lower network heat loss (-6% for Case 2 and -8% for Case 3), lower pumping energy demand (-11% for Case 2 and -15% for Case 3), and slightly higher net electricity production due to the lower pumping energy demand.

The fuel consumption for the three cases studied was approximately the same. This was because the fuel flow to the combined cycle gas turbine was constant. The fuel consumption reduction of the peaking boiler, resulting from the lower DH return temperature of the cascaded systems (Cases 2 and 3), was insignificant (less than 1 MWh).

	Case 1	Case 2	Comparison		Case 3	Comparison	
				%			%
Flow-weighted							
average ΔT at	61.16	65.15	3.99	-	66.46	5.30	-
substations (°C)							
DH water volume							
consumption (m ³ /yr)	2656000	2495000	-161000	-6.1	2446400	-209600	-7.9
Building thermal							
energy consumption	186830	186830	-	-	186830	-	-
(MWh/yr)							
Thermal energy							
supplied to the DH	199042	198302	-740	-0.4	198079	-963	-0.5
system (MWh/yr)							
Network heat losses							
(MWh/yr)	12212	11472	-740	-6.1	11249	-963	-7.9
Pumping energy							
(MWh/yr)	184	163	-21	-11.4	156	-28	-15.2
Electricity production							
(MWh/yr)	210720	210910	190	0.09	210950	230	0.11
Fuel consumption							
(MWh/yr)	479400	479400	-	-	479400	-	-

Table 8.2: Simulation and comparison results of case studies for the large multi-functional buildings (45 buildings/substations)

It should be noted that the building thermal energy consumption, obtained from the simulation raw data for the three cases, were slightly different. The differences resulted from the slightly different control methodology between the systems as well as the transients in the simulations. However, the difference was very small (less that $\pm 0.05\%$). The average value was used in Table 8.2 in order to keep the income from the DH energy sales identical between the economic analyses.

8.3 Annual Cost Comparisons

As described in the previous section, the cascaded systems have lower DH return temperatures. Consequently, the cascaded systems resulted in lower heat losses, lower pumping energy demand and higher net electricity production. The annual cost savings of the cascaded systems (Case 2 and Case 3) were compared to the reference case.

The annual net equivalent worth of a DH system is the difference between the income and expenses and can be calculated by:

Annual net equivalent worth = (a): Income from sale thermal energy to consumers

- + (b): Income from sale electricity to grid
- (c): Fuel cost
- (d): Operational and maintenance costs
- (f): Labour cost
- (g): Annual repayment of capital costs

The operational and maintenance costs as well as the labour costs were similar in each case. The total capital cost was considered the same for all three cases since the heating component size, network size and heat production plant size were the same.

Therefore, only income from the sale of thermal energy, electricity and fuel cost for the three cases were calculated and compared. The cost comparisons are shown in Table 8.3. The prices listed in the table were based on the recent Canadian market. The DH energy price was the price charged to the consumers and the electricity price was the selling price to the grid. The thermal and fuel consumption values can be found in Table 8.2. The electrical energy available for sale to the grid is the electrical energy produced subtracted by the pumping energy. This data can also be found in Table 8.2.

Table 8.3: Annual cost comparison results of case studies for large multi-functional buildings (45 buildings/substations)

	Price	Annual Income/(Cost)						
	(\$/1010011)	(\v) Case1	Case2	Difference	Case3	Difference		
DH	70	13,078,100	13,078,100	-	13,078,100	-		
Energy								
Elec.	50	10,526,800	10,537,350	\$10,500	10,539,700	\$12,900		
Energy								
Fuel Cost	16	(7,670,400)	(7,670,400)	-	(7,670,400)	-		
Total				\$10,500		\$12,900		

Table 8.3 shows that the cascaded systems have a higher overall annual income compared to the parallel system. Case 2 resulted in a yearly cost saving of \$10,550 while Case 3 resulted in a yearly cost saving of \$12,900, compared to Case 1. The savings resulted from lower pumping energy requirements in the cascaded systems.

Because the choice of plant type (extraction/condensation) and all other hardware were kept identical between cases (as per the request of the Experts Group), the savings in electricity for the pumps are insignificant compared to the sale of electricity. The reduced flow does allow for an increase in the connected load of almost 10% without increasing the pipe sizes. This will certainly make the economics more favourable.

9 Case Study Results for Single-family Homes

During the simulations, the DH supply and return temperatures, as well as the DH flow rate at the substations and at the heat production plant, were captured and recorded every ten minutes. The data resulting from the simulations were then processed to obtain hourly-based data.

The simulation results are illustrated in time-series format for a typical winter week in Figure 9.1 and 9.2 for Amsterdam and Toronto respectively. The seasonal and yearly averaged data, such as flow-weighted ΔT and system flows of the two connection schemes, are shown in Tables 9.1 and 9.2. Yearly performance data, such as DH water consumption, heat losses, pumping energy consumption and electricity production are shown in Tables 9.3 and 9.4. The cost analysis is described in Section 9.3

Comparisons between the parallel system (Case 1) and cascaded system (Case 2) were made based on the simulation results.

9.1 Simulation Results

Figures 9.1 and 9.2 show the hourly DH supply and return temperatures as well as DH flow rates at a substation for a typical winter week for the Amsterdam and Toronto case studies respectively. The outdoor temperature is also illustrated in the figures.

It can be seen that while the space heating load was high at night, cascaded system (Case 2, see Figure 4.5 for system configuration) resulted in lower DH return temperatures and flow compared to the parallel system (Case 1, see Figure 4.4 for system configuration). This is because in the cascaded system, the return water from the radiator system was further cooled in the fan-coil system. Due to small ventilation heating loads, the reduction in the DH return temperature was in the $1-3^{\circ}$ C range for both climates.

While the heating loads were relatively low during the daytime, due to higher outside temperatures or solar gain, the DH return temperatures from the two cases were more or less the same. There were two reasons for this result. At low heating loads, if the return water temperature from the radiator system was lower than 27°C, it bypassed the fan-coil system in the cascaded system. In this situation, Case 2 functioned as a parallel connection, similar to Case 1. The second reason was that at low heating loads (especially resulting from high solar gains), fan-coil power dominated and radiator power was relatively low. In this situation, fan-coil power was provided mainly by bypassed water instead of the radiator water, which is similar to the parallel connection.

As there were no or very low space heating loads in the summer, the DH return temperatures and flow from the parallel connection (Case 1) and the cascaded system (Case 2) were approximately the same. For this reason, simulation results for the summer months are not shown here.



Figure 9.1: District heating supply and return temperature and flow rate at a substation for a single-family home in Amsterdam during a week in February



Figure 9.2: District heating supply and return temperature and flow rate at a substation for a single-family home in Toronto during a week in January.

9.2 Seasonal and Yearly Averaged Performance Data

Tables 9.1 and 9.2 summarize the seasonal and yearly flow-weighted average ΔT and total system flow (1800 homes/substations) for Amsterdam and Toronto respectively. The ΔT improvement in °C and percentage of flow reduction relative to Case 1 are also shown in the tables.

The four seasons were defined in the following way:

Summer –	June, July and August
Fall –	September, October and November
Winter –	December, January and February
Spring –	March, April and May

Table 9.1: Seasonal flow-weighted average district heating temperature difference (ΔT) and average system flows for single-family homes in Amsterdam.

	Summer		Fall		Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔT		ΔT		ΔT		ΔT	
Case1	53.34	-	53.87	-	52.78	-	53.77	-	53.33	-
Case2	55.45	0.11	53.98	0.11	53.05	0.27	53.95	0.18	53.43	0.1
	Flow	%	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)		(l/s)	
Case1	11.40	-	31.3		57.2	-	30.8	-	32.5	-
Case2	11.34	0.53	31.2	0.32	56.8	0.70	30.7	0.32	32.4	0.31
F.W.A. Δ	T: Flow-weig	hted Averag	ge ∆T, (°C).							
D.I.: Deg	ree (°C) Impi	rovement re	lative to Cas	se1.						
%. Porco	ntage of eve	tom flow roc	luction relati	ve to Casel						

Table 9.2: Seasonal flow-weighted average district heating temperature difference (ΔT) and average system flows for single-family homes in Toronto

	Summer		Fall		Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔT		ΔT		ΔT		ΔT	
Case1	52.57	-	53.10	-	53.50	-	53.37	-	53.34	-
Case2	52.76	0.19	53.28	0.18	55.04	1.54	53.83	0.46	54.24	0.9
	Flow	%	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)		(l/s)	
Case1	9.10	-	35.0	-	80.8	-	39.4	-	40.9	-
Case2	9.05	0.55	34.9	0.29	78.7	2.60	39.1	0.76	40.2	1.71
F.W.A. Δ	F.W.A. ΔT: Flow-weighted Average ΔT, (°C).									
	D L. Degree (°C) Impovement relative to Case1									

%: Percentage of system flow reduction relative to Case1.

It can be seen from the tables that the cascaded system (Case 2) has a higher DH water temperature difference in all seasons. Overall, the cascaded system improved the ΔT by 0.1°C in Amsterdam and 0.9°C in Toronto. Due to relatively higher ΔT , Case 2 resulted in lower system flows.

The results shown in the tables indicate that the largest ΔT improvement occurred in the winter, when the space heating loads were high, both for Amsterdam and Toronto. The ΔT of Case 2 increased by 0.3°C in Amsterdam and by 1.54°C in Toronto. As explained previously, at high space heating loads, the radiator water return temperature was high and this water was further cooled by the ventilation air in the cascaded system (Case 2). Under these conditions, only a small amount of the high temperature water (secondary water) was ever required (valve BP2 either closed or open slightly, see Figure 4.5). The largest system flow reduction happened in the winter

for both climates. The average flow reduction in Toronto during the winter season was 2.60%. Flow is limited during winter making a substantial system capacity expansion possible.

In the summer, fall and spring, it seems that Case 2 had a smaller degree of improvement in ΔT and system flows compared to Case 1. This was due to the fact that while the space heating loads were low, more secondary water was required in the fan-coil system as the return water from the radiator was not sufficient to meet the ventilation heating demand. Therefore, Case 2 resulted in small ΔT improvement.

The above results agree with what could be observed from Figure 9.1 and 9.2.

The results shown in the above tables indicate that the overall system performance improvement due to cascading was greater in Toronto. This was because Toronto has a higher amount of degree-days (more days with high space heating loads) compared to Amsterdam.

It should be noted that the size of all components was kept the same in both cases. This means the fan-coil was oversized in the parallel connection. The ΔT of the parallel system would be smaller than the value shown in Table 9.1 and 9.2 if the fan-coils had been sized for the corresponding supply temperature at design conditions. In other words, the increase in ΔT and reduction of the system flow due to cascading would be higher than indicated here.

Tables 9.3 and 9.4 summarized the yearly average data for Amsterdam and Toronto respectively. The data shown in the tables are yearly average flow-weighted DH temperature difference at the DH substations and yearly DH water consumption, building thermal energy consumption, fuel consumption, DH network heat losses, pumping power as well as electricity production. Comparisons between Case 1 (parallel connection) and Case 2 (cascaded heating system) are also shown in the tables.

	Case 1	Case 2	Comparison	
				%
Flow-weighted average ΔT at substations (°C)	53.33	53.43	0.1	
DH water volume consumption (m ³ /yr)	1026300	1023900	-2400	-0.23
Building energy consumption (MWh _{tt} /yr)	62897	62897		
Thermal energy supplied to the DH system (MWh_th/yr)	68597	68470	-127	-0.19
Network heat losses (MWh/yr)	5700	5573	-127	-2.23
Pumping energy (MWh/yr)	48	47	-1	-2.08
Electricity production (MWh/yr)	198530	198555	25	0.01
Fuel consumption (MWh/yr)	417530	417530	0	-

Table 9.3: Simulation and comparison results of case studies for single-family homes in Amsterdam (1800 homes/substations)

Table 9.4: Simulation and comparison results of case studies for single-family homes in Toronto (1800 homes/substations)

	Case 1	Case 2	Compariso	n
				%
Flow-weighted average ΔT at substations (°C)	53.34	54.24	0.90	
DH water volume consumption (m ³ /yr)	1289200	1269200	-20000	-1.56
Building energy consumption (MWh _{th} /yr)	79013	79013		
Thermal energy supplied to the DH system (MWh _{tt} /yr)	85840	85540	-300	-0.35
Network heat losses (MWh/yr)	6827	6527	-300	-4.39
Pumping energy (MWh/yr)	65	63	-2	-3.08
Electricity production (MWh/yr)	230000	230088	88	0.04
Fuel consumption (MWh/yr)	483450	483450	0	-

From the above tables, it can be seen that the difference in yearly flow-weighted average district heating ΔT was relatively small; about 0.1°C for Amsterdam and 0.9°C for Toronto. However, due to the lower DH return temperature, the cascaded system (Case 2) resulted in lower network heat losses (-2.23% for Amsterdam and -4.39% for Toronto) and lower pumping energy demand (-2.08% for Amsterdam and -3.08% for Toronto).

The fuel consumption for the two cases studied was approximately the same. This was because fuel flow to the combined cycle gas turbine was kept constant. The fuel consumption reduction of the peaking boiler resulted from the lower heat losses in the DH return line of Case 2.

It should be noted that the building thermal energy consumption, obtained from the simulation raw data for the two cases, were slightly different. The differences resulted from the slightly different control methodology between the two systems as well as the transients in the simulations. However, the difference was very small (less than $\pm 0.1\%$ for both Amsterdam and Toronto). The average value was used in the tables in order to keep the income from selling the heat the same in the economic analyses that will be discussed in the following section.

9.3 Annual Cost Comparisons

The cascaded system has lower DH return temperatures. Consequently, it resulted in lower heat losses, lower pumping energy demand and higher net electricity production. The annual cost savings of the cascaded system (Case 2) were compared to the reference case.

The annual net equivalent worth of a DH system is the difference between the income and expenses and can be calculated by:

Annual net equivalent worth	= (a):	Income from sale thermal energy to consumers
	+ (b):	Income from sale electricity to grid
	- (c):	Fuel cost
	- (d):	Operational and maintenance costs
	- (f):	Labour costs
	- (g):	Annual repayment of capital costs

The operational and maintenance costs, as well as the labour costs, were similar for both cases. Although the cascaded system has one more control valve installed in the DH substations, the additional cost of those valves is negligible compared to the CHP plant capital cost which is over 75 million dollars (assuming $3000/kW_e$). The total capital cost, therefore, was considered to be the same for the two cases.

Therefore, only income from the sale of thermal energy and electricity and fuel cost for the two cases were calculated and compared. The cost comparisons are shown in Tables 9.5 and 9.6 for Amsterdam and Toronto respectively. The prices listed in Table 9.5 were based on the recent Canadian market. The DH energy price was price charged to the consumers and the electricity price was the selling price to the grid. The thermal and fuel consumption values can be found in Tables 9.3 and 9.4. The electrical energy available for sale to the grid is the electrical energy produced subtracted by the pumping energy. These data can also be found in Tables 9.3 and 9.4.

Table 9.5: Annual cost comparison results of case studies for single-family homes in Amsterdam (1800 homes/substations)

	Price	Annual Income/(Cost)	Difference	
	(\$/MWh)	(\$)	(\$)	
		Case1 Case2		
DH Energy	70	\$4,402,790	\$4,402,790	-
Elec. Energy	50	\$9,924,100	\$9,925,400	\$1,300
Fuel Cost	16	(\$6,680,480)	(\$6,680,480)	-
Total				\$1,300

Table 9.6: Annual cost comparison results of case studies for single-family homes in Toronto (1800 homes/substations)

	Price	Annual Income/(Cost)	Difference	
	(\$/MWh)	(\$)	(\$)	
		Case1 Case2		
DH Energy	70	\$5,530,910	\$5,530,910	-
Elec. Energy	50	\$11,496,750	\$11,501,250	\$4,500
Fuel Cost	16	(\$7,735,200)	(\$7,735,200)	-
Total				\$4,500

The above tables show that the cascaded system has higher overall annual income compared to the parallel system, both for Amsterdam and Toronto. The annual cost savings from the cascaded system was \$1,300 for Amsterdam and \$4,500 for Toronto. The difference was mainly the result of higher net electricity production due to the lower pumping energy of the cascaded system.

10 Case Study Results for Multi-family Homes

During the simulations, the DH supply and return temperatures, as well as the DH flow rate at the substations and at the heat production plant, were captured and recorded every ten minutes. The simulation data were then processed to obtain hourly-based data.

Similar to the single-family home cases, the simulation results are illustrated in time-series format for a typical winter week and a typical summer week. The seasonal and yearly averaged data, such as flow-weighted ΔT and system flows of the two studied connection schemes are shown in Section 10.2. Other yearly performance data, such as DH water consumption, heat losses, pumping energy consumption and electricity production results are also shown in this section. The cost analyses are described in Section 10.3.

Comparisons between the parallel system (Case 1) and cascaded systems (Case 2) were made based on the simulation results.

10.1 Simulation Results

Figures 10.1 and 10.2 show the hourly DH supply and return temperatures as well as DH flow rate at a substation for a typical winter week for the Amsterdam and Toronto case studies respectively. Figures 10.3 and 10.4 show the results for a typical summer week for the two climates. The outdoor temperatures are also illustrated in the figures.

The figures show that the 2-stage system (see Figure4.7 for system configuration), where the DHW was heated in two stages, resulted in lower DH return temperatures compared to the parallel system (Case 1, see Figure 4.6 for system configuration) both for the Amsterdam and Toronto cases. This was because the DH return water from the space heating system was further cooled in the DHW pre-heater. Also, the DH re-circulation water was mixed with pre-heated municipal water, reducing exergy losses.

The results shown in the figures also indicated that the improvement of ΔT , due to cascading, was more significant in the summer than in the winter. As stated in Section 7.2.4, the design space heating load was 234 kW for Amsterdam and 324 kW for Toronto. The peak DHW load was 74 kW. The space heating load in the winter was considerably higher than the domestic hot water load. Due to the high flow, the DH return water temperature from the space heating system was reduced only slightly by the cold municipal water in the pre-heater. While at very low or no space heating loads, such as in the summer, the 2-stage system resulted in a much lower DH return temperature.

Due to reduced return temperature, the DH flow from the 2-stage system was also lower than that of the parallel system, although this is difficult to distinguish from the figure due to the scale of the graph.



Figure 10.1: District heating supply and return temperature and flow rate at a substation for multi-family homes in Amsterdam during a week in February


Figure 10.2: District heating supply and return temperature and flow rate at a substation for multi-family homes in Toronto during a week in January



Figure 10.3: District heating supply and return temperature and flow rate at a substation for multi-family homes in Amsterdam during a week in August



Figure 10.4: District heating supply and return temperature and flow rate at a substation for multi-family homes in Toronto during a week in August

10.2 Seasonal and Yearly Averaged Performance Data

Tables 10.1 and 10.2 summarize the seasonal and yearly flow-weighted average ΔT and total system flow (3600 homes/180 substations) for Amsterdam and Toronto respectively. The ΔT improvement in °C and percentage of flow reduction relative to Case 1 are also shown in the tables.

The four seasons were defined in the following way:

Summer	-	June, July and August
Fall	-	September, October and November
Winter	-	December, January and February
Spring	-	March, April and May

Table 10.1: Seasonal flow-weighted average district heating temperature difference (ΔT) and average system flows for multi-family homes in Amsterdam

	Summer		Fall		Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔT		ΔT		ΔT		ΔT	
Case1	39.91	-	48.97	-	49.76	-	48.55	-	48.23	-
Case2	44.17	4.26	50.16	1.19	50.69	0.93	49.74	1.19	49.66	1.43
	Flow	%	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)		(l/s)	
Case1	30.9	-	70.3	-	122.2	-	69.1		72.8	-
Case2	29.1	5.83	68.6	2.42	120.0	1.80	67.5	2.32	71.0	2.47
F.W.A. ∆1	F.W.A. ΔT: Flow-weighted Average ΔT, (°C).									
D.I.: Degr	ee (°C) Impr	rovement re	lative to Cas	se1.						

%: Percentage of system flow reduction relative to Case1.

<i>Table 10.2: Seasonal flow-weighted average district heating temperature difference (ΔT) and</i>
average system flows for multi-family homes in Toronto

	Summer		Fall		Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔT		ΔT		ΔT		ΔT	
Case1	36.38	-	47.89	-	50.31	-	48.4	-	48.27	
Case2	40.63	4.25	48.96	1.07	51.08	0.77	49.36	0.96	49.47	1.2
	Flow	%	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)		(l/s)	
Case1	27.6	-	79.3	-	172.9	-	88.5	-	91.7	-
Case2	25.4	7.97	77.6	2.14	170.3	1.50	86.8	1.92	89.5	2.40
F.W.A. ∆1	: Flow-weig	hted Averag	je ∆T, (°C).							
D.I.: Degr	ee (°C) Impi	rovement re	lative to Cas	e1.						

%: Percentage of system flow reduction relative to Case1.

It can be seen from the tables that the cascaded systems have higher DH temperature difference in all seasons. Overall, the 2-stage system improved the ΔT by 1.2°C in Amsterdam and 1.4°C in Toronto. The yearly system flow reduction was over 2.4% for both climates.

The results indicate that the big ΔT improvement and system flow reduction occurred in the summer for both climates. This agrees with what has been observed from the time-series data illustrated in Figures 10.1 through 10.4. The summer time flow-weighted ΔT increased more than 4°C both in Amsterdam and Toronto. The system flow reduction was 5.8% in Amsterdam and 7.9% in Toronto.

Tables 10.3 and 10.4 summarize the yearly average data for Amsterdam and Toronto respectively. The data shown in the tables are yearly average flow-weighted DH temperature difference at the DH substations and yearly DH water consumption, building thermal energy consumption, fuel consumption, DH network heat loss, pumping power as well as electricity production. Comparisons between the parallel connection (Case 1) and cascaded systems (Case 2) are also shown in the tables.

	Case 1	Case 2	Comparison	
				%
Flow-weighted average ΔT at substations (°C)	48.23	49.66	1.43	-
DH water volume consumption (m ³ /yr)	2296600	2230400	-66200	-2.88
Building energy consumption (MWh _{tt} /yr)	127322	127322	-	-
Thermal energy supplied to the DH system (MWh $_{\mbox{\tiny th}}/\mbox{yr})$	137570	137240	-330	-3.22
Network heat losses (MWh/yr)	10248	9918	-330	-3.22
Pumping energy (MWh/yr)	110	106	-4	-3.64
Electricity production (MWh/yr)	183430	183465	35	0.02
Fuel consumption (MWh/yr)	400140	400140	0	-

Table 10.3: Simulation and comparison results of case studies for multi-family homes in Amsterdam (3600 homes/180 substations)

Table 10.4: Simulation and comparison results of case studies for multi-family homes in Toronto (3600 homes/180 substations)

	Case 1	Case 2	Comparison	
				%
Flow-weighted average ΔT at substations (°C)	48.27	49.47	1.2	-
DH water volume consumption (m ³ /yr)	2890200	2820400	-69800	-2.42
Building energy consumption (MWh _{th} /yr)	160375	160375	-	-
Thermal energy supplied to the DH system (MWh_th/yr)	172870	172600	-270	-0.16
Network heat losses (MWh/yr)	12495	12225	-270	-2.16
Pumping energy (MWh/yr)	190	183	-7	-3.68
Electricity production (MWh/yr)	265090	265120	30	0.01
Fuel consumption (MWh/yr)	571600	571600	0	-

These tables indicate that due to lower DH return temperatures, the cascaded system (Case 2) resulted in lower network heat losses (3.22% for Amsterdam and 2.16% for Toronto), and lower pumping energy demand (3.64% for Amsterdam and 3.68% for Toronto).

The fuel consumption for Cases 1 and 2 was approximately the same. This was because fuel flow to the combined cycle gas turbine was assumed as constant. The fuel consumption reduction of the peaking boiler resulted from the lower heat losses in the DH return line.

It should be noted that the building thermal energy consumption obtained from the simulation data for Cases 1 and 2 were slightly different. The differences resulted from the transients in the simulation and the control methodology. However, the difference was very small (less than $\pm 0.02\%$ both for Amsterdam and Toronto). The average value was used in the tables in order to keep the income from selling the heat identical between the economic analyses.

10.3 **Annual Cost Comparisons**

The cascaded system had lower DH return temperatures. Consequently, it resulted in lower heat losses, lower pumping energy demand and higher net electricity production. The annual cost savings of the cascaded system

(Case 2) were compared to the reference case.

The annual net equivalent worth of a DH system is the difference between the income and expenses and can be calculated by:

Annual net equivalent worth = (a):

Income from sale thermal energy to consumers

- + (b): Income from sale electricity to grid
- (c): Fuel costs
- (d): Operational and maintenance costs
- (f): Labour costs
- (g): Annual repayment of capital costs

The operational and maintenance costs as well as the labour costs were similar for both cases. Although the cascaded systems had one more heat exchanger in the DH substations, the additional costs were negligible compared to the CHP plant capital cost which is over 75 million dollars (assuming $3000/kW_e$). Therefore, the total capital cost was considered the same for the two cases.

Therefore, only income from the sale of thermal energy and electricity, in addition to the fuel cost for the two cases, were calculated and compared. The cost comparisons are shown in Table 10.5 and 10.6 for Amsterdam and Toronto respectively. The prices listed in the tables were based on the recent Canadian market. The DH energy price was the price charged to the consumers and the electricity price was the selling price to the grid. The thermal and fuel consumption values can be found in Tables 10.3 and 10.4. The electrical energy available for sale to the grid was the electrical energy produced subtracted by the pumping energy. This information can also be found in Tables 10.3 and 10.4.

Table 10.5: Annual cost comparison results of case studies for multi-family homes in Amsterdam (3600 homes/180 substations)

	Price	Annual Income/(Cost)	Difference			
	(\$/MWh)	(\$)				
		Case1	Case2			
DH Energy	70	\$8,912,540	\$8,912,540	-		
Elec. Energy	50	\$9,199,000	\$9,167,950	\$1,950		
Fuel Cost	16	(\$6,402,240)	(\$6,402,240)	-		
Total				\$1,950		

Table 10.6: Annual cost comparison results of case studies for multi-family homes in Toronto (3600 homes/180 substations)

	Price	Annual Income/(Cost)	Difference	
	(\$/MWh)	(\$)	(\$)	
		Case1	Case2	
DH Energy	70	\$11,226,250	\$11,226,250	-
Elec. Energy	50	\$13,245,000	\$13,246,850	\$1,850
Fuel Cost	16	(\$9,145,600)	(\$9,145,600)	-
Total				\$1,850

The above tables show that the cascaded system has a higher overall annual income compared to the parallel system, both for Amsterdam and Toronto. The annual cost savings from the cascaded system was \$1,950 for Amsterdam and \$1,850 for Toronto. The difference resulted from the lower pumping energy of the cascaded systems.

11 Case Study Results for Small Office Buildings

For small office buildings, simulations were carried out for systems designed with different fancoil to radiator ratio (FC/RAD ratio), such as FC/RAD=0.5,1 and 2. This was done in compliance with a request from the Experts Group.

During the simulations, the DH supply and return temperatures, as well as the DH flow rate at the substations and at the heat production plant were captured and recorded every ten minutes. The simulation data were then processed to obtain hourly-based data.

Similar to the home cases, the simulation results are illustrated in time-series format for a typical winter week in Section 11.1. The seasonal and yearly averaged data, such as flow-weighted ΔT and system flows of the three studied connection schemes are shown in Section 11.2. Other yearly performance data, such as DH water consumption, heat losses, pumping energy consumption and electricity production results, are shown in the second section. The comparison of operation costs between the different systems is described in Section 11.3.

Comparisons between the parallel system (Case 1, see Figure 4.8 for system configuration) and the cascaded systems (Cases 2 and 2a, see Figure 4.9 and 4.10 for system configurations) are made based on the simulation results for different FC/RAD ratios.

11.1 Simulation Results

Simulation Results for the Original Design Case (FC/RAD=0.5)

Figures 11.1 and 11.2 illustrate the hourly DH supply and return temperatures as well as flow rate at a substation for a typical winter week for Amsterdam and Toronto respectively. The fan-coil to radiator power ratio (FC/RAD), at design conditions was 0.52 for Amsterdam and 0.49 for Toronto. As mentioned in Section 7.2.5, at these FC/RAD ratios, there was no air re-circulation in the ventilation system. The simulation results showed that the return temperatures from Case 2 were in the range of 1-8°C lower than that of the Reference Case (Case 1). Because the radiator water was further cooled by the ventilation air in the cascaded system (Case 2), it resulted in lower return temperatures. The higher return temperature differences between Case 1 and Case 2 occurred when heating loads were high. Under these conditions, the radiator water flow and exit temperature were high and less secondary water was bypassed to the fan-coil system for Case 2. When the heating loads were low, the return temperature differences between the two systems became smaller as more secondary water was required.

From the figures, it also can be seen that the series connection system (Case 2a) has the lowest DH return temperatures compared to the Reference Case (Case 1) as well as Case 2. However, the return temperature differences between Case 2 and Case 2a were relatively small, especially at high heating loads. At conditions when the radiator water flow and temperature were high enough to provide required ventilation load without drawing any secondary high temperature water, the return temperatures from the two cases (Case 2 and 2a) were the same. In this situation, Case 2 acted as in series connection as well.

In Case 2a, the fan-coil system was using radiator water only to heat the ventilation air (no secondary bypass flow). In this case, the ventilation air temperature was not controlled. This could result in room temperature lower than its set point and/or very cool ventilation outlet air. The simulation results showed that the room temperature variation was between 20-22°C with its set-point of 20°C and the ventilation air temperature variation was between 18-26°C. Low ventilation outlet air temperatures happened in the spring and fall when the building heating loads were low. The results also indicated that even with reduced fan-coil heat output, the room temperature was still able to keep at the required level. This is because while the fan-coil power reduced, the radiators provided more heat in order to keep the required room temperature. Since the designed radiator capacity was larger than the fan-coil capacity, it was possible to maintain the room temperature at the required level.



Figure 11.1: District heating supply and return temperature and flow rate at a substation for small office buildings in Amsterdam during a week in February. FC/RAD=0.5



Figure 11.2: District heating supply and return temperature and flow rate at a substation for small office buildings in Toronto during a week in January. FC/RAD=0.5

Simulation Results for FC/RAD=1

Figures 11.3 and 11.4 show the hourly DH supply and return temperatures as well as flow rate at a substation for a typical winter week for Amsterdam and Toronto for an FC/RAD ratio of 1. Similar to the original design (FC/RAD=0.5), the simulation results showed that the return temperatures from Case 2 were lower than that of the Reference Case (Case 1) when the building heating loads were high. However, the return temperature differences between the two cases were smaller compared to the original design. This was because the radiator capacity was reduced and secondary high temperature water was almost always required by the fan-coil system of Case 2. At low heating loads, heat was mainly provided by the fan-coil system. This resulted in the parallel connection (Case 1) and cascaded system (Case 2) having approximately the same return temperatures.



Figure 11.3: District heating supply and return temperature and flow rate at a substation for small office buildings in Amsterdam during a week in February. FC/RAD=1

The simulation results indicated that for Case 2a (series connection), the room temperature variation was between 18-23°C and air temperature variation was between 18-24°C. Lower room temperature occurred in the winter although low ventilation air temperatures happened in the spring and fall. It seems that the room temperature was relatively low from a comfort point of view. The return temperature and flow of Case 2a therefore, were not shown in the figure since this comparison was not very meaningful.



Figure 11.4: District heating supply and return temperature and flow rate at a substation for small office buildings in Toronto during a week in January. FC/RAD=1

Simulation Results for FC/RAD=2

Figures 11.5 and 11.6 show the hourly DH supply and return temperatures as well as flow rate at a substation for a typical winter week for Amsterdam and Toronto for an FC/RAD ratio of 2. Similar to the original system (FC/RAD \cong 0.5) and FC/RAD=1 cases, the simulation results showed that the return temperatures from the cascaded system (Case 2) were lower than that of the parallel system (Case 1) when the building heating loads were high. However, the return temperature differences between the two cases were smaller compared to the original system (FC/RAD \cong 0.5) as well as the FC/RAD=1 case. This was because the radiator capacity was further reduced for the FC/RAD=2 case and secondary high temperature water was almost always required by the fan-coil system of



Figure 11.5 District heating supply and return temperature and flow rate at a substation for small office buildings in Amsterdam during a week in February. FC/RAD=2

Case 2. At low heating loads, heat was mainly provided by the fan-coil system. This resulted in the parallel connection (Case 1) and the cascaded system (Case 2) having approximately the same return temperatures.

The simulation results indicated that for Case 2a, the room temperature variation was between 17-21°C and air temperature variation was between 18-22°C. Compared to the FC/RAD=1 case, the room temperature and ventilation outlet air temperature were reduced further. Lower room temperatures occurred in the winter although low ventilation air temperatures happened in the spring and fall as well. It seems that the supply temperature was at times too low from a comfort point of view. As a result, the return temperature and flow of Case 2a were not shown in the figure. Therefore, these high FC/RAD ratios are not recommended in practice.



Figure 11.6: District heating supply and return temperature and flow rate at a substation for small office buildings in Toronto during a week in January. FC/RAD=2

11.2 Seasonal and Yearly Averaged Performance Data

Similar to the home cases, the seasonal and yearly performance data were generated from the simulation results. As mentioned in Section 7.2.5, the small office buildings were simulated for an entire heating season from September to May. The summer period was not simulated, as domestic hot water load was not provided by the DH system.

The three seasons were defined in the following way:

Fall	-	September, October and November
Winter	_	December, January and February
Spring	_	March, April and May

Results for the Original Design Case (FC/RAD=0.5)

Tables 11.1 and 11.2 summarize the seasonal and yearly flow-weighted average ΔT and total system flow (335 small office buildings/substations) for Amsterdam and Toronto respectively at FC/RAD ratio of approximately 0.5. The ΔT improvement in °C and percentage of flow reduction relative to Case 1 are also shown in the tables.

	Fall		Winter	Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	
	ΔT		ΔT		ΔT		ΔT		
Case1	55.83	-	55.13	-	55.27	-	55.36	-	
Case2	56.47	0.64	56.77	1.64	55.91	0.64	56.42	1.06	
Case2a	57.25	1.42	58.06	2.93	57.13	1.86	57.55	2.19	
	Flow	%	Flow	%	Flow	%	Flow	%	
	(l/s)		(l/s)		(l/s)		(l/s)		
Case1	80.7	-	130.4	-	95.3	-	101.8	-	
Case2	79.6	1.36	126.0	3.37	94.0	1.36	99.6	2.16	
	75.8	6.07	121.5	6.83	89.3	6.30	95.2	6.48	

Table 11.1: Seasonal flow-weighted average district heating temperature difference (ΔT) and average system flows for small office buildings in Amsterdam. FC/RAD=0.5

D.I.: Degree (°C) Improvement relative to Case1.

%: Percentage of system flow reduction relative to Case1.

<i>Table 11.2: Seasonal flow-weighted average district heating temperature difference</i> (ΔT) <i>and</i>
average system flows for small office buildings in Toronto. FC/RAD=0.5

	Fall		Winter		Spring		Full Year	
	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.	F.W.A.	D.I.
	ΔT		ΔΤ		ΔT		ΔT	
Case1	54.87	-	55.51	-	54.99	-	55.21	-
Case2	55.13	0.26	58.55	3.04	55.76	0.77	56.89	1.68
Case2a	55.96	1.09	59.46	3.95	56.68	1.69	57.80	2.59
	Flow	%	Flow	%	Flow	%	Flow	%
	(l/s)		(l/s)		(l/s)		(l/s)	
Case1	93.6	-	192.2	-	118.9	-	134.2	-
Case2	92.8	0.85	181.0	5.83	116.8	1.77	129.6	3.43
Case2a	88.3	5.66	177.4	7.70	112.6	5.30	125.5	6.48
F.W.A. ∆T:	Flow-weighted	Average ΔT , (°C).					
D.I.: Degree	e (°C) Improver	ment relative to	Case1.					
% [·] Percent	age of system f	low reduction r	elative to Case	e1.				

The results shown in the tables indicate that the cascaded system (Case 2) has higher DH temperature difference in all seasons. Overall, the cascaded system improved the ΔT by 1.1°C in Amsterdam and 1.7°C in Toronto. Due to the higher ΔT , Case 2 resulted in lower system flow. The system flow reduction was 2.2% in Amsterdam and 3.4% in Toronto.

The results shown in the tables also indicate that the significant ΔT improvement occurred in the winter, when the space heating loads were high, both for Amsterdam and Toronto. The wintertime ΔT of Case 2 increased by 1.6°C in Amsterdam and approximately 3°C in Toronto. As explained previously, at high space heating loads, the radiator water return temperature was high and it was further cooled by the ventilation air in the cascaded system. Under these conditions, very little or none of the high temperature water (secondary water) was required. Due to the high ΔT improvement in the winter, a large system flow reduction occurred in the winter as well. A DH flow is limiting during the winter making substantial system capacity expansion possible.

In the summer, fall and spring, it seems that Case 2 had a lower degree of improvement in ΔT and system flows compared to Case 1. In these seasons, the space heating loads were low, and more secondary water was required in the fan-coil system as the return water from the radiator was not sufficient to meet the ventilation heating demand. Therefore, during these seasons, Case 2 resulted in a small ΔT improvement compared to the parallel system.

Tables 11.1 and 11.2 show that the series connection (Case 2a) has the highest ΔT and lowest system flow rate compared to the parallel connection (Case 1) and cascaded system (Case 2). However, any conclusions should be drawn carefully. As mentioned previously, the ventilation air outlet temperatures from Case 2a were lower than the set-point of Case 1 (25°C) because it was not controlled. The trade-off for the lower DH return temperatures for Case 2a was reduced comfort level.

It should also be mentioned again that the size of all components was kept the same in all case studies. This means that the fan-coil was oversized in the parallel connection case. The ΔT of the parallel system would be smaller than the value shown in Tables 11.1 and 11.2 if the fan-coils were sized for the corresponding supply temperature at design conditions. In other words, the increase of ΔT and the reduction of the system flow due to cascading would have been higher.

Tables 11.3 and 11.4 summarize the yearly average data for Amsterdam and Toronto. The data shown in the tables are yearly average flow-weighted DH temperature differences at the DH substations and yearly DH water consumption, building thermal energy consumption, fuel consumption, DH network heat loss, pumping power as well as electricity production. Comparisons between the parallel connection (Case 1) and cascaded systems (Case 2 and Case 2a) are also shown in the tables.

	Case1	Case2	Comparisor	ı	Case2a	Comparison	
				%			%
Flow-weighted							
average ΔT at	55.36	56.42	1.06	-	57.55	2.19	-
substations (°C)							
DH water volume							
consumption (m ³ /yr)	2402000	2357800	-44200	-1.84	2246600	-155400	-6.47
Building energy							
consumption	152885	152885	-	-	148610	-4275	-2.80
(MWh _{ttr} /yr)							
Thermal energy							
supplied to the DH	160880	160520	-360	-0.22	156220	-4660	-2.71
system (MWh _{th} /yr)							
Network heat losses	7995	7635	-360	-4.50	7610	-385	-1.06
(MWh/yr)							
Pumping energy	166	155	-11	-6.63	144	-22	-13.2
(MWh/yr)							
Electricity production							
(MWh/yr)	123670	123730	60	0.05	123980	310	0.25
Fuel consumption	283140	283140	0	-	283110	-30	-0.01
(MWh/yr)							

Table 11.3: Simulation and comparison results of case studies for small office buildings in Amsterdam (335 office buildings/substations) FC/RAD=0.5

From Tables 11.3 and 11.4 it can been seen that the cascaded system (Case 2), resulted in lower DH water consumption, lower network heat losses, reduced pumping energy demand and slightly higher net electricity production. This is due to the higher ΔT resulting from cascading. The DH water consumption was reduced by 1.8% in Amsterdam and 2.9% in Toronto. The network heat losses were reduced by 4.5% in Amsterdam and 3.4% in Toronto. The pumping energy demand was reduced by 6.6% in Amsterdam and 9.9% in Toronto.

Table 11.4: Simulation and comparison results of case studies for small office buildings in Toronto (335 office buildings/substations) FC/RAD=0.5

	Case1	Case2	Comparisor	ı	Case2a	Comparison	
				%			%
Flow-weighted							
average ΔT at	55.2	56.89	1.69	-	57.8	2.6	-
substations (°C)							
DH water volume							
consumption (m ³ /yr)	3166100	3073000	-93100	-2.94	2959800	-206300	-6.52
Building energy							
consumption	200928	200928	-	-	196680	-4248	-2.11
(MWh _{th} /yr)							
Thermal energy							
supplied to the DH	211150	210800	-350	-0.17	206545	-4605	-2.02
system (MWh _{th} /yr)							
Network heat losses	10222	9872	-350	-3.42	9865	-357	-0.12
(MWh/yr)							
Pumping energy	222	200	-22	-9.91	191	-31	-14.0
(MWh/yr)							
Electricity production							
(MWh/yr)	186600	186680	80	0.04	187000	400	0.21
Fuel consumption	421280	421280	0	-	421260	-20	-0.005
(MWh/yr)							

The fuel consumption for Cases 1 and 2 was approximately the same. This was because fuel flow to the combined cycle gas turbine was kept constant. The fuel consumption reduction of the peaking boiler resulted from the lower heat losses in the DH return line.

It should be pointed out again that a comparison between Cases 2a and Case 1 should be made with caution. As mentioned above, the ventilation air outlet temperatures from Case 2a were lower than the set-point of Case 1 (25° C) because it was not controlled. The building energy consumption was therefore lower when compared to Case 1 and Case 2.

It should be noted that the building energy consumption obtained from the simulation data for Case 1 and Case 2 were slightly different. The differences resulted from the transients in the simulation and control methodology. However, the difference was very small (less than $\pm 0.05\%$ both for Amsterdam and Toronto). The average value was used in the tables in order to keep the income from selling the heat the same in the economic analysis, to be discussed in the following section.

11.3 Annual Cost Comparisons

Since the higher FC/RAD ratios are not recommended, only the Case of FC/RAD=0.5 will be discussed here. The cascaded system had lower DH return temperatures. Consequently, it resulted in lower heat losses, lower pumping energy demand and higher electricity production. The annual cost savings of the cascaded system (Case 2) were compared to the reference case.

The annual net equivalent worth of a DH system is the difference between the income and expenses and can be calculated by:

Annual net equivalent worth $=$ (a):	Income from sale thermal energy to consumers
+ (b):	Income from sale electricity to grid
- (c):	Fuel costs
- (d):	Operational and maintenance costs
- (f):	Labour costs
- (g):	Annual repayment of capital costs

The operational and maintenance costs, as well as the labour costs, were considered the same for both cases. Although the cascaded system has one more control valve installed in the DH substations, the additional costs of those valves is negligible compared to the CHP plant capital cost which is over 75 million dollars (assuming $3000/kW_e$). Therefore, the total capital cost was considered the same for the two cases.

Therefore, only income from the sale of thermal energy and electricity, and fuel cost for the two cases were calculated and compared. The cost comparisons are shown in Tables 11.3 and 11.4 at a FC/RAD ratio of 0.5 for Amsterdam and Toronto. The prices listed in the tables were based on the recent Canadian market. The DH energy price was price charged to the consumers and the electricity price was the selling price to the grid. The thermal and fuel consumption values can be found in Tables 11.3 and 11.4. The electrical energy available for sale to the grid is the electrical energy produced subtracted by the pumping energy. These data can also be found in these tables.

Table 11.5: Annual cost comparison results of case studies for small office buildings in Amsterdam

(335 buildings/substations) FC/RAD=0.5

	Price	Annual Income/(Cost)		Difference
	(\$/MWh)	(\$)		(\$)
		Case1	Case2	
DH Energy	70	\$10,701,950	\$10,701,950	-
Elec. Energy	50	\$6,175,200	\$6,178,750	\$3,500
Fuel Cost	16	(\$4,530,240)	(\$4,530,240)	-
Total				\$3,500

Table 11.6: Annual cost comparison results of case studies for small office buildings in Toronto (335 buildings/substations) FC/RAD=0.5

	Price	Annual Income/(Cost)		Difference
	(\$/MWh)	(\$)		(\$)
		Case1	Case2	
DH Energy	70	\$14,064,960	\$14,064,960	-
Elec. Energy	50	\$9,318,900	\$9,324,000	\$5,100
Fuel Cost	16	(\$6,740,480)	(\$6,740,480)	-
Total				\$5,100

The above tables show that the cascaded system has a higher overall annual income compared to the parallel system, both for Amsterdam and Toronto. The difference resulted from the lower pumping energy of the cascaded system. The annual cost savings from the cascaded system was \$5,100 for Toronto and \$3,500 for Amsterdam.

Conclusions

In this study, the principle of cascading loads in building heating systems to increase the temperature difference between the supply and return lines in a district heating system has been examined.

The results of the study show that for all cases studied, using different building types in different climates, the DH Δ T increased by cascading of the heating loads compared to a similar, parallel connected system.

For large multi-functional buildings, there were eight thermal loads requiring different supply temperature levels. Therefore, it provided the greatest opportunity for maximizing ΔT .

The simulation results from the large multi-functional buildings show that the cascaded systems have a higher DH temperature difference during all seasons. Overall, the cascaded system improved the ΔT by 4°C for Case 2 and 5°C for Case 3. Due to this higher ΔT , the cascaded systems resulted in low system flows. The flow reductions achieved were up to 8%.

The results also indicated that significant performance improvements occurred in the fall and spring. In these two seasons, there was over 4.5° C increase in Δ T and more than 7% in system flow reduction for Case2, and with over 6°C increase in Δ T and approximately 9% in system flow reduction for Case3. In the winter, the Δ T improvement was about 3.6°C and flow reduction was 5.6%. In the summer, the cascaded systems had lower performance improvement compared to the other seasons; 2°C increase in Δ T and approximately 3% in system flow reduction. In the summer, there were no or very small loads in the heating subsystems and the effect of the cascading was less significant.

It should be emphasized that the size of all components was kept the same in all cases. The ΔT of the parallel system would have been smaller had the components been sized for their corresponding supply temperature at design conditions. In other words, the improvement of ΔT and reduction of system flow due to cascading would have been more pronounced. This was done on recommendation of the experts group to show the effect of cascading on the ΔT only.

Connection scheme 3 in the large multi-functional building had a more balanced power ratio between the first and second level, both during daytime and nighttime, compared to connection scheme 2. This resulted in Case 3 having a higher average ΔT and a lower system flow compared to Case 2 system.

For single-family homes, the increase of ΔT , resulting from the cascading, was relatively small due to the ventilation load (fan-coil load, which operated at the lower temperature level) being small compared to the radiator load. The largest ΔT improvement and system flow reduction occurred in the winter.

For multi-family home blocks, it was found that the increase of ΔT was quite significant by cascading the domestic hot water (DHW) load with the space heating loads. Mixing of DHW recirculation water with DHW after the pre-heater also improved the system performance compared to the parallel system where DHW re-circulation water was mixed with the cold municipal water. The largest ΔT improvement (4.3°C both for Toronto and Amsterdam) and system flow reduction (8% for Toronto and 5.8% for Amsterdam) occurred in the summer. The yearly ΔT increase was 1.2°C for Toronto and 1.4°C for Amsterdam. The yearly average flow reduction was 2.4% for Toronto and 2.5% for Amsterdam.

For small office buildings, when the fan-coil to radiator power ratio (at design conditions) was about 0.5, the system performance was improved quite significantly by cascading these two heating loads. The largest ΔT improvement (3°C for Toronto and 1.6°C for Amsterdam) and system flow reduction (5.8% for Toronto and 3.4% for Amsterdam) occurred in the winter. Flow is limited during winter, making the DH system capacity expansion possible. Overall, the yearly average ΔT increases were 1.7°C for Toronto and 1.1°C for Amsterdam. The yearly average flow reduction was 3.4% for Toronto and 2.2% for Amsterdam.

Systems with different fan-coil to radiator power ratio (FC/RAD=1 and 2), at design conditions, were studied for the small office building cases. The results showed that if the power ratio of the fan-coil to radiator increased, the effects of the cascading were reduced. The Δ T improvement was relatively small when the FC/RAD ratio was 1 or 2. However, the return temperature did decrease for these cases.

Due to relatively lower DH return temperatures, the cascaded systems, for all cases in different building types, resulted in lower network heat losses, lower pumping energy demand, higher net electricity production and therefore higher revenue compared to the parallel systems.

It can be concluded from the results of this study, that yearly overall improvement in ΔT , resulting from cascading of different thermal loads, largely depended on the following three factors:

- required temperature level of different thermal loads,
- magnitude of different loads, and
- time-of-day usage patterns of different loads.

Therefore, it is important keep the above three factors in mind when design a cascading system.

A system with thermal loads with different temperature levels provides the potential for cascading to maximizing ΔT . Heating subsystems requiring a high temperature should be placed in the first level where the systems are supplied by high temperature water. Heating subsystems requiring low temperature levels can be cascaded to the second level where the return water from the first level can be used. Multi-level (more than two levels) cascading may result in even greater improvements, but more complex control systems will be required.

To achieve maximum improvement in ΔT , the thermal loads in a cascaded system should be arranged in a way that the load ratio between different levels is balanced appropriately during the majority of the time throughout the year. Usually, the magnitude of the total loads in the upper level should be higher than the lower level in order to avoid using high temperature water in the lower level as much as possible. However, too high or low a load ratio between the upper and lower level will cause either water from the upper level to bypass the lower level or require high temperature water in the lower level. Either way, ΔT will decrease. It is expected, for a given system and climate, that there is optimum load ratio between each level, although this was beyond the scope of this study.

The time-of-day use patterns of different loads should also be considered while designing a cascaded system. Some thermal loads are required in heating season only, such as space heating. Some thermal loads are required throughout the year, and some loads are required only during certain times during the day or night. To achieve maximum yearly improvement in ΔT , the thermal loads in a cascaded system should be arranged in a way that the load ratio between different levels is balanced as well as possible.

An additional benefit of the simulation studies was the development of algorithms that greatly speeded up the process, enabling a full year of heating system operation to be simulated in minutes. These use two-dimensional table lookup, with the reference table derived from either fundamental principles or manufacturer's data.

There is potential for further speedup, leading to the possibility of the simulation programs being used for system design optimization at the routine engineering level.

Finally, it can be observed that the benefits of cascading can be realized in both maritime and continental climates. Cascading of building heating systems has its greatest effect when all loads are required for a major portion of the year. Cascading designs that favour wintertime flow reductions facilitate system expansion, since flow is limited in the winter.

References

ASHRAE (2001): "International Weather for Energy Calculations (IWEC Weather Files), User's Manual", ASHRAE, 2001.

Bøhm, B. (1988): "Energy Economy of Danish District Heating Systems – a Technical and Economic Analysis", Technical University of Denmark, 1988. Frederiksen, S. (1982): "A Thermodynamic Analysis of District Heating", Lund Institute of Technology, Sweden, 1982.

Hansson, T. (1990): "Operational Optimization of District Heating Systems", M.Sc. Thesis, Technical University of Denmark, 1990

Onno, T. (1998): "Optimization of Building Heating and Domestic Hot Water Systems Supplied by Low Temperature District Heating Water", Gagest Developments Inc., Ottawa, Canada, 1998.

Snoek, C., Yang, L. and Onno, T. (1997): "Cost-performance Optimization of Low Temperature Heating Systems by Statiscal Analysis of Full-year Simulation", 6th International Symposium on District Heating and Simulation, 1997, Reykjavik, Iceland.

Snoek, C. and Yang, L. (1999): "Optimum Connection of Multiple Thermal Loads in Buildings Connected to District Heating", 7th International Symposium on District Heating and Cooling, 1999, Lund, Sweden.

Van der Meulen, S.F. et al. (1991): "Investing District Heating", Final Summary Report, Twente University, Holland, 1991.

Volla, R. et al. (1996): "Efficient Substations and Installations", IEA District Heating and Cooling Project Report (Annex IV) published by Novem, Sittard, The Netherlands, 1996.

Woods, P. et al. (1999): "Optimisation of Operating Temperatures and an Appraisal of the Benefits of Low Temperature District Heating", IEA District Heating and Cooling Project Report (Annex V) published by Novem, Sittard, The Netherlands, 1999.

Yang, L. (1994): "District Heating House Stations with Hot Water Storage – Simulation and Evaluation of Dynamic Performance", Ph.D. Dissertation, Centre for District Heating Technology, Laboratory of Heating and Air Conditioning, Technical University of Denmark, 1994.

Zhao, H. (1995): Analysis, Modelling and Operation Optimization of District Heating Systems, Ph.D. dissertation, Technical University of Denmark, 1995.

Appendix A Heat Production Plant Model

A.1 Model Description

As mentioned in Chapter 5, a combined cycle gas turbine CHP plant with a single extraction/condensation steam turbine was used as the heat production plant. Figure A.1 shows the flowchart of the plant.



Figure A.1: Flowchart of a combined cycle gas turbine CHP plant with a single extraction/condensation steam turbine.

The cycle operates as follows: air at temperature T_{g1} and pressure P_{g1} is compressed in a compressor to a temperature of T_{g2} and pressure of P_{g2} . The compressed air and natural gas is injected into a combustion chamber and combusted. The exhaust gas is then expanded in a gas turbine. The expanded gas enters a heat recovery steam generator (HRSG), where part of the heat in the exhaust gas is recovered to produce steam. In order to further utilize the heat in the exhaust gas, an economizer is used to decrease the stack gas temperature by heating the DH return water.

The steam generated in the HRSG is used to produce additional electricity using a steam turbine. Part of the steam is extracted from the steam turbine to heat the DH water in a condenser. The remaining steam is further expanded in the steam turbine to the condenser pressure level.

The district heating return water temperature is first heated in the economizer and then in the DH condenser by the steam extracted from the steam turbine. If the DH water temperature leaving the condenser is lower than the required level, it will be raised to its set-point by a peaking boiler. It should be noted that no attempts were made here to optimize the design of the CHP plant as it was beyond the scope of this study.

Computer models for each component shown in Figure A.1 were developed based on general thermodynamic principles and heat transfer theory. These models are described below.

Compressor

It was assumed that the air is a semi-perfect gas. Compression in the compressor is adiabatic. The outlet air temperature from the compressor depends on the compression ratio and inlet air temperature. The theoretical outlet temperature was calculated by:

$$T_{g2}' = T_{g1} (\frac{P_{g2}}{P_{g1}})^{\frac{\gamma_a - 1}{\gamma_a}}$$

where:

T_{g1}	-	air inlet temperature to the compressor, °K
T _{g2} '	-	theoretical air outlet temperature from the compressor, °K
P_{g1}	-	air inlet pressure to the compressor, kPa
P_{g2}	-	air outlet pressure from the compressor, kPa
γ _a -		specific heat ratio $\gamma_a = Cp_a/Cv_a$
Cp _a	-	specific heat of air at constant pressure, kJ/kg°K
Cva	-	specific heat of air at constant volume, kJ/kg°K.

The actual work done to the compressor, Wc_{act} (kW), was determined by:

$$Wc_{act} = \frac{W_c}{\eta_c} = \frac{m_a C p_a (T_{g2}' - T_{g1})}{\eta_c}$$

where:

W _c	-	theoretical work required by the compressor, kW
m _a	-	flow rate of air, kg/s
η_c	-	efficiency of the compressor.

The actual air outlet temperature from the compressor, T_{g2} , was obtained from the equation below:

$$T_{g2} = T_{g2}' + \frac{(Wc_{act} - Wc)}{m_a C p_a}$$
$$= T_{g2}' + (T_{g2}' - T_{g1})(\frac{1}{\eta_c} - 1)$$

Combustion Chamber

The heat generated by natural gas combustion was Q_{fuel}:

$$Q_{fuel} = m_{fuel} \cdot q_{LHV}$$

where:

m _{fuel}	-	natural gas flow rate, kg/s
q_{LHV}	-	lower heating value of natural gas, kJ/kg.

The outlet temperature of the hot flue gases from the combustion chamber was then calculated by:

$$T_{g3} = T_{g2} + \frac{Q_{fuel}}{m_{exgas}Cp_{exgas}}$$

where:

m_{exgas} - flow rate of the flue gases in the combustion chamber, kg/s
Cp_{exgas} - average specific heat capacity of the flue gases in the combustion chamber, kJ/kg°K.

It was assumed that the theoretical air/fuel ratio was R_{af}:

$$R_{af} = \frac{(m_a)_o}{m_{fuel}}$$

Where $(m_a)_o$ is the theoretical air flow required for complete combustion. Usually, excess air is used in the combustion chamber. The flue gases in the combustion chamber are the sum of the air and natural gas flow rates:

$$m_{exgas} = m_{fuel} + m_a = m_{fuel} + (1 + P_{exair}/100) \cdot (m_a)_o$$

where P_{exair} is the percentage of excess air flow. The above equation can be re-written as:

$$m_{exgas} = m_{fuel} + (1 + P_{exair} / 100) \cdot R_{af} m_{fuel}$$
$$= m_{fuel} \cdot (1 + R_{af} + R_{af} P_{exair} / 100)$$

The equation for calculating the outlet temperature of the hot flue gases from the combustion chamber, shown above, can be expressed as:

$$T_{g3} = T_{g2} + \frac{q_{LHV}}{(1 + R_{af} + R_{af} \cdot P_{exair} / 100) \cdot Cp_{gas}}$$

Gas Turbine

Similar to the compressor model, it was assumed that the exhaust gas was a semi-perfect gas and the expansion in the gas turbine was adiabatic. The theoretical outlet temperature from the gas turbine was calculated by:

$$T_{g4}' = T_{g3} (\frac{P_{g4}}{P_{g3}})^{\frac{\gamma_g - 1}{\gamma_g}}$$

where:

T _{g3}	-	exhaust gas inlet temperature to the gas turbine, °K
T _{g4} '	-	theoretical exhaust gas outlet temperature from the gas turbine, °K
P_{g3}	-	exhaust gas inlet pressure to the gas turbine, kPa
P_{g4}	-	exhaust gas outlet pressure from the gas turbine, kPa
$\gamma_{\rm g}$	-	specific heat ratio $\gamma_{g} = Cp_{exgas}/Cv_{exgas}$
Cp _{exgas}	-	specific heat of exhaust gas at constant pressure, kJ/kg°K
Cv _{exgas}	-	specific heat of exhaust gas at constant volume, kJ/kg°K.

The actual work generated by the gas turbine, Wt_{act} (kW), was determined by the following equation:

$$Wt_{act} = Wt\eta_t = m_{exgas}Cp_{exgas}(T_{g3} - T_{g4}')\eta_t$$

where η_t is the efficiency of the turbine.

The actual exhaust gas outlet temperature from the gas turbine, Tg4, was obtained from:

$$\begin{split} T_{g4} &= T_{g4}' + \frac{(Wt_{act} - Wt)}{m_{exgas} C p_{exgas}} \\ &= T_{g4}' + (T_{g3} - T_{g4}')(1 - \eta_t) \end{split}$$

Steam Turbine

It was assumed that the steam from the steam boiler was superheated at a temperature of T_{st1} and a pressure of P_{st1} . The steam extracted from the steam turbine was saturated at a pressure of P_{st2} and a corresponding temperature of T_{st2} .

Theoretically, the steam expansion in the steam turbine is isentropic, i.e. the entropy of inlet and outlet steam is the same. This can be expressed by the following equation, Sears (2000):

$$Sst_{T_{st1}} = R_{st}'Sst_{Tst2} + R_{w}'Sw_{Tst2}$$

where:		
Sst _{Tst1}	-	entropy of steam at temperature T_{st1} and pressure P_{st1} , kJ/kg°K
Sst _{Tst2}	-	entropy of saturated steam at temperature T_{st2} , kJ/kg°K
Sw _{Tst2}	-	entropy of saturated water at temperature T _{st2} , kJ/kg°K
R _{st} '	-	steam ratio in isentropic condition
R _w '	-	water ratio in isentropic condition, R_w '=1- R_{st} '.

The entropy values of saturated steam, water, and superheated steam can be found in steam tables (Ketin 1979).

By rearranging the above equation, the isentropic steam ratio was calculated by the equation below:

$$R_{st}' = \frac{Sst_{Tst1} - Sw_{Tst2}}{Sst_{Tst2} - Sw_{Tst2}}$$

It should be noted that if the turbine inlet is highly superheated, the R_{st} value calculated from the above equation could be greater than 1. This means that the extracted steam from the steam turbine is still superheated. The actual steam temperature at pressure P_{st2} can then be determined from the steam tables by assuming that the entropy at this condition equals the entropy of inlet steam, Sst_{Tst1} . However, in our situation, the temperature of superheated steam from the boiler was such that the steam at extraction or exit (for back-pressure unit) of the turbine is saturated.

The enthalpy of the outlet steam, for isentropic conditions, then can be calculated by:

$$H_{Tst2}' = R_{st}' Hst_{Tst2} + R_{w}' Hw_{Tst2}$$

where:

H _{Tst2} '	-	enthalpy of extracted steam for isentropic conditions, kJ/kg
Hst _{Tst2}	-	enthalpy of saturated steam at temperature T _{st2} , kJ/kg
Hw _{Tst2}	-	enthalpy of saturated water at temperature T _{st2} , kJ/kg.

The actual outlet enthalpy, H_{Tst2} (kJ/kg), was obtained by the equation below:

$$H_{Tst2} = Hst_{Tst1} - \eta_{st}(Hst_{Tst1} - H_{Tst2}')$$

where:

 $\begin{array}{lll} H_{Tst1} & - & \mbox{inlet enthalpy of the superheated steam at temperature } T_{st1} \mbox{ and pressure } P_{st1}, kJ/kg \\ \eta_{st} & - & \mbox{efficiency of the steam turbine.} \end{array}$

The actual outlet steam quality then can be calculated by:

$$R_{st} = \frac{H_{Tst2} - Hw_{Tst2}}{Hst_{Tst2} - Hw_{Tst2}}$$

The electricity generated by the steam turbine, El_{st} (kW_e), was obtained by the equation below:

$$El_{st} = m_{st}(H_{Tst1} - H_{Tst2})\eta_{sts}$$

where $\eta_{\mbox{\tiny stg}}$ was the combined efficiency of the generator and the gear box.

The same principle was applied to the calculation of the electricity generated in the two-stage steam turbine. A fraction of the steam was expanded from T_{st1}/P_{st1} to T_{ex}/P_{ex} and the remaining amount of steam was expanded from T_{st1}/P_{st1} to T_{st2}/P_{st2} . The total electricity generated in the steam turbine was the sum of the two parts.

Heat Recovery Steam Generator

The condensed water from the condenser is heated in the HRSG by the exhaust gas from the gas turbine. The pressure in the steam boiler was kept constant.

To simplify the analysis, it was assumed that the condensed water was first heated to the boiling point at pressure P_{st1} and vaporized, and then was heated to superheated steam with a temperature of T_{st1} , as shown in Figure A2. The temperature T_{g5} " should be higher than T_{st1} '. The difference between the two temperatures is the so called pinch point.



Figure A.2: Illustration of Temperature Profile of Exhaust Gas and Water/Steam in the HRSG.

The total energy recovered from the exhaust flue gas is:

$$Q_{HRSG} = m_{st}Cp_{st}(T_{st1} - T_{st1}') + m_{st}H_{st1}' + m_{st}Cp_{w}(T_{st1}' - T_{cond})$$

where:

m _{st}	-	steam flow rate, kg/s
Cp _{st}	-	specific heat capacity of steam, kJ/kg°C
Cp _w	-	specific heat capacity of water, kJ/kg°C
H _{st1} '	-	latent heat of steam at pressure P _{st1} , kJ/kg
T _{cond}	-	HRSG feed water temperature, °C.

The superheated steam temperature T_{st1} and boiling temperature at pressure P_{st1} were known variables, as well as the steam flow rate m_{st} and the boiler feed water temperature T_{cond} . The outlet exhaust gas temperature from the HRSG, T_{g5} , was calculated using the following equation:

$$T_{g5} = T_{g4} - \frac{Q_{HRSG}}{m_{exgas}Cp_{exgas}}$$

Economizer

The economizer was used to further utilize the heat in the exhaust gas by preheating the DH return water. It was designed so that the minimum stack temperature was kept at T_{g6} , e.g. 90°C, to avoid condensation.

The DH outlet temperature from the economizer was calculated by:

$$T_{r}^{'} = T_{r} + \frac{m_{exgas}Cp_{exgas}(T_{g5} - T_{g6})}{m_{dh}Cp_{dh}}$$

where

m _{dh}	-	district heating flow rate, kg/s
Cp _{dh}	-	specific heat capacity of district heating water, kJ/kg°C
Γ _r	-	district heating return temperature, °C
Γ _r '	-	district heating temperature from the economizer, °C
Γ_{g6}	-	stack gas temperature, °C

Condenser

The CHP plant was designed to cover P percent of the district heating peak load. Therefore, the condenser has a maximum capacity of $Q_{end,max}$:

$$Q_{cnd,\max} = P\% Q_{dh,peak}$$

If the district heating load was higher than the condenser's maximum capacity, the condenser capacity was:

$$Q_{cnd} = Q_{cnd, \max}$$

If the district heating load was lower than the condenser's maximum capacity, the required condenser capacity was:

$$Q_{cnd} = Q_{dh}$$

where Q_{dh} was the district heating load and was calculated by:

$$Q_{dh} = m_{dh} C p_{dh} (T_s - T_r)$$

where T_s was the required district heating supply temperature and T_r , was the inlet DH temperature to the condenser.

The district heating water is heated by the extraction steam in the condenser to this temperature:

$$T_s' = T_r' + \frac{Q_{cnd}}{m_{dh}Cp_{dh}}$$

where:

Q _{cnd}	-	heat generated by condensing the steam extracted from the steam turbine, kW
T _s '	-	district heating water temperature leaving the condenser, °C
T _r '	-	district heating return water temperature from the economizer, °C
m _{dh}	-	district heating water flow rate, kg/s
Cp_{dh}	-	specific heat capacity of DH water, kJ/kg°C

It was assumed that extracted steam was condensed and further cooled to a lower temperature, T_{cond} , in the condenser. When the extracted steam temperature, T_{st2} or T_{ex} was known, the steam flow rate was then determined by the equation below:

$$m_{st} = \frac{Q_{cnd}}{R_{st}(Hst_{Tst2} - Hw_{Tst2}) + Cp_w(T_{st2} - T_{cond})}$$

In the above equation, R_{st} was the steam ratio of the saturated steam extracted from the turbine, c.f. the description of the steam turbine model. T_{cond} was the condensed water temperature leaving the condenser and was assumed to be 5°C above the DH water entry temperature.

Electricity and Thermal Efficiency

The total electricity generated by the combined cycle gas turbine plant was calculated by the following equation:

$$El = El_{gt} + El_{st}$$
$$= (Wt_{act} - Wc_{act})\eta_{tg} + El_{st}$$

In the above equation, η_{tg} was the combined efficiency of the generator and gear box. The calculations of Wt_{act} and Wc_{act} can be found in the descriptions of the turbine and compressor models respectively. The determination of electricity generated by steam turbine, E_{el} , was described in the steam turbine model.

The electricity efficiency was determined by:

$$\eta_{el} = \frac{El}{Q_{fuel}}$$

The thermal efficiency was determined by:

$$\eta_{th} = \frac{Q_{cnd}}{Q_{fuel}}$$

And the total efficiency of the combined cycle was calculated by:

$$\eta_{chp} = \frac{El + Q_{cnd}}{Q_{fuel}}$$

Peaking Boiler

A natural gas-fired boiler was used to satisfy the remaining district heating load. The required natural gas flow rate was determined by:

$$m_{f,pb} = \frac{m_{dh}Cp_{dh}(T_s - T_s')}{q_{LHV}\eta_{pb}}$$

where:

m _{f,pb}	-	required natural gas flow rate for the peaking boiler, kg/s
η_{pb}	-	efficiency of the peaking boiler.

A.2 Model Verification

The heat production model was constructed using Simulink[®] software (by Math Works Inc.). Other simulation software, PRO/II from Simulation Science Inc., was used to verify the production model. PRO/II is a comprehensive process simulation software used for steady state simulation of refinery process, chemical process, batch process, etc. This software is used as a tool for design, troubleshooting and yield optimization in chemical and refinery industries. Besides models for special components used in these processing industries, it contains models for general components, such as compressors, expanders (turbine), reactors, simple heat exchanger, etc. With these standard block models, it was possible to build a simulation system for the combined cycle gas turbine CHP plant.

A verification test was carried out by comparing the simulation results from the Simulink[®] and the SimSci PRO/II models. The input values were arbitrarily selected. Average inlet and outlet physical properties values obtained from the SimSci PRO/II model were used in the Simulink[®] model. The comparison results are shown in Table 1.

The deviations shown in the table were calculated by:

$$Deviation = \frac{Simulink - SimSci}{SimSci} \cdot 100\%$$

Table A.1: Comparison	results between	Simulink® and	l SimSci PRO/II mod	lel
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		Outputs		
	Inputs	Simulink	SimSci PRO/II	Deviation
	T _{g1} =15 ℃	T _{g2} =185.62 °C	T _{g2} =188.05 °C	-1.29%
	P _{g2} /P _{g1} =4	Wc=325.96 kW	Wc=330.63 kW	-1.41%
Compressor	m _a =2.36 kg/s	Wc _{act} =407.45 kW	Wc _{act} =413.29 kW	-1.41%
	γ _a =1.378			
	η _c =0.8			
	m _{fuel} =0.04 kg/s	Q _{fuel} =1909.67 kW	Q _{fuel} =1910.47 kW	-0.04%
Combustion	q _{LHV} =802.72 kJ/kg	T _{g3} =873.00°C	T _{g3} =874.29°C	-0.15%
Chamber	m _{exgas} =2.4 kg/s			
	P _{g3} /P _{g4} =4	T _{g4} =613.94°C	T _{g4} =613.96°C	-0.01%
	γ _a =1.315	Wt=928.63 kW	Wt=935.24 kW	-0.71%
Gas Turbine	m _{exgas} =2.4 kg/s	Wt _{act} =742.91 kW	Wt _{act} =748.19 kW	-0.71%
	η _t =0.8	El _{gt} =318.69 kW	El _{gt} = 318.16 kW	+0.17%
	η _{tg} =0.95			
	T _{st3} =75 °C	T _{g5} =134.72°C	T _{g5} =131.72°C	+2.28%
Heat	T _{st1} =200 °C	Q _{st} =1276.21 kW	Q _{st} =1284.72 kW	-0.66%
Recovery	P _{st1} =1000 kPa			
Steam	m _{st} =0.51 kg/s			
Generator	m _{exgas} =2.4 kg/s			
	m _{st} =0.51 kg/s	R _{st} =89.93%	R _{st} =91.31%	-1.51%
Steam	T _{st1} =200 °C	E _{st} =208.45 kW	E _{st} =190.83 kW	+9.23%
Turbine	T _{st2} =75°C (P _{st2} =38.55			
	kPa)			
	η_{st} =0.8			
	η_{stg} =0.95			
	T,'=55 °C	Q _{cnd} =1064.30 kW	Q _{cnd} =1084.44 kW	-1.86%
Condenser	m _{dh} =17.05 kg/s	T _s '=69.90 °C	T _s '=70.21 °C	+0.44%
	m _{st} =0.51 kg/s			
	T _{st2} =75 °C			
Electricity		η _{el} =27.60%	η _{el} =26.64%	+3.60%
Efficiency		(El=527.14 kW _e)	(El=508.99 kW _e)	
Thermal		η_{th} =55.73%	η_{th} =56.76%	-1.81%
Efficiency				
Total Cycle		$\eta_{chp} = 83.34\%$	η_{chp} =83.40%	-0.07%
Efficiency				

The comparison results showed that the deviations between the two models were within $\pm 4\%$, except for the deviation for electricity generated by the steam turbine which was 9.23%. It was found that the entropy and enthalpy used for inlet and outlet steam in the steam turbine model were slightly different (within 0.7%) between the two models. The electricity generated by the steam turbine is the difference between the inlet and outlet enthalpy. This difference was less than 10% of the inlet and outlet steam enthalpy values. Therefore, small deviations in the enthalpy values used in the two models resulted in larger deviations in the calculated electricity generated by the steam turbine. However, for the purposes of this study, the Simulink[®] model accuracy was considered acceptable.

A.3 References

Kestin, J., 1979: "A Course in Thermodynamics". McGraw-Hill Book Company.

Sears, P., 2000: Natural Resources Canada, CANMET Energy Technology Centre. Personal communication.

A.4 Nomenclature

С	-	heat capacity, kW/°C
Ср	-	specific heat at constant pressure, kJ/kg
Cv	-	specific heat at constant volume, kJ/kg
El	-	generated electricity, kW _e
Н	-	enthalpy, kJ/kg
m	-	flow, kg/s
Р	-	pressure, bar
q	-	heat value, kJ/kg
Q	-	thermal power, kW
R	-	steam ratio
S	-	entropy, kJ/kg°K
Т	-	temperature, °K or °C
UA	-	overall heat transfer coefficient, kW/°C
W	-	work, kW
ΔT	-	temperature difference, °C
γ	-	specific heat ratio, equals Cp/Cv
η	-	efficiency

Subscript

a	-	air
act	-	actual value
c	-	compressor
chp	-	CHP plant
cnd	-	condenser
dh	-	district heating
el	-	electricity
exair	-	excessive air
exgas	-	exhaust gas
fuel	-	fuel
g	-	gas
gt	-	gas turbine
LHV	-	low heating value
max	-	maximum
min	-	minimum
pb	-	peaking boiler
peak	-	at peak condition
r	-	return
S	-	supply
st	-	steam or steam turbine
stg	-	generator of the steam turbine
t	-	turbine
th	-	thermal
W	-	water

B.1 Description of the Network Model

When the structure of the network, the heat load, and the difference between the supply and return temperatures at each of the consumers are known, it is possible to develop a physical model to perform the dynamic simulation of the DH network. However, if there are several hundred or thousands of consumers connected to a DH network, it requires an enormous amount of effort to build up the model. Therefore, it was decided to use a simplified network model to describe the basic dynamics of the system.

Based on previous studies, it was decided to use a single pair of supply and return pipes, a single consumer and a single bypass to describe a DH network in this study.

The network model was used to calculate the heat losses from the DH pipes and the required pumping power for transporting the DH water. As described previously, an equivalent network with a pair of supply and return pipes were used to represent a DH network. It was assumed that the supply and return pipes had an inner diameter of D_i and outer diameter of D_o . The pipe equivalent length was L and heat loss coefficient was k_1 .

Outlet Temperature and Heat Losses

The simplified analytical solution for calculating the outlet temperature of a pipe can be expressed as, Zhao (1995):

$$T_o(t) = T_i(t-\tau) \cdot e^{-K\frac{V}{f}}$$

In the above equation, V and K were calculated by the following equations:

$$V = V_{w} + V_{s} \frac{\rho_{s} C p_{s}}{\rho_{w} C p_{w}}$$
$$K = \frac{k_{l} L}{k_{l} L}$$

$$\mathbf{X} = \frac{1}{V_w \rho_w C p_w + V_s \rho_s C p_s}$$

where:

T _o	-	outlet temperature, °C
T _i	-	inlet temperature, °C
t	-	at time t, sec.
τ	-	time delay, $\tau = V/f$, sec.
f	-	volume flow rate, m ³ /s
V_{w}	-	volume of water, m ³
Vs	-	volume of pipe material, m ³
$\rho_{\rm w}$	-	water density, kg/m ³
ρ _s	-	pipe density, kg/m ³
Cp _w	-	water specific heat capacity, J/kg°C
Cp _s	-	pipe specific heat capacity, J/kg°C
k _l	-	equivalent heat loss coefficient of pipe, W/m°C
L _e	-	equivalent pipe length, m

The equation for calculating the pipe outlet temperature indicated that for a specific pipe, the parameter K and the equivalent volume of the pipe are constant. This means that the dynamic temperature response at the outlet can be obtained when the historical temperature at the inlet and flow in the pipe are given. It should be noted that the undisturbed ground temperature is not presented in the equation as the influence of this temperature on the outlet temperature is insignificant in normal cases.

When the outlet temperature at time t is known, the heat loss from a pipe was calculated by:

$$Q_{loss} = K_l L_e \left(\frac{T_o(t) - T_i(t)}{2} - T_g \right)$$

 T_g is the undisturbed ground temperature, °C.

The total heat losses from the equivalent network were the sum of the heat loss from the supply and return pipes. It should be noted that the influence between supply and return pipes on the heat loss is not present in the above equation. However, a slightly lower heat loss coefficient than that based on a single pipe was used to compensate for the over-estimation of the heat losses from the two-pipe network.

Pressure Drop and Pumping Power

It was assumed that the pressure drop at design flow is known. It is also known that the pressure drop is proportional to the square of the flow rate. The pressure drop of the network was then calculated by the following equation:

$$\Delta P = (\frac{f}{f_o})^2 \cdot \Delta P_o + \Delta P_{sub}$$

where:

ΔP	-	total pressure drop of the network, Pa
ΔP_o	-	pressure drop of the DH supply and return lines at design flow rate, Pa.
ΔP_{sub}	-	pressure drop at a consumer substation, Pa
f	-	flow rate, m ³ /s
f _o	-	flow rate at design condition, m ³ /s

It was assumed that the pressure drop in DH supply and return line were the same.

The required pumping power then was computed using the following equation:

$$Q_p = \frac{f \cdot \Delta P}{\eta_p}$$

where:

 $\begin{array}{c} Q_p \\ \eta_p \end{array}$

required pumping power, W
overall pump efficiency

B.2 References

Zhao, H. (1995): "Analysis, Modelling and Operation Optimization of District Heating Systems", Ph.D. thesis, Technical University of Denmark, 1995.

B.3 Nomenclature

Cp _w	-	water specific heat capacity, J/kg°C
Cp _s	-	pipe specific heat capacity, J/kg°C
f	-	volume flow rate, m ³ /s
fo	-	flow rate at design condition, m ³ /s
k ₁	-	equivalent heat loss coefficient of pipe, W/m°C
L _e	-	equivalent pipe length, m
Q _p	-	required pumping power, W
Q _{loss}	-	heat loss from the pipe, W
T _g	-	undisturbed ground temperature, °C.
Ti	-	inlet temperature, °C
To	-	outlet temperature, °C
t	-	at time t, sec.

V_w	-	volume of water, m ³
V	-	volume of pipe material, m ³
ΔP	-	total pressure drop of the network, Pa
ΔP_o	-	pressure drop of the DH supply and return lines at design flow rate, Pa.
ΔP_{sub}	-	pressure drop at a consumer substation, Pa
η_{p}	-	overall pump efficiency
ρ_{w}	-	water density, kg/m ³
ρ_s	-	pipe density, kg/m ³
τ	-	time delay, sec.

Part II: Graphical and Mathematical Analysis of Cascaded Substations

Svend Frederiksen JFR Engineering I/S

(on part-time leave from Lund Institute of Technology Department of Heat and Power Engineering)

List of symbols

A, B, C	intermediate variables in solution of 2 nd order, algebraic equations, used in table: Expressions for dimensionless primary return temperature
FC	heat rate transferred in fan-coil
HC	rate of heat loss in re-circulation of domestic hot water
HW	heat rate transferred for heating of domestic hot water
HW1	heat rate transferred in pre-heater of domestic hot water heater
HW2	heat rate transferred in after-heater of domestic hot water heater
LMTD	heat exchanger Logarithmic Mean Temperature Difference
М	mass flow rate ratio (Figure II. 4)
R	heat rate transferred in radiator heat exchanger
c _p	specific heat of water (primary water and domestic hot water)
m _c	re-circulated domestic hot water mass flow rate
m _{hw}	domestic hot water flow rate
r	excess primary return temperature = t_r minus incoming municipal water temperature
Δr	gain in return temperature r when changing connection scheme
r*	(see below)
t _{ind}	induced air temperature in fan-coil heating
t _f	primary forward water temperature (Figure II.19)
t _f	secondary forward water temperature
t _r	primary water return temperature
t _r	(resulting) primary return temperature
t _r	return temperature from radiator (combination with fan-coil, Figure II.19)
t _r	return temperature from fan-coil
ts	primary supply temperature
α	part of re-circulation heat loss HC deriving from re-circulation line (Figure II.4)
δ	difference between primary supply temperature and temperature of municipal water (Figure II.7)
3	(excess) water temperature resulting from mixing of cold water with re-circulated water (Figure II.4)
μ	primary mixing temperature, in excess of incoming cold water temperature in 2-stage connection scheme (Figure II.7)
ρ	radiator return water temperature in excess of incoming municipal water temp. (Figure II.7)
ω	domestic hot water temperature in excess of incoming municipal water temperature
Δω	temperature drop in hot water circulation line (Figure II.4)

r*, μ *, ρ *, ω * dimensionless temperatures created by dividing with δ , e.g. μ * = μ/δ

12 Space Heating and Domestic Hot Water Connection Schemes

12.1 Connection Schemes And Secondary Temperatures

Space heating systems and heaters for the provision of domestic hot water heating (DHW) can be connected in various ways in district heating (DH) substations, Figure II.1. In the first row here, we find very simple connections, while in the second row more complicated systems, incorporating staged heating of DHW, are shown. In all schemes, re-circulation may or may not be present, as indicated by dotted lines. For simplicity, no controls and valves are shown. In all schemes, hot water is provided instantaneously, i.e. in heat exchangers without hot water storage.



Figure II.1: Various district heating substation connection schemes with domestic hot water provision and building radiator space heating

The parallel connection is probably the most widely used scheme, speaking broadly about district heating in various countries. Series connection, with either DHW provision or space heating on the top, is rarely used, although series connection can be found in some high-temperature systems in eastern Europe.

In Sweden, staged DHW provision, with the heater being divided into a pre-heater and an afterheater, has been used for many years, especially in larger buildings. 2-stage has been the most commonly adopted scheme. For a number of years, Swedish type 3-stage was sometimes adopted in geographical regions with soft potable water. Today, it is hardly used any more. Also, to reduce first costs, simple parallel connection is often used where previously 2-stage would have been preferred.

An alternative 3-stage connection scheme, with origins in Russian district heating practice, represents an interesting possibility, as will be shown here below. Both in Swedish and Russian type 3-stage, there is a possibility to by-pass the space heating heat exchanger at high ratios of DHW/space heating load. When this ratio instead is small, primary water demand for space

heating may be bigger than what is required for after-heating of domestic hot water. In Swedish 3stage, this results in overheating of hot water leaving the after-heater, which is compensated for by inmixing of colder, by-passed domestic water. In Russian 3-stage, the after-heater is instead by-passed on the primary side.

Previous studies have confirmed that the use of pre-heating of DHW increases cooling of primary water, although it is often debated if the gain achieved is big enough to justify higher investment costs and slightly increased complication of equipment.

When considering the numerical examples below, we shall assume that the radiator space heating system is operated at relatively low water temperatures. If, for instance, radiators are sized for 60/40°C supply/return, this means that the average water temperature during a season will be in the order of 35°C, which is between the incoming municipal water temperature of typically 5-10°C and supply domestic hot water temperature of 50-60°C. This latter temperature is determined by national or local regulations concerning the minimum hot water temperature required to prevent growth of Legionella bacteria.

12.1.1 Graphical Method

A graphical method of showing temperatures and of constructing primary (DH) return temperature will be employed here. This method is explained and demonstrated by application to a 2-stage connection scheme with finite (not infinite, as used later) sizes of domestic hot water heat exchangers.

The graphical method has been developed at the Lund Institute of Technology, Sweden. In a previous IEA report (Volla, Frederiksen et.al. 1966, cf. list of references at the end of Part II of this report) use of this method was demonstrated.

In diagrams, variations of primary and secondary fluid temperature in heat exchangers and in mixing processes are shown as a function of heat transferred as the abscissa, Cf. Figure II.2. Here, the connection scheme has been so adapted that each of the 3 heat exchangers appears below that part of the graphical construction which shows the particular heat transfer in question. To keep the figure legible, the visualization of the heating-up of radiator water has been omitted, whereas the heating-up of domestic hot water is shown, both for the pre-heater and for the after-heater.

Since specific heat value for water is assumed to be constant, all cooling and warm-up processes in the graphics will be depicted as straight lines. Lines differ in slope: The bigger a certain mass-flow rate is, the smaller the slope in the diagram. In the after-heater, for instance, the mass flow rate is smaller on the primary side than on the secondary side. Therefore, the slope of line A-B is greater than the slope of the warm-up line for domestic water below. For the heat exchanger, no heat losses are assumed; thus the amount of heat given up by the hotter fluid equals the amount of heat received by the colder fluid on the secondary side. Therefore, the abscissa of hotter fluid leaving the heat exchanger at B is equal to the abscissa of the hot water fluid entering the heat exchanger. The ordinate of B is higher due to the finite size of the heat exchanger; if instead a heat exchanger of infinite size had been assumed (as will generally be the case in derivations here below), point B would coalesce with the point of domestic hot water entering the heat exchanger.

Sometimes in heat exchanger literature, changes of fluid temperature in heat exchangers are shown as a function of a length co-ordinate or as a function of a heat transfer surface co-ordinate instead of heat transferred. In diagrams on such a basis, temperature variations appear as exponential, curved lines instead of straight lines.

To simplify notation, (resulting) primary temperature r to the right in the graph – and in further diagrams below other temperatures of fluids – is taken as the temperature in excess of the temperature of incoming municipal water to the domestic hot water pre-heater.

When the primary fluid leaves the after-heater at point B, it mixes with (in this case) colder fluid leaving the radiator heat exchanger at point D to a mixed fluid at point E entering the pre-heater at point F. The common ordinate of points E and F can be constructed by continuing line A-B beyond B to point E where this straight line intersects with cooling line C-D of primary fluid in the radiator heat exchanger. The validity of this graphical construction can be explained as follows:
The resulting, equal temperature of all fluid particles after mixing is independent of details in the mixing process. Let us imagine the process as composed of a heat-exchange process and then a mixing process. First, all hotter-fluid particles cool down to the mixing temperature, accompanied by heating-up of all cooler particles to the same, final temperature. The second step will be a mixing of all particles of equal temperature. In the heat-exchange process, the hotter fluid will cool along a line of the same downward slope as in the previous cooling process of the after-heater. The cooler fluid will warm-up backwards along line C-D, not up to point C, but to point E where heating ends.

The cooling of the primary water along line F-G in the graphical construction follows a slope determined as the continuation of line A-F. This line has the slope that the mixed fluid from E would have had, had it not resulted from mixing of the two primary fluids giving up their respective amounts of heat in the hot-water after-heater and radiator separately, but instead from a mixed total fluid of the total flow rate giving up the same total amount of heat.



Figure II.2: Graphical method of constructing primary return temperature r, illustrated by application to 2-stage connection scheme with finite size of heat exchangers

12.1.2 Comparison of Parallel and Series Connection

Figure 3 shows graphical constructions of primary return temperature, r, for parallel connection and the two types of series connection, in a comparison with identical secondary temperatures for the 3 cases. There is no re-circulation of DHW. As in Figure II.2, all temperatures are shown as values in excess of incoming municipal water temperature, and heating of radiator water is not shown.

It can be seen that, as one would expect, parallel connection produces a lower return temperature, r, than both types of series connection. If, as is not the case here, DHW re-circulation had been arranged for, it would be possible to find situations where series connection with DHW on the top can produce lower primary return temperature than in series connection.

12.2 DHW Re-circulation in Simple Substations

Re-circulation of DHW is commonly adopted in medium-sized and larger buildings to ensure that water leaving a hot water faucet will actually become hot only shortly after its opening, even when the faucet prior to opening has been kept closed for a long time, and even if the general hot water consumption in the building is small. Primarily, it is a matter of convenience. However, there is also an energy conservation aspect, in that slow temperature rise leads to bigger hot water consumption: If you know by experience that it will take minutes to get hot water, you may acquire the habit of opening the valve and wait for a quarter of an hour or so before returning to the open faucet. Another aspect is that re-circulation tends to counteract Legionella growth in branches from which DHW is not being tapped.

Single-family buildings seldom incorporate DHW re-circulation, because DHW supply pipes in such buildings normally are short and not very branched, which makes re-circulation less necessary.

The magnitude of heat losses from DHW pipes depends much on the length of pipes, and is thereby related to building size and to the degree of pipe insulation. The magnitude of useful DHW heat rate is here termed HW, while the size of aggregate DHW distribution heat losses from both supply and re-circulation lines is termed HC. Typically, for a large building, the ratio between HC and HE may be 0.5.

When DHW heating takes place in a single heat exchanger only, and re-circulated water is being mixed with incoming cold water, there will be an increase in the temperature of the water entering the exchanger of magnitude " ϵ ", Figure II.4. The return temperature on the primary side of the heat exchanger will increase by exactly, or approximately the same amount, depending on the size of the heat exchanger (exactly the same amount in the theoretical case of infinitely large heat exchanger).



Figure II.3: Graphical derivation of primary return temperature, r, for parallel connection and for series connection with either domestic hot water or radiator circuit on top

The warming up of cold water to DHW temperature ω when there are no heat losses in the DHW system, and when the heat exchanger is infinitely large, is shown in Figure II.4 by dotted lines, for comparison.



Figure II.4: Graphical derivation of return temperature when a finite mass flow of re-circulated hot water is mixed with incoming municipal water

Analytical derivation:

$$HW = m_{hw}c_{p}\omega \qquad \alpha HC = m_{c}c_{p}\Delta\omega \qquad M = \frac{m_{c}}{m_{hw}} = \alpha \frac{HC/HW}{\Delta\omega/\omega}$$
$$m_{c}(\omega - \Delta\omega) = (m_{hw} + m_{c})\varepsilon \quad gives:$$
$$\frac{\varepsilon}{\omega} = \left(1 - \frac{\Delta\omega}{\omega}\right)\frac{M}{1 + M}$$
$$Numerical \ example: \ M = 0.5 \frac{0.5}{5C/50C} = 2.5 \ gives:$$
$$\frac{\varepsilon}{\omega} = \left(1 - \frac{5C}{50C}\right)\frac{2.5}{1 + 2.5} = 0.64$$

When heat losses occur instead, outgoing hot water must be heated to an excess supply temperature, so that the expected DHW temperature is delivered to the faucets. Obviously, in a branched system at a given moment, there will normally be a temperature variation among branches due to variations in flow rate. The temperature graph indicated in Figure II.4 accordingly can be interpreted as a mean graph for the entire system. The re-circulated water will arrive at the heat exchanger at a temperature being lower than ω by the amount of $\Delta \omega$. Below the figure, a formula for the relative temperature rise, ϵ/ω , is derived and an example of a calculation of this quantity is given.

The temperature drop in the DHW supply line will in general differ from $\Delta \omega$. Also, the heat loss from the supply line, in general, will differ from the heat loss from the re-circulation line, due to different pipe diameters etc. In the numerical example, α , defined as the part of HC lost from the re-circulation line, has been set to 0.5.



Figure II.5: Single stage heating of domestic hot water, increased primary return temperature, r, when hot water recirculation is introduced, first as infinite mass flow rate and then as a finite mass flow rate



Figure II. 6: With 2-stage heating of domestic hot water, detrimental influence of hot water re-circulation on primary return temperature, r, can be eliminated (provided re-circulation load HC is not too big)

In particular, when the radiator system is of a low temperature type, DHW circulation heat losses are rather harmful from a thermo dynamical point of view in that heat transfer to building interiors takes place at a rather high temperature level, whereby such losses tend to reduce primary water cooling substantially. This can be seen from Figure II.5 showing examples of temperature graphs for simple parallel connection, without and with, DHW circulation. Influence of circulation on primary return temperature is shown for two cases; (middle graph) a theoretical limit case with infinite circulation mass flow rate, m_c, and below a case with a finite circulation mass flow and a temperature drop, both in the DHW supply line and in the circulation line. In both cases, incoming cold water mixes with return circulation water of a higher temperature, which also causes an increase in (excess) return temperature, r, to the network, when the higher return from the DHW heat exchanger mixes with return water form the space heating circuit.

With infinite m_c there is a dramatic rise in r. When m_c is finite, the temperature rise is more moderate, but still substantial. What circulation flow is appropriate cannot be settled immediately. In any case, from the point of view of Legionella prevention, one cannot accept too big a temperature drop in the circulation line because of the associated risk of microbial multiplication in the circulation return line. Figure II.6 shows that dividing the DHW heater into two stages and arranging the circulation return to enter between the two stages can reduce the detrimental influence of DHW circulation on r substantially. With finite heat exchanger sizes, the very low r to the bottom right of course, cannot be achieved. The average temperature difference between primary and secondary sides of the pre-heater is very small in the theoretical case of infinite heat exchanger shown in the figure. Still, it can be assumed that also with finite heat exchangers, the staged solution will produce lower primary return temperature.

12.3 3-Stage Substation for Minimal Influence of DHW Circulation on Primary Return

The simple solution to divide DHW heating into two stages, shown to the right in Figure II.6, can only reduce DHW circulation influence on r as long as there is some DHW consumption. When this is not the case, the substation will perform as the simple parallel connection scheme (or even worse in practice, if the after-heater is small).

Figure II.7 shows graphical constructions of return (excess) temperature r for both 2-stage and Russian type 3-stage connection in two cases: first with no DHW re-circulation load, and below when there is such a load, HC. As in previous figures, heat exchanger sizes are assumed to be infinite, as a theoretical limit case and for simplicity. The graphical constructions here are a little more complicated than with simple parallel connection of heat exchangers but hopefully not too difficult to understand. As can be seen, the aggregate useful hot water load HW, due to the preheating stage in both schemes, falls in two parts, HW1 and HW2, referring to the pre-heater and the after-heater, respectively.



Figure II.7: Comparison of 2-stage and Russian type 3-stage connection with and without hot water re-circulation of infinite mass flow rate

HW1

R

HW1

HC HW2

In order that the figures should not become too crammed with information, no heating curves for radiator water have been included in the graphical constructions. It should be noted, however, that the various constructions pre-suppose that the slope of heating lines for radiator water are not bigger than the slope of the cooling line for primary water passing the radiator heater. This in turn means that the constructions are made under the assumption that the flow rate in the radiator circuit is not too small. In fact, since a smaller radiator water flow rate can be used to lower the primary return water temperature leaving the radiator heat exchanger, cascading and choice of low radiator water flow rate to some extent can be seen as alternative ways of achieving high cooling of primary water.

3-stage solutions have a potential for minimizing the temperature influence of DHW circulation, as Figure II.8 shows. With no DHW circulation (HC = 0), 2- and 3-stage schemes for the load case shown are seen to be equivalent in terms of r. But with a DHW circulation load added, the

HC HW2

R

accompanying rise in r turns out to be smaller with 3-stage than with 2-stage. This can be explained by the observation that in the 3-stage case, primary water having been cooled in the after-heater - at a rather high temperature level and therefore itself being returned from the after-heater at a rather high temperature - is passed on to the radiator heat exchanger operating at a medium temperature level instead of directly to the predominantly low-temperature pre-heater. Since return water from the after-heater mixes with hot water being fed to the radiator heat exchanger directly from the supply line of the primary side, the flow arrangement has the character of mixed series/parallel, which is thermodynamically better than the pure parallel connection of after-heater and radiator heater of the 2-stage scheme.

The thermodynamic advantage of the 3-stage scheme, in terms of lower r, is biggest when there is no DHW consumption, as Figure II.8 shows. Here, the top left figure is representative of, in addition to, 2-stage: simple parallel connection and parallel connection with staged DHW heating, which schemes are all thermodynamically identical in such a case, giving the same r.

The bottom diagrams of Figure II.8 show that at a certain DHW load (in the example of the figure when HW/R = 1.25) r vanishes both for 2- and 3-stage, which thereby become equivalent. This case, though, represents a very idealized situation; in both connection schemes, the primary circuit water cooling line in the pre-heater coincides with the heating line of potable water on the secondary side, i.e. the infinitely large heat exchanger must have the theoretical capacity of transferring heat with no temperature gradient across the heat transfer surface.

Summing up, our idealized, graphical constructions of primary return r for various connection schemes verify an intuitive presumption that a 3-stage connection scheme can be advantageous in terms of r being as low as possible.

A control scheme for a Russian type 3-stage connection scheme may be developed according to Figure II.9. The control logic is more complicated than a typical control system of a 2-stage connection. Therefore, it will probably be best suited for bigger installations where sophisticated controls may be economically viable.

There are 4 modes of operation. In mode IV, valve Va3 is closed and the substation functions as a 2-stage connection scheme. This mode is sensible when the secondary forward temperature of the space heating loop is rather high, which will typically occur at low outside air temperature. It is clear that passing water from the DHW after-heater to the space heating system through valve Va3 only is sensible when the temperature level of the radiator circuit is so low that it can cool the water coming from the after-heater.



Figure II. 8: Comparison of 2-stage and Russian type 3-stage, given that there is a re-circulation load HC of infinite mass flow rate, for two limit cases:

Top (2 cases): No draw-off of domestic hot water (for instance at night)

Below (2 cases): Domestic hot water load being precisely so big that (excess) primary return temperature, r, vanishes (very theoretical cases)





Figure II. 9: Possible control arrangement and switch-logic for Russian-type 3-stage connection scheme. The bottom diagram roughly describes the switch-logic including 4 modes

In the other three operational modes, I, II, and III, Va3 is open, and either Va1 or Va2 or both is/are closed, please cf. the bottom part of Figure II.9 where the modes are defined, and rough indications are made in a map-diagram at which combinations of hot water and space heating demands the various modes are valid.

In mode I, Va1 is open, Va2 is closed, and Va3 is open. Va1 controls domestic hot water temperature and Va3 controls the forward temperature of the space heating system.

In mode II, Va1 and Va2 are closed, while Va3 is open. This is a transitional mode where there is such balance between secondary side demands and temperatures that there is no need for Va1 or Va2 to be open.

In mode III, Va1 is closed, while Va2 and Va3 are both open. Va2 controls space heating forward temperature and Va3 controls domestic hot water temperature.

It is envisaged that traditional thermostatic feedback controls are being supplemented by a switchlogic, built into a small process computer ('SwC' in top control box of the figure). For instance, in a control situation where mode I prevails, it can be observed that if the forward temperature of the hot water exceeds its set-point value by more than a certain amount for more than a certain period of time, the operation is switched to mode III, i.e. Val closes, Va2 opens, and the control of Va3 switches from this valve being controlled by the space heating forward temperature to being controlled by domestic hot water temperature instead.

The borderline between modes III and IV can be determined from calculations establishing under which load conditions 2- or 3-stage operation produces the lowest primary return temperature. A simple strategy could be to set a certain outside air temperature, perhaps defined as a 24-hour temperature value, as a condition, so that at outside, air temperatures below this limit, 2-stage operation is prescribed.

The control system shown in Figure II.9 could be extended to include an additional feedforward control function for the domestic hot water temperature. Hereby, the settings of primary valves influencing domestic hot water temperature could be influenced directly by the size of the domestic hot water demand. This principle has been used with success in smaller substations built on conventional connection schemes.

The Swedish type of 3-stage connection incorporates a mixing valve in the outgoing line of the domestic hot water, which provides a direct and fast hot water control. From a pure control dynamic point of view, a Swedish 3-stage may appear more appealing than a Russian-type 3-stage. But a Swedish 3-stage cannot be used with hard potable water, since overheating of hot water in the after-heater precipitates scale fall-out. This drawback is not present with Russian type 3-stage, where the amount of primary water passing the after-heater is always being apportioned according to hot water demand.

12.4 Analytical Expressions for Return Temperature of Various Connection Schemes

By rather simple algebraic equations, it is possible to derive explicit analytical expressions for the return temperature of parallel 2-stage and 3-stage connections under simplified conditions.

Expressions for dimensionless primary return temperature $r^* = r/\delta =$

HC = 0, m_c=0: HC
$$\ge$$
 0, m_c = ∞ :

<u>Parallel (single stage):</u> $\rho * \frac{1}{1 + (1 - \rho^*) \frac{HW}{R}}$ $\rho * \frac{1 + \frac{1 - \rho^*}{1 - \omega^*} \frac{\omega^*}{\rho^*} \frac{HW + HC}{R}}{1 + \frac{1 - \rho^*}{1 - \omega^*} \frac{HW + HC}{R}}$

$$\underline{2\text{-stage:}} \qquad \rho * \frac{1 + (1 - \frac{1}{\omega^*})\frac{HW}{R}}{1 + (1 - \frac{\rho^*}{\omega^*})\frac{HW}{R}} \qquad \mu * - (1 - \mu^*)\frac{\frac{\mu^*}{\omega^*}}{\frac{R + HC}{HW} + 1 - \frac{\mu^*}{\omega^*}}$$

where:
$$\mu^* = \omega^* \frac{-B - \sqrt{B^2 - 4AC}}{2A}$$

 $A = \frac{1 - \rho^*}{1 - \omega^*} \frac{HW}{R}, B = -\frac{1 - \rho^*}{1 - \omega^*} \frac{2HW + HC}{R} - 1$

$$C = \frac{1 - \rho^*}{1 - \omega^*} \frac{HW + HC}{R} + \frac{\rho^*}{\omega^*}$$

Russian

$$\frac{n \ 3-stage:}{R} \qquad \text{If:} \quad \frac{HC}{R} + \frac{HW}{R} \frac{\omega^* - \rho^*}{\omega^*} \le \frac{1 - \omega^*}{\omega^* - \rho^*}:$$

$$\rho^* \frac{1 + (1 - \frac{1}{\omega^*})\frac{HW}{R}}{1 + (1 - \frac{\rho^*}{\omega^*})\frac{HW}{R}} \qquad \text{then:} \quad \rho^* \frac{1 + (1 - \frac{1}{\omega^*})\frac{HW}{R + HC}}{1 + (1 - \frac{\rho^*}{\omega^*})\frac{HW}{R + HC}}$$

$$\text{If:} \quad \frac{HC}{R} + \frac{HW}{R} \frac{\omega^* - \rho^*}{\omega^*} \ge \frac{1 - \omega^*}{\omega^* - \rho^*}:$$

$$\text{then:} \quad \beta^* \left(1 - \left(1 - \frac{\beta^*}{\omega^*}\right)\frac{HW}{R}\right)$$

$$\text{where:} \quad \beta^* = \omega^* \frac{-B - \sqrt{B^2 - 4AC}}{2A}$$

$$A = \frac{HW}{R} \quad B = -\frac{HC}{R} - 2\frac{HW}{R}$$

$$C = \frac{HC}{R} + \frac{HW}{R} + 1 - \frac{1}{\omega^*}$$

Significance of symbols are given by Figure II.7 and as follows:

$\omega^* = \omega/\delta =$	dimensionless domestic hot water temperature
$\rho^* = \rho/\delta =$	dimensionless return from space heating system
$\mu^* = \mu/\delta =$	dimensionless hot water temperature between pre- and after-heater
HW =	domestic hot water load
HC =	hot water circulation heat load
R =	radiator space heating load

Idealizations made (as in previous, graphical analysis):

- Infinite heat transfer areas (or heat transfer capacity) of heat exchangers, yielding 0 least • temperature difference in heat exchangers
- When domestic hot water circulation is used, then circulation mass flow rate = ∞ ٠

The expressions are valid when yielding $r^* \ge 0$. When an expression yields a negative value, $r^* =$ 0 will apply instead.

It can be seen that:

- Putting HC = 0 in the right-hand equations yields the left-hand equation only with 3-٠ stage-connection, and when the first inequality condition is fulfilled. With parallel- and 2-stage connection, circulation raises the return temperature even if HC = 0
- When there is no hot water re-circulation, expressions for 2- and 3-stage are identical. •

12.5 Simple DHW Storage With Hot Water Re-Circulation

Storage tanks are sometimes used in DH substations instead of instantaneous water heaters. Such tanks come in many types, some of them incorporating internal heat transfer, e.g. across an internal heating coil. Here, we shall only consider substations with storage tanks relying on external heat transfer. This makes performance easier to predict, and it can be argued that properly performing systems with external heat transfer are thermodynamically better than systems where heat transfer is mixed with heat storage. It must be admitted that systems with external heat transfer tend to be a little more costly.



Figure II.10: Dimensionless return temperatures, $r^* = r/\delta$, as a function of hot water load calculated by formulae given above, for 2-stage and Russian 3-stage, HC/R = 0.25, $\omega^* = \omega/\delta = 0.5$, $\rho^* = \rho/\delta = 0.25$



Figure II.11: Parallel connection substation with hot water storage tank with external heat exchanger, hot water recirculation, and diverter valve Va5

Figure II.11 presents such a storage tank system. DHW is here being produced by heating in an external heat exchanger, HE1. In a charge cycle, the charge pump draws out cold water from the tank bottom and feeds the top of the tank with hot water. Inside the tank, at intermediate charging situations between full charging and full discharging, a thermocline, i.e. a thermal transitional layer, separates top hot water from bottom cold water. At charging, the thermocline, which in the figure has been indicated by a temperature graph inside the tank, moves downward with some rather low speed v, causing minimal mixing between hot and cold water.

The charge pump is thermostatically controlled to stop when the tank is filled by hot water, i.e. when the thermocline reaches tank bottom. The detailed timing in this control is essential. From a purely thermodynamic point of view, it can be seen as desirable that charging stops before any luke-warm water of the thermocline has been drawn out, since such water of higher temperature than that of cold water would cause raised return temperature on the primary side of HE1. However, from the point of view of *Legionella* prevention, it is dubious to have the same luke-warm water staying inside the tank infinitely. Therefore, it can be argued that on some occasions the charge pump should continue its function until thermocline water has been drawn out from the tank bottom. If the thermostatic control of valve Va1 is programmed to close down at the same time, i.e. for a short while towards the end of a delayed charging phase, the harmful effect on primary return temperature will be minimal.

For example, in Figure II.11, a rather high setting of the thermostatic control of DHW leaving heat exchanger HE1, at 75°C, has been indicated. This will enhance temperature disinfecting, i.e. killing of microbes like *Legionella* inside the tank. When such a strategy is chosen, it is necessary to reduce the outgoing DHW temperature to avoid scalding accidents. In the system shown, this is done by a three-way mixing valve, Va4, fitted with a thermostatic control set at 60°C.

The hot water re-circulation line is also fitted with a three-way diverter valve Va5. As long as recirculated water keeps a temperature above 50°C, such water is being directed into the top of the tank, where it mixes with somewhat hotter water and thereby gradually reduces the overall temperature of hot water above the thermocline. If the temperature of the re-circulated water falls below 50°C, as may happen when, for example, hot water consumption has been low for a long time, re-circulated water is directed to mix with incoming cold water instead of being fed into the tank. This is thermodynamically less favourable because of raised primary return temperature. However, in this way, it is guaranteed that heat is fed into the DHW system to compensate for heat losses which could otherwise cause very low DHW temperatures, being unacceptable from the point of view of *Legionella* prevention.



Figure II.12: Typical network supply and return temperatures vs. outside air temperature, possibility to eliminate return temperature rise in warm season when applying a raised supply temperature to a substation with hot water storage tank according to Figure II.10.

Figure II.12 shows typical daily mean DH network supply and return temperatures, as measured at a heat production plant, versus outside air temperature for a network which is trimmed to minimize transfer of hot supply water directly into the return line and which is fitted with thermodynamically efficient substations maximizing primary water cooling. For such a DH network, a raised supply temperature at a given heat load would cause a somewhat lower return temperature. In a system with many by-passes and/or leaking substation control valves, a raised supply temperature at a given heat load would instead cause raised return temperature as well.

Even for a well-trimmed DH system of the kind described, there will typically be a tendency for t_r to increase with falling heat load in summer, as indicated by the solid t_r curve. This tendency can be attributed to a thermodynamically harmful influence of re-circulated DHW inside buildings. In buildings with DHW re-circulation and once-through DHW heating, at mid-night (when there is minimal DHW consumption) in summer (when there is no space heating load) the substation primary return temperature will be determined by the temperature of the re-circulated DHW. In the daytime, during the summer period, due to the influence of incoming cold water, t_r will be lower. The average t_r for a 24-hour period in summer, though, will be higher when there is no space heating load than with a moderate space heating load, due to the favourable influence of a rather low return temperature from the radiator system in the latter case.

By contrast, in the case of a substation with storage tank, according to Figure II.11, there is a possibility that t_r can be regarded as virtually unaffected by DHW re-circulation, even in the periodical absence of DHW consumption and space heating load. Such a thermodynamically favourable situation may exist if the temperature of re-circulated water is high enough, so that valve Va5 continually directs re-circulated water into the tank, and no mixing with incoming cold water takes place. Requisites for this are:

- Primary supply temperature t_s should be sufficiently high to allow for a not too low DHW charge temperature.
- Heat exchanger HE1 should be designed big enough to avoid a too big least temperature difference between primary return temperature and incoming cold water temperature, related to the actual difference between primary and secondary temperatures at the hot end of the heat exchanger.
- Periods with no or small DHW consumption should not be too long, depending on size of circulation heat losses.
- Heat losses through tank walls should not be too big.

Dotted lines for t_s and t_r in Figure II.12 illustrate that properly designed hot water storage systems can result in low primary return temperatures, even in mid-summer, and when DHW circulation is present in substations.



Figure II.13: Superheating of domestic hot water need not affect primary return temperature of a substation with a storage tank according to Figure II.10

Figure II.13 illustrates how DHW superheating in a tank system according to Figure II.10 can function with no influence on primary return temperature.

12.6 DHW Storage Combined With Cascading



Figure II.14: Substation connection scheme incorporating a hot water storage tank into Russian type 3-stage connection

Figure II.14 displays another strategy to avoid DHW re-circulation causing unnecessary increases in primary return temperature. Here, a Russian type 3-stage heat exchanger connection scheme has been combined with a DHW storage tank, thereby being charged by two-stage hot water heating. This system is rather robust in terms of how low t_s can be allowed to be, how big DHW circulation heat losses can be accepted, etc. Compared to the equivalent once-through scheme of

Figure II.9, domestic hot water dynamics is easier to handle, since the tank evens out load variations from the DHW system.

Both systems shown in Figures II.11 and II.14 appear somewhat complicated. It is, however, difficult to envisage simpler systems incorporating hot water storage, designed for bigger buildings with DHW re-circulation, of equal thermodynamical performance and equal safety against *Legionella* growth.

Compared to substations with once-through DHW heating in two stages, corresponding systems with storage tanks, as the one depicted in Figure II.14, have a potential for lower average return temperature t_r of the system, especially in the case of smaller buildings with large relative variations in DHW load. The reason for this can be found in the non-linear relationship between t_r and DHW load, cf. Figure II.15. At low DHW loading of a substation with DHW pre-heating, r of the substation will go down almost linearly with load. At some point, however, the curve will bend, viz. when a point is approached where there is no more heat to be taken out from the return water leaving the radiator heat exchanger. Eventually the curve will flatten and thereafter even rise again.



Figure II.15: Typical return temperature vs. hot water load for a connection scheme incorporating a pre-heater stage

As long as DHW is small enough for r to vary with the almost linear range, the average r will be rather unaffected. However, if there are longer periods with higher DHW loads, opportunities for utilizing pre-heating in terms of average r will be lost.

This latter point of combining DHW storage with cascading applies rather generally, i.e. it is not specific for systems with DHW re-circulation or for Russian type 3-stage. For instance, the favourable effect on mean primary return temperature will also be found for a system combining DHW storage with a 2-stage connection scheme.

13 Cascading of Radiator and Fan-Coil Space Heating

This section analyzes the magnitude of the return temperature decrease obtained by cascading radiator and fan-coil space heating, compared to simple parallel connection of the two types of space heating. Starting out with a comparison between the two types of connection in fixed operational modes, the analysis continues by exploring two ways of minimizing return temperature of the cascaded connection by optimizing the operational mode. All results are derived by use of graphical constructions.

For simplicity, only the space heating system as such is being analyzed here, i.e. effects of combination of space heating with domestic hot water provision are not being considered.

13.1 Return Temperature vs. Size of Fan-Coil - Radiator Heat Load Size

Figure II.16 shows results of graphical constructions of secondary return temperature t_r ' of a space heating heat exchanger of a district heating substation incorporating both radiator and fan-coil space heating. Parallel and series (cascaded) connections are compared. Five cases of different parameter values have been considered, with ratio of fan-coil (FC) to radiator (R) heat load, FC/R being varied in each case at constant FC-LMTD (Logarithmic Mean Temperature Difference). When designing a system for a building, the size of this ratio can be selected. Figures II.17 and II.18 show, as examples, graphical constructions for two of the five cases of Figure II.16.

A heat recovery unit is assumed to preheat air to a certain air temperature before entering the fancoil, heating the air further to the air induction temperature, which is kept fixed irrespective of the FC/R ratio.

Both in the parallel and series (=cascaded) connection schemes, a common water loop passes water heated in a common water-water heat exchanger on to fan-coils and radiators. In the series connection case, there are two by-pass pipes, each fitted with a controlled by-pass valve: When water flow leaving the radiator is too cold and too small to supply the amount of heat required by the fan-coil to heat the air to the desired air induction temperature, BP1 opens to supplement heat supply to the fan-coil. When instead the water flow leaving the radiator is so hot and too big that it would cause over-heating of the inducted air, BP2 opens instead.

Later we shall examine the consequences of allowing the air induction temperature to float instead of being kept fixed.

The two top diagrams of Figure II.16 represent the theoretical limit case of FC-LMTD = 0° C. Here, the gain achieved by cascading increases with increasing FC/R ratio up to a value of around 1, from which the gain slowly decreases with increasing ratio.

With a more realistic size of FC heat exchanger, defined by FC-LMTD = 25° C, for the $90/50^{\circ}$ C radiator case, cascading still produces a significant gain in return temperature. But in the $70/40^{\circ}$ C radiator case, the gain becomes insignificant (and even slightly negative at small FC/R). With a smaller FC heat exchanger, defined by FC-LMTD = 35° C. There is no point at all in cascading with the low-temperature $70/40^{\circ}$ C radiator system.



Figure II.16: Secondary return temperature, t_r', for series and parallel connections, for a number of design cases, based on graphical constructions exemplified by Figures II. 17 and II.18



Figure II.17: Graphical construction of secondary return temperature, t_r , with a radiator designed for temperatures 70/40°C, and a fan-coil heat exchanger designed for LMTD = 0°C (Theoretical Limit Case)

Series:

Parallel:



Figure II.18: Graphical construction of secondary return temperature, t_r , with a radiator system designed for temperatures 70/40°C, and a fan-coil heat exchanger designed for LMTD = 25°C (Constant)

13.2 Optimization Of Space Heating System Forward Temperature

As already shown in a previous IEA-study, the secondary forward temperature of a space heating system with cascaded fan-coil and radiator system can be optimized to minimize the primary return temperature. To gain a better understanding of this optimization, an example is here singled out for analysis by graphical construction, cf. Figure II.19.

The connection scheme considered is of the same cascaded type as was analyzed above. Ideal thermostatic valves are assumed to control radiator flow rate to keep a constant indoor temperature. The pressure differential across the space heating system is kept constant by speed control of the circulation pump. A feed-back from a temperature sensor in the space heating forward line keeps a desired forward temperature t_f , for instance, according to given control curve

specifying t_f ' as a function of outside air temperature, t_a . By-pass valves BP1 and BP2 are controlled by air induction temperature t_{ind} .

An operational mode variation is now being performed by which the secondary forward temperature t_f ' is varied at constant outside air temperature, constant primary forward temperature t_f , as well as constant heat load components FC and R for the fan-coil and the radiator, respectively. Since inducted air temperature and indoor air temperatures are kept constant, this means that logarithmic mean temperature differences FC-LMTD and R-LMTD also are constant, as are the heat transfer coefficients, the air flow rates across the fan-coil, which is assumed to be kept unchanged during the mode variation.

We shall follow changes in four return temperatures: The primary side return temperature t_r , the common, secondary side return temperature t_r , the return temperature t_r , from the radiator, and the return temperature t_r , from the fan-coil, which coincides with t_r as long as BP2 is closed.

In Figure II.19, the diagram furthest to the left shows how these temperatures vary with the secondary forward temperature. For the primary return temperature t_r two curves are shown, referring to two examples of sizes of the water-water heat exchanger of the space heating system.

The next graph shows temperature lines for a relatively high t_{f} ', resulting in a relatively low t_{r} ''; BP1 is open. When in the next graph t_{f} ' is lowered, t_{r} '' goes up, and so does t_{r} '; BP1 is still open. When t_{f} ' in the next graph is lowered even further, a situation is created, where such a balance of heat demands, flows and temperatures exists that neither BP1 nor BP2 is open; both t_{r} '' and t_{r} ' are higher than in the previous graph. The last graph shows the theoretical limit case of infinite flow rate in the space heating system, with BP2 open, so that t_{r} '' no longer is identical with t_{r} '; instead t_{r} '' and t_{r} ' become identical, because of the infinite flow rate.

It can be seen that t_r' increases all the time when t_f' is lowered. In the theoretical case of an infinitely large water-water heat exchanger of the space heating system, the lowest primary return temperature t_r will result when t_f' is set equal to t_f . When the heat exchanger size is finite, a certain, lower t_f' produces a minimal t_r . The bigger the heat exchanger is, the higher t_f' should be to minimize t_r .



Figure II.19: Optimization of secondary forward temperature, t_j , to minimize primary return temperature, t_n , when radiator and fan-coil space heating systems are connected in series (Cascaded)

13.3 Varying Inducted Air Temperature

The inducted air temperature should neither be too far below, nor too far above, the desired indoor air temperature, to avoid draft and other unpleasant thermal exposure to humans inside the buildings provided with air heating. However, what is ideal or acceptable in terms in inducted air temperature seems to vary substantially according to differing national practices. For instance, German practice (at least sometimes) allows for much higher inducted air temperature than what is practiced in many other countries.



Figure II.20: Variation of FC/R ratio and varying inducted air temperature, t_{ind}

Thus, it seems reasonable to allow for at least some deviation in targeted air induction temperature, in particular if there is a thermo dynamical advantage in terms of lower DH return temperature. This is illustrated by Figure II.20. Here, a constant total heat supply is assumed, while the ratio FC/R is allowed to vary, accompanied by variations in inducted air temperature t_{ind} .

The forward temperature of the space heating system, t_r , is kept constant. Varying R is performed by letting the radiator return t_r vary, as indicated in the top left diagram.

The two diagrams to the right in the top row of diagrams refer to the theoretical case of FC-LMTD = 0° C, while in the bottom row of diagrams FC-LMTD = 25° C. In the second case, FC-LMTD varies when FC/R is changed, UA assumed to be constant for the air heat exchanger.

It can be seen that in the first case, increasing FC at the expense of R, at the same time allowing t_{ind} to increase from 25 to 35°C, lowers the return temperature t_r ' right down to the temperature of incoming air (from the pre-heater).

With FC-LMTD = 25°C, it can be seen that (at least for the numerical examples shown) the lowest t_r ' is achieved when neither BP1 nor BP2 is open: When in the left diagram FC is decreased and t_{ind} is lowered, BP2 has to open. In the diagram to the right FC is increased and t_{ind} goes up; BP1 opens. In both cases, t_r ' increases.

From a thermodynamical point of view, it seems natural to seek operational modes associated with minimal mixing of flows in valves, so that the result derived from the lowest row of diagrams may not seem too surprising. It has not, however, been examined to which extent modes with no by-pass flows can be considered optimal in general.

A general optimization of operational modes should include simultaneous variation of secondary forward temperature t_{f} ' and inducted air temperature t_{ind} .