BUILDINGS AND DISTRICT HEATING

- contributions to development and assessments of efficient technology

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To my family

Karin, Ylva and Alfons

ABSTRACT

The objective of this thesis is to contribute to improved performance of district heating systems by focusing on three items.

Item 1: Study of three methods for improving the performance of sub-systems within buildings connected to the distribution network. The methods can be considered as more or less innovative. Results from two of these measures are quantified in terms of so-called Primary Energy Factors.

Item 2: Investigation of the extent to which space heating supply to buildings connected to a district heating network can be maintained in the event of a failure in the electric power supply.

Item 3: An experimental investigation of the cavitation phenomena within primary side substation control valves.

More specifically, with respect to item 1, three methods were investigated;

a) Deriving substation connection schemes that will improve overall thermodynamic efficiency, taking into account a complete range of load cases, including variations in hot water load,
b) Adding small fan blowers onto the surfaces of existing radiators, with the aim of increasing convective heat transfer, thereby making it possible to lower the operating temperatures of the hydronic space heating system such that supply and return temperatures on the primary side of substations can be lowered,
c) Advanced control of hydronic space heating systems such that for all heat loads an operational mode is automatically selected that optimises the performance of a substation in terms of the lowest possible return temperature on the primary side.

The thesis shows how Primary Energy Factors for district heating can be affected when the heat is produced in a combined heat and power station and

in a heat-only boiler. Calculations of the factors were performed according to the EU standard, EN 15603:2008. The results of the analysis show that the size of the factor depends strongly upon the particular production mix of the electric power system to which the combined heat and power plant is assumed to be connected.

Concerning item 2, district heating in case of power failure, both numerical analysis and field experiments showed that a significant share of the heat supply to buildings can be maintained even if a circulation pump is no longer running. This is due to the fact that gravity driven circulation takes over to variable extent. Generally, 20-80% of the initial heat load can be supplied, provided that circulation is maintained in the district heating network. Further improvements are possible if some minor modifications are made.

Item 3 pertains to a small laboratory study of cavitation in modern state-of-the art control valves. It was found that cavitation primarily occurred under conditions that in practice only seldom occur, such as a large differential pressure across the valve combined with a situation when the valves are close to being fully opened.

Keywords: District heating, hydronic space heating, natural circulation, lowered return temperature, primary energy, add-on-fan blower

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And finally,
Thanks to my family!

List of publications

This thesis is based on the following papers, referred to in the text by their Roman numerals. The papers are appended at the end of the thesis.

Paper I **Optimized space heating system operation with the aim of lowering the primary return temperature**
Ljunggren, P., Johansson, P.-O., Wollerstrand, J.
Conference proceeding from the 11th International Symposium on District Heating and Cooling, 2008, Reykjavik, Iceland.

Paper II **Obstacles for natural circulation in heating systems, connected to district heating via heat exchangers, during a power failure**
Johansson, P.-O., Ljunggren, P., Wollerstrand, J.
Conference proceeding from the 11th International Symposium on District Heating and Cooling, 2008, Reykjavik, Iceland.

Paper III **Modelling Space Heating Systems Connected to District Heating in Case of Electric Power Failure**
Johansson, P.-O., Ljunggren, P., Wollerstrand, J.
Conference proceeding from the 48th Scandinavian Conference on Simulation and Modeling (SIMS 2007), 2007, Göteborg (Särö), Sweden.

Paper IV **District heating in case of power failure**
Lauenburg, P., Johansson, P.-O., Wollerstrand, J.
Applied Energy 87, p. 1176-1186, 2010

Paper V **Improved cooling of district heating water in substations by using alternative connection schemes**
Johansson, P.-O., Lauenburg, P., Wollerstrand, J.
Conference proceedings from ECOS 2009 (22nd International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems), Foz do Iguaçu, Brazil.

Paper VI **Influence of district heating temperature level on a CHP station**
Johansson, P.-O., Jonshagen, K., Genrup, M.
Conference proceedings from ECOS 2009 (22nd International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems), Foz do Iguaçu, Brazil.

Paper VII **Improved temperature performance of radiator heating system connected to district heating by using add-on-fan blowers**
Johansson, P.-O., Janusz Wollerstrand, J.
Conference proceeding from the 12th International Symposium on District Heating and Cooling, 2010, Tallinn, Estonia.

Paper VIII **Heat Output from Space Heating Radiator with Add-on-fan Blowers**
Johansson, P.-O., Janusz Wollerstrand, J.
Conference proceeding from COMSOL – conference Paris 2010, 2010, Paris, France.

Paper IX **The impact from building heating system improvements on the primary energy efficiency of a district heating system with cogeneration**
Johansson, P.-O., Lauenburg, P., Wollerstrand, J.
Submitted to ECOS 2011 (24th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems), Abstract accepted, 2011, Novi Sad, Serbia.

Summary of report A
Cavitation in control valves
Johansson, P.-O., Wollerstrand J.
Summary of the Swedish report "Kavitation i styrventiler - laboratorieundersökning", P.-O. Johansson and J. Wollerstrand. The full report is published by the Swedish District Heating Association, report 2009:45, 2009

My contributions to the publications

Paper I I performed the calculations and wrote the paper together with Patrick Ljunggren and Janusz Wollerstrand. Janusz also performed the field experiment.

Paper II I performed the field experiments, the calculations and wrote the paper together with Patrick Ljunggren with guidance from Janusz Wollerstrand.

Paper III I performed the simulations and wrote the paper together with Patrick Ljunggren and with guidance from Janusz Wollerstrand.

Paper IV I performed the field experiments and the calculations together with Patrick Lauenburg with guidance from Janusz Wollerstrand. Patrick Lauenburg wrote the paper.

Paper V I performed most of the simulations with assistance from Patrick Lauenburg and Janusz Wollerstrand. I wrote the paper together with Patrick Lauenburg.

Paper VI I performed the calculations and wrote the paper together with Klas Jonshagen with guidance from Magnus Genrup.

Paper VII I performed the field study and the calculations with guidance from Janusz Wollerstrand. I wrote the paper together with Janusz Wollerstrand.

Paper VIII I performed the simulations with guidance from Janusz Wollerstrand. I wrote the paper together with Janusz Wollerstrand.

Paper IX I performed most of the calculations with assistance from Patrick Lauenburg and guidance from Janusz Wollerstrand. I wrote the paper together with Patrick Lauenburg.

Summary of report A
I performed the experiments with guidance from Janusz Wollerstrand. I wrote the Swedish report and the English summary together with Janusz Wollerstrand.

Other related publications by the author:

Consequences of improvements in domestic hot water circulation circuits – Field Studies
Johansson, P.-O., Wollerstrand, J.
Conference proceeding from the 10th International Symposium on District Heating and Cooling, Hannover, Germany., 2006

Förändringar i tappvarmvattenanvändning vid införande av tappvarmvattencirkulation – fallstudie (Consequences of domestic hot water consumption when applying circulation– Case Studies)
Johansson, P-O.
Project report, Department of Energy Sciences, Lund University, Faculty of Engineering, 2006.

Fjärrvärmeanslutna byggnaders värme- och varmvattensystem-samverkan, komfort och sårbarhet (District heating connected buildings heating and domestic hot water systems- interaction, comfort and vulnerability)
Johansson, P.-O.
Licentiate thesis, Department of Energy Sciences, Lund University, Faculty of Engineering, 2007.

Optimal reglering av radiatorsystem (Optimized control algorithm for space heating system)
Wollerstrand, J., Ljunggren, P., Johansson, P.-O.
Report 2007:6, Swedish District Heating Association, 2007.

District Heating Supplies in Case of Power Failure
Ljunggren, P., Johansson, P.-O., Wollerstrand, J.
Conference proceedings from ECOS 2008 (21st International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems), Krakow, Poland, 2008.

Fjärrvärme vid elavbrott – slutrapport (District heating in case of power failure – final report)
Lauenburg, P., Johansson, P.-O.
Project Report, Department of Energy Sciences, Lund University, Faculty of Engineering, 2008.

Husinterna värmesystem (House-internal heating systems)
Johansson, P.-O. , Wollerstrand, J.
Project report, FoU 2005:136, Swedish District Heating Association, 2005.

Modelling space heating systems connected to district heating in case of electric power failure
Lauenburg, P., Johansson, P.-O., Wollerstrand, J.
Conference proceedings from Building Simulation 2009 (11th International Building Performance Simulation Association Conference and Exhibition), Glasgow, UK., 2009

Kavitation i styrventiler (Cavitation in control valves)
Johansson, P.-O. , Wollerstrand, J.
Report 2009:45, Swedish District Heating Association, 2009.

POPULÄRVETENSKAPLIG SAMMANFATTNING

Fjärrvärme är en väletablerad teknik som främst används för uppvärmning av byggnader och varmvatten. I Sverige värms över 80% av flerbostadshusen med fjärrvärme, vilket innebär att de flesta svenskar faktiskt bor i ett fjärrvärmeuppvärmt hus. Fjärrvärme är inte enbart en svensk teknik, utan används över hela värden där behov av uppvärmning finns. Länderna i Norden och Östeuropa har en betydande andel byggnader värmda med fjärrvärme.

Ett fjärrvärmesystem består av centralt producerat varmvatten som distribueras i ett rörnät till de anslutna byggnaderna. Genom en central produktion av värme ökar möjligheten till bränsleflexibilitet. Historiskt sett har fjärrvärme även bidragit till en kraftig minskning av oljeanvändning för uppvärmningsändamål i Sverige. Fjärrvärme utgör även grunden för en ökad kraftvärmeutbyggnad, något som fortfarande förespråkas både i Sverige och i Europa. I ett kraftvärmeverk produceras el och värme samtidigt till en hög verkningsgrad tillskillnad från elproduktion i kondenskraftverk där endast el produceras. För svenska förhållanden innebär en ökad kraftvärmeutbyggnad oftast att andelen el producerad med biobränslen ökar, något som är ett eftersträvat politiskt mål.

I denna avhandling berörs två områden relevanta för fjärrvärmeforskning: förbättrad temperaturutnyttjande i fjärrvärmesystemet och möjligheten att upprätthålla värmeförsörjning i fjärrvärmeanslutna byggnader vid elavbrott.

Inom ramen för det första området undersöks tre metoder för att förbättra temperaturutnyttjandet av fjärrvärme, vilket i praktiken betyder att sänka temperaturnivån i fjärrvärmenätet. De undersökta metoderna är utnyttjande av radiatorfläktar för att öka värmeavgivningen från radiatorer, optimerad styrning av radiatorsystemet och införande av alternativa kopplingsprinciper i fjärrvärmecentralen. Genom att använda radiatorfläktar ökar värmeavgivningen från radiatorerna, vilket innebär att samma mängd värme

kan avges vid lägre temperatur med radiatorfläkt i drift. Nyttan med radiatorfläktar och den optimerade radiatorstyrningen exemplifieras med en analys av förändringar av primärenergifaktorer för ett fjärrvärmesystem med kraftvärmeproduktion. När primärenergifaktorer beräknas för fjärrvärme med kraftvärmeproduktion allokeras den samtidigt producerade elen och betraktas som ersättning till el producerad med andra metoder. Hur man betraktar den ersatta elen har stor inverkan på fjärrvärmens primärenergifaktor. Om den producerade elen antas ersätta el producerad på marginalen i kolkondenskraftverk blir påverkan på fjärrvärmens primärenergifaktor stor då fjärrvärmetemperaturerna sänks. Om den ersatta elen istället hade producerats som den genomsnittliga nordiska elen blir påverkan på fjärrvärmens primärenergifaktor nästan helt försumbar. Hur den allokerade elen skall betraktas är därför avgörande för fjärrvärmens primärenergifaktor för fjärrvärmenät med kraftvärmeproduktion. Nya EU-direktiv angående energideklarering av byggnader föreskriver att energianvändningen skall anges i form av primärenergianvändning, varför denna fråga är av stor vikt.

Inom detta arbete har även byggnaders möjlighet att ta emot värme under elavbrott studerats. Genom fältstudier i bostadshus och lokaler där elavbrott har simulerats har det visat sig att för de flesta fall finns det goda möjligheter att upprätthålla, eller i alla fall förlänga tiden för en under omständigheterna acceptabel inomhustemperatur. Ur krisberedskapssynpunkt har det stor betydelse att det finns goda möjligheter för boende att bo kvar under långvariga elavbrott om försörjningen av det heta fjärrvärmevattnet kan upprätthållas.

CONTENTS

INTRODUCTION

At the present time, district heating is a well-established technology in many countries and plays an important role in energy supply. In Sweden, district heating accounts for approximately 80% of the heating of apartment buildings, and nearly 70% of construction category premises [1]. On a national level, district heating accounts for 56% of the total energy used for heating [2].

District heating is centrally produced heat which is distributed to customers through pipes buried in the ground. The production and distribution of district heating benefit from good cooling of primary water in the connected buildings. A low primary supply water temperature is also beneficial. The district heating supply temperature is dependent on the temperature and energy demand in the connected buildings. Therefore it is important to lower the temperatures needed for space heating, since these, provide the basis for the primary district heating temperatures during the heating season.

District heating enables the use of fuels and waste heat that would otherwise be difficult to use effectively in the energy system. Through simultaneous production of heat and electricity in combined heat and power stations, district heating contributes to an increased share of renewable electricity generation with a high overall efficiency. To evaluate and compare the effectiveness of different energy sources is complex but important work. According to a recently presented EU directive on the energy performance of buildings, primary energy factors are to be used for this comparison.

Although district heating is generally considered a reliable and robust heating source, recent storms and power outages, particularly related to the Gudrun storm in 2005, have increased the focus on secure heat supply in Sweden.

Scope of the thesis

The objective of this thesis is divided into three areas of interest, of which the two main ones are:

- Reduction of district heating temperatures and demonstration of the benefits of reduced district heating temperatures in terms of primary energy factors.
- Reliability of district heating supply during electric power outages.

In addition, an experimental study of cavitation in control valves is also performed.

Under the first objective, three different methods to lower temperatures in the district heating system are examined. The benefits of lowered district heating temperatures are exemplified with calculations of possible impact on primary energy factors for a district heating system with cogeneration and a heat-only boiler. The three methods to improve the temperature performance of the district heating connected buildings studied here are; use of add-on-fan blowers mounted on radiators, optimized control of space heating systems and use of alternative connection principles in district heating substations.

The second objective is to examine the ability of buildings connected to district heating to receive heat in case of a power failure. This part of the work is to a large extent performed through field studies in which the electricity to the circulation pump in the heating system and its control equipment is deliberately turned off.

The experimental study of cavitation is performed for a limited sample of commonly used primary control valves. According to data sheets from control valve manufacturers, cavitation may occur in the district heating context. Despite this, cavitation is generally not considered to be a problem. The aim of the experimental study is to identify the conditions in which cavitation occurs.

The three methods chosen to reduce district heating temperatures are all especially suitable for improvements in the existing building stock. If newly constructed buildings are considered, a larger variety of suitable low temperature heating systems are available. Examples of such systems are low temperature radiator systems and building-embedded heating systems such as

under-floor heating. These systems are not further studied within this thesis. The studies here focus on heating systems with radiators, which means that systems with, e.g., air-coil heaters are not considered.

Another limitation is that the thesis focuses on district heating connected buildings with indirect connection through heat exchangers. Even though the principle of add-on-fan blowers on radiators will, of course, also have an influence on the primary temperatures in DH networks using direct connection, no economic evaluation is carried out. The thesis mostly focuses on the technology and conditions relevant for Sweden, but not in any narrow sense.

The work starts out from a system perspective, where production units, district heating networks and space heating systems are affected. Analysis of district heating from a systems perspective is in itself nothing new: among others, Frederiksen and Werner have both presented PhD theses in which district heating is analyzed at a system level.

The title of this thesis is intended to accommodate the different papers within the title. However, it must be noted that the author has worked under the premise that the selection of papers is determined by how the work was financed at different stages. From a strict academic point, this has meant some difficulty in keeping the work a coherent whole. On the other hand, these conditions resulted in good opportunities to conduct studies that are judged important in the development of practical district heating technology. Readers who are aware of other technical dissertations published recently are probably familiar with this problem.

SYSTEMS APPROACH TO DISTRICT HEATING

District heating (DH) currently accounts for over half of the energy used for heating purposes in Sweden. In Sweden the use of DH differs for different building types. For apartment buildings and premises, DH is the predominant heating method, at 82% and 68% respectively [1]. For single family houses, electricity and bio-fuels are the dominant heating sources, and district heating accounts for only 12% [1]. DH is therefore an important part of the Swedish energy system. Also, DH is internationally an important part of the energy system, especially since DH allows use of efficient cogeneration technology.

A district heating system consists of one or more production units that produce heat, distribution pipes used for heat transportation and DH substations in the connected buildings where heat is transferred to the heating and domestic hot water (DHW) system. In the heat production facility, district heat can be produced in a heat-only boiler, heat pump or from waste heat. Also, combined heat and power generation (CHP) is common. In the DH-connected buildings, the space heating system and DHW system is connected to the DH network through a substation. The DH distribution pipes consist almost solely of buried insulated pipes that connect the production facilities and DH customers. Together, the distribution pipes form a distribution network consisting of a piping network in which hot water is circulated (see Figure 1). [3]

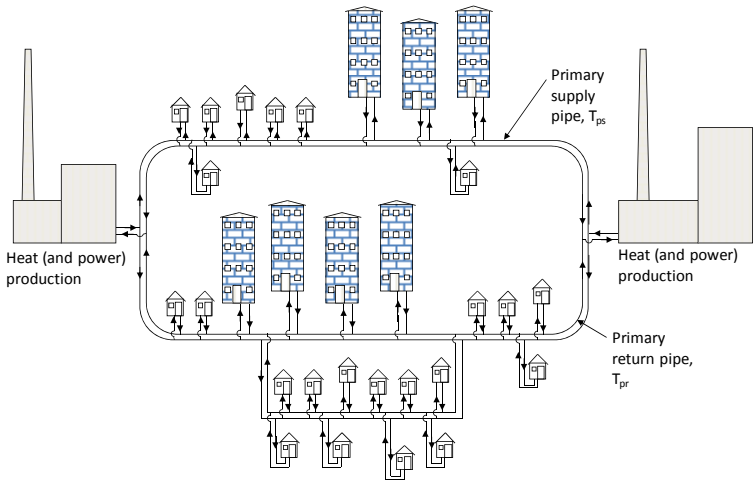

Figure 1. Principle of a simplified district heating network with two heat production units.

The central heat production in the district heating network offers environmental benefits through cost-efficient exhaust gas purification, efficient combustion and utilization of waste heat from industries and sewage treatment plants. This contributes to increased energy conservation. Another important aspect of the benefit of district heating is its large fuel flexibility. District heating not only contributes to heat supply, but also provides the basis for approximately 10% of Sweden's electricity production through the use of efficient cogeneration technology in which electricity and heat are produced simultaneously.

The water in the district heating network, called primary water, is circulated constantly with pumping power. A high cooling of primary water in the connected buildings is important for the efficiency of district heating network and affects the efficiency and/or the coefficient of performance in heat production units. The cooling of the primary water depends on a number of factors, such as the type of district heating substation and the temperature levels in the secondary system. An increased cooling of primary water reduces the pressure losses in the district heating network. The pressure losses are proportional to the squared flow. This results in a reduction of the pumping energy required to circulate the primary water. Increased cooling also reduces

the total heat losses in the district heating network due to decreased primary temperatures in the network.

The primary supply temperature in the district heating network depends on the heat load. During winter time, when the heat load is large, the district heating temperature is high, often around or above 100°C. The primary supply temperature decreases when the heat load decreases and reaches a minimum temperature, about 70°C, during low relative heat loads and during summer time. The fact that the primary supply temperature is higher during high heat loads is mainly because of two factors: 1) the temperature demand for space heating in the connected buildings is higher during winter; 2) to prevent excessive primary flow rate in the district heating network.

As previously mentioned, DH enables efficient utilization of various fuels. An evaluation of DH technology effectiveness can be done in several ways. The most obvious is a thermodynamic approach from which the efficiency can be evaluated. Today, when environmental issues are on the agenda, evaluation tools such as life cycle assessments (LCA) and CO_2 indicators are commonly used for different energy carriers. Use of primary energy factors for the energy carrier is another common method. The primary energy factor is similar to the LCA, but focuses more on energy input and less on environmental impact. According to the EU directive on building performance certification, the energy usage in buildings shall be expressed in terms of primary energy consumption. The primary energy concept is discussed in this thesis and used as an evaluation tool for lowered DH temperatures (see Paper IX).

Buildings connected to district heating

In DH-connected buildings, DH is used for space heating and domestic hot water (DHW) preparation. The hot DH water is, in Sweden, hydronically separated from the water in the DH network. The heat is transferred to the heating and DHW system through DH substations. The predominant type of heat exchanger used in DH substations is instantaneous plate heat exchangers. The DH substation can be designed according to several different connection principles. The two most common in Sweden are parallel and 2-stage connections. A schematic overview of different connection principles of DH substations can be found in [3] and [4]. In the section “District heating substations - connection principles” and in Paper V the district heating return temperature is evaluated for substations with two alternative connection principles.

In many countries the most common type of heating system in buildings connected to DH is waterborne radiators. The heating system is designed to have a maximum heat output at dimensioning outdoor temperature (DOT). The heating demand in the buildings is then considered to linearly decrease to zero when the outdoor temperature increases to the balancing temperature, at which point the heating demand is covered by the internal heat production. Space heating systems are often designed for a constant mass flow through the radiators, while the heat output from the radiators is controlled by regulating the radiator supply temperature according to a preprogrammed scheme called the radiator temperature program. The radiator temperature program is a function of the outdoor temperature and a time constant for the building. In some cases the indoor temperature is used as a feedback signal. Within the building the heat output from the radiators is also often controlled locally by using thermostatic radiator valves which can limit the mass flow through the radiator. The secondary water used to distribute the heat from the DH substations to the radiators is circulated with an electric circulation pump.

The heat output from the radiators arises from convection and radiation. The relation between the share of radiation and convection depends on the design of the radiator. The DH return temperature, during the heating season, depends strongly on the return temperature from the space heating system. Reduced temperature levels in the space heating system are therefore an issue of interest for DH research, and also within this thesis. Paper I describes an alternative method to control the heat output in the space heating system to lower the primary DH return temperature. In Paper VII and Paper VIII a method to improve the heat output from existing radiator systems is described with the aim of reducing the temperature demand for space heating.

For DHW temperatures there are specific guidelines provided by the Swedish National Board of Housing, Building and Planning (in Swedish “Boverket”). At the taps, the DHW temperature must exceed 50°C to avoid the growth of legionella, but be lower than 60°C in order to avoid scalding [5]. To reduce the waiting time for DHW, the hot water can be circulated within the building. This is commonly the case for larger buildings such as multifamily houses. In the case of DHW circulation, the temperature of the circulated water must be at least 50°C in the whole DHW circuit. This results in a rather high primary return temperature when no consumption of DHW occurs.

METHODS TO EVALUATE ENERGY USAGE

In many developed countries it has for a long time been a primary objective of energy policy to save energy. Standards for energy consumption in new houses have constantly been tightened, and there are now houses known as "passive-houses". Giving priority to energy-saving is indeed included in this thesis as an important premise, but not in the sense that it overrules our other goals.

When energy savings associated with different heating technologies are implemented, it is important to have an insight into how the energy system is structured in order to range different types of energy-saving efforts properly. In this context it is essential to be precise about which system boundaries should be included in such an assessment. For example, investment in exploitation of CHP can be more efficient than increased building insulation, as shown by Joelsson [6].

Energy use can be evaluated in a number of different ways. The preferred approach will often depend on who is evaluating and for whom the evaluation is made. Example of evaluation methods for energy usage in buildings are:

- Classic thermodynamic system performance
- Economic performance
- Environmental performance
- Performance according to the primary energy concept

In this thesis the primary energy approach is considered, in which energy usage is evaluated according to the relatively new regulations set out in the EU Energy Performance of Buildings Directive (2010/31/EU [7]). This intends to promote a widened system approach when energy consumption is presented. What makes it particularly interesting in the present context is that it tends to improve the competitiveness of district heating.

By using primary energy consumption or primary energy factors for different energy carriers, not only the bought energy, but also the upstream energy usage is considered. The primary energy concept has many similarities to life cycle assessments (LCA), but with the difference that the primary energy approach focuses on the energy chain while LCA focuses more on environmental impact [8]. With the use of primary energy factors, a wider system approach is considered. This could be a useful tool when energy efficiency decisions are made. Joelsson and Gustavsson show that the choice of heating system and energy carrier has a greater impact on the primary energy consumption than house envelope measures [6].

In the following section, we define the primary energy concept, show how it is implemented in the DH context and give an overview of implementation in some EU countries.

Primary Energy

The European Environment Agency (EEA) has defined primary energy as [9]:

> *"Energy embodied in natural resources (e.g. coal, crude oil, sunlight, uranium) that has not undergone any anthropogenic conversion or transformation".*

The evaluation of a building's energy supply will often be made quite differently depending on how the system boundaries of the control surface are defined. A conventional energy bill does not consider the upstream energy demand in a larger energy system including production, conversion and distribution, etc. With a primary energy approach, this upstream process is considered and the energy supplied is converted into primary energy by multiplying the supplied energy with a primary energy factor (PEF). Figure 2 gives an overview of how different definitions of the system control surfaces will affect the energy performance of a building.

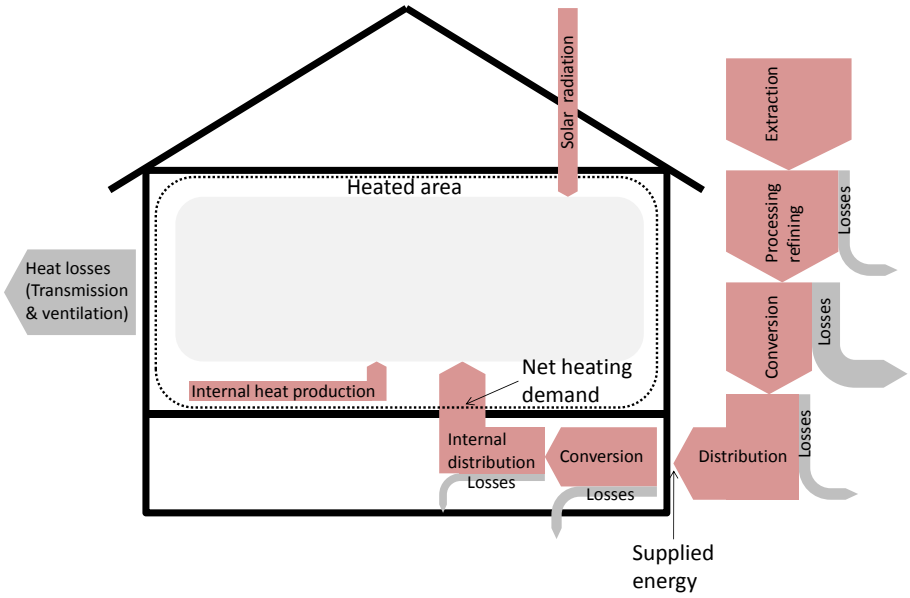

Figure 2. Illustration of primary energy system boundaries.

In standard EN 15603:2008, a primary energy concept is defined and primary energy factors (PEFs) are given for thirteen energy carriers based on average European values [10]. To compute the primary energy usage, the energy used is multiplied by the PEF for the energy carrier used. The standard defines two PEFs, one PEF-total (PEFT), and one PEF-non-renewable (PEFN-R). The PEFT is a sum of the non-renewable and the renewable primary energy factor [10].

By using primary energy as an evaluation tool for energy consumption, a cradle-to-grave perspective is possible. Also, the thermodynamic efficiency is included when the primary energy factors are calculated for different energy carriers.

At present there is a lack of standardized PEFs for several energy carriers. This may cause improper decisions to be made regarding primary energy efficiency and unnecessary CO_2 emissions. [11]

Primary energy concept and district heating

In a DH network, the heat delivered can be generated from various primary energy sources. When calculating the primary energy factor, this must be taken into account.

In the European standard EN15316-4-5, PEF system boundaries for DH are described. In general, the PEF for DH is the quota of primary energy input divided by the energy delivered at the control surface of the supplied buildings (Q_{del}) [12], as shown in equation (1).

$$f_{P,DH} = \frac{E_P}{Q_{del}} = \frac{\sum_i E_{F,i} \cdot f_{P,F,i}}{\sum_j Q_{del,j}} \tag{1}$$

The primary energy input in equation (1) is the sum of the energy input (fuel), E_F, in the production unit multiplied by the respective PEF ($f_{P,F}$). However, in cases of cogeneration, when both heat and electricity are produced simultaneously, equation (1) is not appropriate. In EN 15316-4-5 a method based on the energy balance, equation (2), is given [12]

$$f_{P,DH} \cdot \sum_j Q_{del,j} + f_{P,el} \cdot E_{el,CHP} = \sum_i E_{F,i} \cdot f_{P,F,i} \tag{2}$$

where:

$E_{el,CHP}$ is the amount of cogenerated electricity exported outside the system border

$f_{P,el}$ is the PEF for the amount of electricity being substituted

From the energy balance, the PEF for DH can be determined according to equation (3). This approach is often called the Power Bonus Method.

$$f_{P,DH} = \frac{\sum_i E_{F,i} \cdot f_{P,F,i} - f_{P,el} \cdot E_{el,CHP}}{\sum_j Q_{del,j}} \tag{3}$$

From equation (3), it can be seen that the PEF for DH depends on how the PEF for the substituted electricity ($f_{P,el}$) is defined. This opens up a discussion of which value of the PEF for the replaced electricity should be used. For

example, the PEF for electricity produced in hydro power stations and in coal-fired power stations differs greatly.

Therefore, the main question is which electricity mixture should be considered. One approach is to consider the electricity as marginal electricity produced in coal-fired condensing power stations.

In Paper IX an analysis of the impact of three different PEFs for electricity is performed for a DH system with heat production mainly based on a CHP station. The three PEFs tested for electricity arise from: 1) coal-fired condensing power stations; 2) the EU electricity mixture; 3) the Nordic electricity mixture. The results presented in the paper show that the influence of how the substituted electricity is produced is substantial.

In a Swedish study from 2009, it was stated that the choice of how the substituted electricity is considered has only a minor impact on the PEF for district heating, but a large impact on CO_2 emissions [13]. This is because of the low PEF for electricity produced with coal that is used in the study. The PEFs used for substituted electricity in the study are: 2.5 if from coal-fired power stations; and 1.8 if Nordic electricity is considered. In the EU standard [10], the PEF for electricity produced by coal-fired power stations is 4.05 and in Paper IX the PEF for electricity in the Nordic mixture is calculated as 2.15. With these values, the impact of how the allocated electricity is considered can be seen to be substantial, as shown in Paper IX.

The primary energy factor for a DH network depends strongly on how the heat is produced. First of all, waste heat utilization reduces the PEF for DH. A large share of heat produced with CHP also reduces the PEF if the produced electricity is considered to replace electricity produced at the margin. When discussing incentives for increased heat and power production, Carlson states that [14]:

> *"To keep the district heating as a good basis for CHP production in the nearby future, it is necessary to use the primary energy concept when energy efficiency is discussed".*

In an annex of a recent SABO report [15], the PEFs for 372 Swedish DH networks are presented. The average PEF for DH in Sweden is set at 0.98, but

the variation is large (see Figure 3). Similar results were presented in a study by Gode et al. [13].

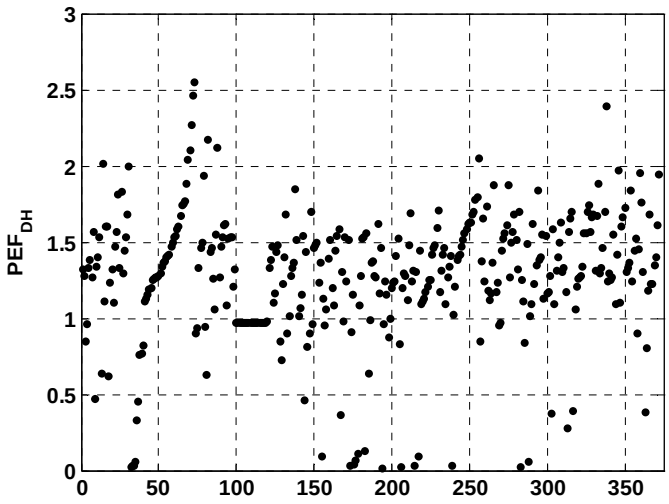

Figure 3 . PEFs for 372 DH networks in Sweden (based on data from [16]).

As seen, the PEFs for DH vary widely for different DH networks. To determine the primary energy consumption for a specific DH-connected building, it is therefore important to use the local, not the national average, value for the PEF. The use of local PEFs for DH opens up the possibility of visualizing the benefits from an increased share of the heat utilized from waste energy and CHP production. The EU directive 2010/31/EU states that [7]:

> *"The energy performance of a building shall be expressed in a transparent manner and shall include an energy performance indicator and a numeric indicator of primary energy use, based on primary energy factors per energy carrier, which may be based on national or regional annual weighted averages or a specific value for on- site production".*

Joelsson and Gustavsson have studied the reduction of primary energy consumption in detached houses with different end-use technologies for heating purposes. Detached houses supplied by DH, together with those supplied by heat pumps, have the lowest primary energy consumption, while electric heated houses have the largest consumption [6], [17]. Also, Karlsson

shows similar results when comparing different heating sources using a primary energy approach [18].

Implementation of primary energy concept

At the present time it is unclear to what extent this directive will actually have an impact in practice, both in the short and the long term. Consistent implementation of the directive involves both principal and practical difficulties. However, the author of this thesis has chosen to view the directive as sufficiently important to merit an attempt to analyze how the different approaches used in DH technology are affected in accordance with the directive.

On October 1st 2006 a Swedish regulation regarding the energy performance of buildings was implemented [19]. The aim of this regulation is to reduce the energy demand of residential and non-residential buildings by stipulating informative instruments. The energy performance certificate must be publicly available, must include the energy performance values and a reference value, and should also contain suggestions for energy improvements [19]. In the Swedish handbook for the energy performance of buildings certification, the energy demand is defined as the delivered energy [20]. In a new EU directive (2010/31/EU), stricter requirements than those currently in effect in Sweden are set out regarding the energy performance of buildings.

In Sweden, Boverket (Swedish National Board of Housing, Building and Planning) is the authority that sets the guidelines for the energy performance of buildings. Boverket also sets standards for new buildings in the national building regulations [5]. In a publication [21] concerning consequences of the EU directive, Boverket states that:

> *"I Sverige har vi inget uttryckligt krav på numerisk indikator för primärenergi men SIS skulle kunna ta fram ett klassningssystem med indikatorer även för primärenergi. I dagsläget skiljer vi bara mellan användning av el och användning av annan energi. Svenska byggregler har strängare krav för eluppvärmda byggnader."*

("In Sweden we have no explicit requirement for the numerical indicator for primary energy, but the SIS[1] could develop a rating system with indicators also for primary energy. In the current situation, a distinction is only made between the use of electricity and use of other energy carriers. The Swedish building regulations have more stringent requirements for electrically heated buildings.")

This is stated when the new expression concerning the requirement for the energy performance of buildings to include a numerical indicator of the primary energy use is discussed. In an impact assessment under consideration on the revision of the Swedish building regulations, in the section on energy conservation in buildings, the primary energy approach is also discussed. It is stated that there is no legally binding requirement to adopt the primary energy concept in the national regulation of buildings, and such an approach is not included in the regulation concerning new buildings [22]. The argument for not adopting the primary energy approach is that new buildings will exist for a long time, whereas the energy supply might change during the building's lifetime [22].

However, several European countries have already adopted the primary energy approach in both building regulations concerning new buildings and energy certifications of existing buildings. Germany, France and Poland are examples of large EU countries where primary energy consumption is adopted in the energy performance of building certification [23]. In an article [24] Kurnitski summarized the various calculating methods for energy use in building regulations concerning new buildings for EU member states present in June 2008. The energy usage requirements implemented are divided into three groups [24]:

1) Requirements on component level (e.g. Finland)
2) Requirements on energy frame (e.g. Sweden, Norway)
3) Requirements on primary energy use (e.g. Germany, France, UK)

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[1] SIS is the centre for work on standards in Sweden and a working partner in the European and global networks, CEN and ISO, of which Sweden is also a member. SIS is a member-based non-profit association. [25]

The trend regarding these requirements is that more countries are including a primary energy approach for energy utilization in building regulations for new buildings [24].

METHODS TO LOWER AND EVALUATE DISTRICT HEATING TEMPERATURES

Generally it is well known that low DH temperatures are beneficial for the DH network. Heat losses from DH networks decrease when the temperatures are lowered and production units increase their efficiency. Lowered DH temperatures also increase the lifetime of the DH piping network and open up possibilities for an increased use of plastic DH pipes which are more sensible for high temperatures. Altogether, this has made primary DH temperatures an interesting field within DH research.

In order to use the DH network as efficiently as possible, it is a central issue to maximize the cooling of DH water in DH substations. This allows the same amount of heat to be supplied to the DH connected buildings with a reduced DH mass flow. Often, the energy consumed is not the only basis for the price of DH. In Sweden, both the consumed energy and the primary cooling or flow-rate affect the energy bill for a large share of the energy supplied from DH [26]. With this type of price model, not only the DH companies but also the connected DH customers favor increased cooling.

There are a number of well-known ways in which district heating network supply and return temperatures can be lowered. Several of these fall into the category of increasing dimensions, such as piping diameters, size of the heat exchangers in substations, and size (and number) of radiators within buildings. A further example is improved insulation of buildings, since this will allow radiators of a given size to be operated at lower supply and return temperature.

For a given size of district heating systems, including internal distribution systems within buildings, there are degrees of freedom when deciding on operation. For instance, at a given heat load of a network, it may be possible to lower the supply temperature at the expense of an increased amount of circulated water, which in turn will result in an increased supply of power to circulation pumps. This pre-supposes that there is a margin in terms of systems pressure. For example, if the lowest pressure level is controlled to be fixed, the

maximum allowable flow-rate will be dictated by the maximum allowable pressure level in the system, for instance 16 bar.

In this work, three methods to reduce the primary DH return temperature are studied. The methods are utilization of:

- Increased heat output from radiators with add-on-fan blowers
- Alternative connection principles
- Optimized control of hydronic heating system

All these three methods are very suitable for implementation in the existing building stock, since no major changes are necessary in the secondary hydronic heating system, which is very costly. When heat output from the radiators is increased according to the first strategy, the space heating temperature demand in the DH connected buildings is reduced. This allows not only lowered DH return temperatures, but also a lowered DH supply temperature.

Looking at the construction dates of Sweden's building stock (see Figure 4), it is seen that many multifamily houses were built before 1970 with significantly lower insulation standards than the current state of the art. Space heating systems in older buildings are often considered as oversized due to conservative original dimensioning or recent energy improvement measures within the building. If so, the advanced control of the heating system is an efficient method to lower the DH return temperature. The implementation of the optimized control of the space heating system can be adopted without any changes in the heating system except to the control algorithm and speed control of the circulation pump.

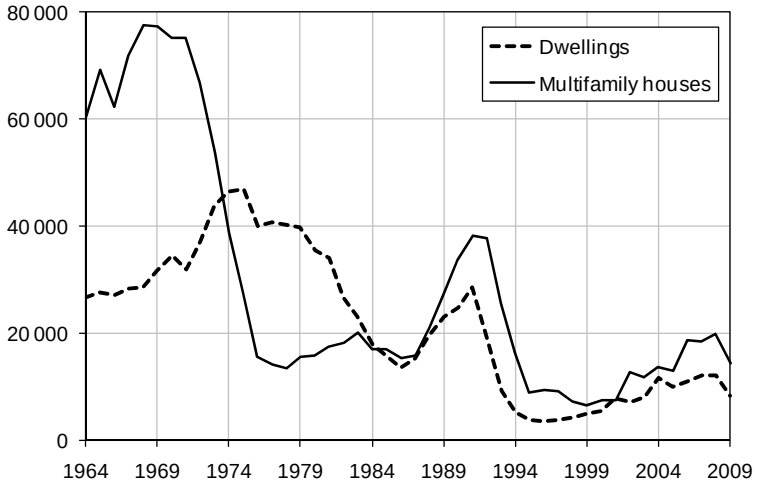

Figure 4. Construction date for Swedish building stock. [27]

The use of add-on-fan blowers makes it possible to reduce the temperature program dramatically in an existing radiator system, without the need to replace the radiators with low temperature ones. Since the space heating temperature program is reduced by using add-on-fan blowers, the DH supply temperature can also be reduced. As early as 1977, Brumm [28] was advocating fans mounted on radiators to reduce space heating temperatures for existing radiators. The add-on-fan blowers can be installed in a whole building to reduce the space heating temperature program, or just in some parts of it where heat output from the radiators is insufficient.

Even though studies have shown that the connection principle of the DH substation is not considered to be the major factor to achieve good cooling of the DH water (see e.g. [29]), the influence of different DH connection principles has a substantial impact on the DH return temperature when low temperature heating systems are considered. In Paper V the DH return temperature is calculated for buildings using DH substations with alternative connection principles.

Both replacement of the DH substation and utilization of the optimized space heating temperature program can be achieved without access to the radiators in the space heating system, which in many cases is beneficial.

As pointed out in [30] , the function and the balancing of the secondary system is an important prerequisite to achieve good cooling of the district heating water. However, failures in the district heating substation can cause unnecessarily high primary return temperatures. Typical failures are fouling of heat exchangers, leaking control valves or other control problems. Cavitiation in control valves, which can cause leakage, has been studied experimentally in [31] (see also Summary of report A). In the report the cavitation phenomena is examined for six commonly used control valves. The results show that for conservatively sized (meaning oversized) control valves cavitation is uncommon during normal operation. However, high differential pressure and large opening of the valve increase the tendency to cavitate [31]. It is especially important to keep this in mind in large district heating networks with areas with high differential pressures, and in district heating networks with lowered primary supply temperature at the expense of increased primary flow rate. Malfunctions that can arise from cavitation are material damage within the control valve and noise problems. A partially damaged control valve may not close completely, which is a problem especially during summertime when no space heating is required.

During years of research, many studies have focused on the impact of changed DH temperature levels in a DH network. One example of such a study is the Swedish computational program “LAVA-kalkyl” [32], in which the impact of changed DH temperatures is computed for a DH network. The results are then presented in economic terms and as changed levels of emissions. In other studies, the benefits of reduced DH return temperature are described in more general terms. Werner, for instance, estimates the economic benefits from reduced DH return temperature to 1 kr/MWh°C [33]. In Paper IX, the benefits from an optimization of the space heating control algorithm and use of add-on-fan blowers in a heating system are evaluated using a primary energy approach.

In the following sections the three methods of lowering DH temperatures are presented. Then the impact of lowered DH temperatures in a CHP station is described. The last section presents an evaluation of the implementation of optimized control of a hydronic heating system and reduced space heating temperature program with the use of add-on-fan blowers in terms of changed primary energy factors for DH.

The calculations and modelling of DH substations, heat exchangers and radiators are based upon principles described by Gummérus [34]. The models based on these principles have been used for several years at our division, and have over the years been further developed by Wollerstrand [35] and Persson [36], among others. Theoretical calculations are performed using Matlab® software, and real-time simulations are performed in Simulink®.

Add-on-fan blowers

The heat output from a radiator arises from radiation and natural convection. Both radiation and convection are dependent on the temperature of the radiator and the surrounding temperature. For a standard panel radiator without a convection panel, the share of heat output from radiation is about 50% [37]. By mounting fans under or on top of an existing radiator the total heat transfer coefficient for the radiator can be increased due to the introduction of forced convection. At present, some simple systems based on this principle can be found on the market (see e.g. [38], [39] and [40]), but these products are today marketed only to a small extent. Within this study the effect of increased heat output from a radiator with add-on-fan blowers is studied. In Figure 5 a schematic illustration of an add-on-fan blower mounted on a panel radiator is presented.

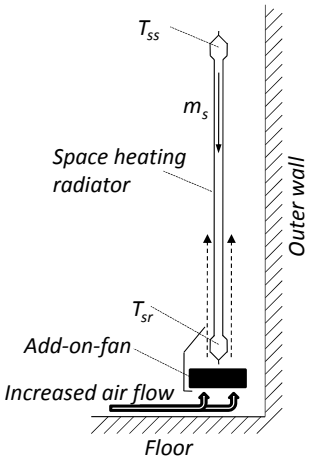

Figure 5. Principle of add-on-fan blower. [Paper VII]

Paper VII describes a field study with add-on fan blowers where heat output from the radiators is increased by more than 60%. The increased heat output

is, of course, dependent on the speed of the fan, which is studied in Paper VIII through CFD simulation of a panel radiator equipped with add-on-fan blowers.

With an increased heat output ratio from the radiators, the space heating temperature program can be lowered without reducing the amount of delivered heat (see Figure 6). This is according to the same principle as when dealing with oversized space heating systems where either the flow through the radiators or the supply temperature to the radiators must be lowered in order to avoid too much heat output and too high indoor temperatures. A simple illustration of this phenomenon can be found in a licentiate thesis by Ljunggren [26].

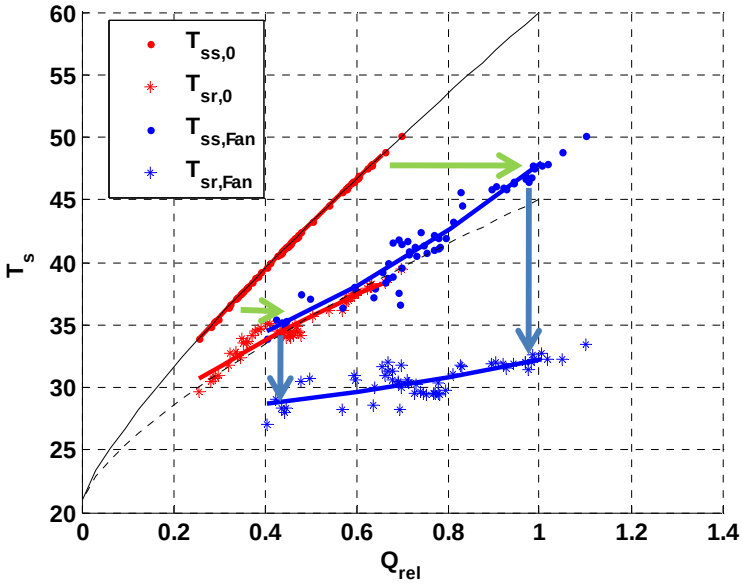

Figure 6. Space heating temperatures with and without add-on-fan blowers in operation. (Red markers without fan operation. Blue markers with fan in operation.) [Paper VII]

A similar effect would be achieved if the existing radiators were replaced by hydronic fan-coil heaters. However, installation of add-on-fan blowers is cheaper and a hydronically non-intrusive method.

The electricity demand for fan operation in the study is in the magnitude of 1% of the heat output at the dimensioning outdoor temperature with a 60/45°C space heating temperature program. The physical work performed by the fan is less than 0.1%, which indicates that the efficiency of the add-on-fan blower could be increased. The lowered space heating temperatures would normally result in lowered district heating return temperature, but can also be used to lower the primary supply temperature without increasing the flow rate through the district heating substation.

The new temperature programs for radiators presented in Paper VII are based on a field study of two types of radiators equipped with add-on-fan blowers. To complement this, a simulation with COMSOL software is performed for a panel radiator with add-on-fan blower in Paper VIII. The simulations of the add-on-fan blower are made with varying air velocities. In Paper VII and Paper IX the possible reduction of primary DH temperatures is calculated. In Paper IX the benefits from reduced primary DH temperatures are presented in terms of reduced primary energy consumption.

A drawback of using the add-on-fan blowers is the consumption of additional electricity. However, by lowering both the DH supply and the return temperature, more electricity can be produced in CHP stations, since the power-to-heat ratio is affected. For relatively high space heating loads the additional electricity is greater than the additional electricity consumed by the electricity-driven add-on-fan blowers. Figure 7 illustrates the possible increased electricity production for a DH network supplied with heat by a CHP station and heat-only boiler used for peak and base load heat production (see Paper IX for a detailed description of the DH network). The additional electric power used by the fan is a constant 1% of the dimensioned heat load during the heating season. The electricity used for fan operation is subtracted from the electricity produced.

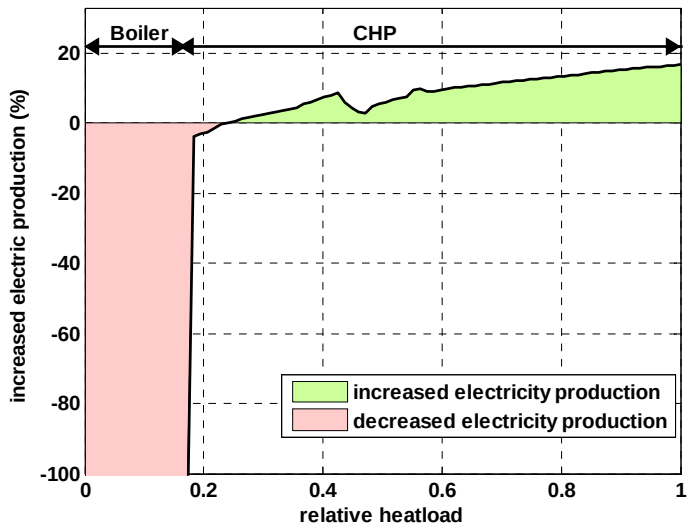

Figure 7. Possible increased electricity production in a DH network with utilization of add-on-fan blowers.

As seen in Figure 7, the operation of the add-on-fan blowers is beneficial for the electrical production for most of the heating season. Only when the CHP station is in operation at low relative heat loads do the add-on-fan blowers use more electricity than the additional electricity production. At low heat loads, when all the heat is produced in heat-only boilers, the fan is of course not beneficial. In Paper IX a control strategy for the operational time is considered when evaluating the benefits from add-on-fan blowers.

District heating substations - connection principles

At present two types of district heating substations are predominant on the market: the simple parallel-connected and the more advanced 2-stage connected substation. The parallel-connected substation is simple and robust in its design with only slightly lower primary cooling than the two-stage connected substation. Parallel-connected substations predominate in smaller buildings with relatively low heat demand, e.g. single family houses and small apartment buildings. On the Swedish market 3-stage connected substations also occur. The connection principle provides somewhat better primary cooling than the other variants, but is more complicated [3].

In a comparison of old and new district heating substations, it has been stated that the connection principle of the substation is of secondary importance when the annual cooling of primary water is considered [30]. Despite this, it is still of interest to study alternative connection principles in combination with modern low temperature heating systems and for buildings with reduced space heating demand.

Alternative connection principles

In Paper V primary return temperatures for district heating substations with four different connection principles are calculated. A parameter study is made for buildings of different sizes and with different temperature programs for space heating. Also, the effect of reduced domestic hot water use is calculated. The studied connection principles are: parallel, 2-stage, Russian 3-stage (R3) and series connection. The idea of the R3-stage and the series connection principle is to utilize a rather high primary temperature after the domestic hot water heat exchanger, especially when no hot water is used. Both these connection schemes require more sophisticated control equipment than conventional schemes. See Figure 8 for a schematic view of the two alternative connection principles. The performance of the R3-connection has been studied previously in e.g. [41] and [42].

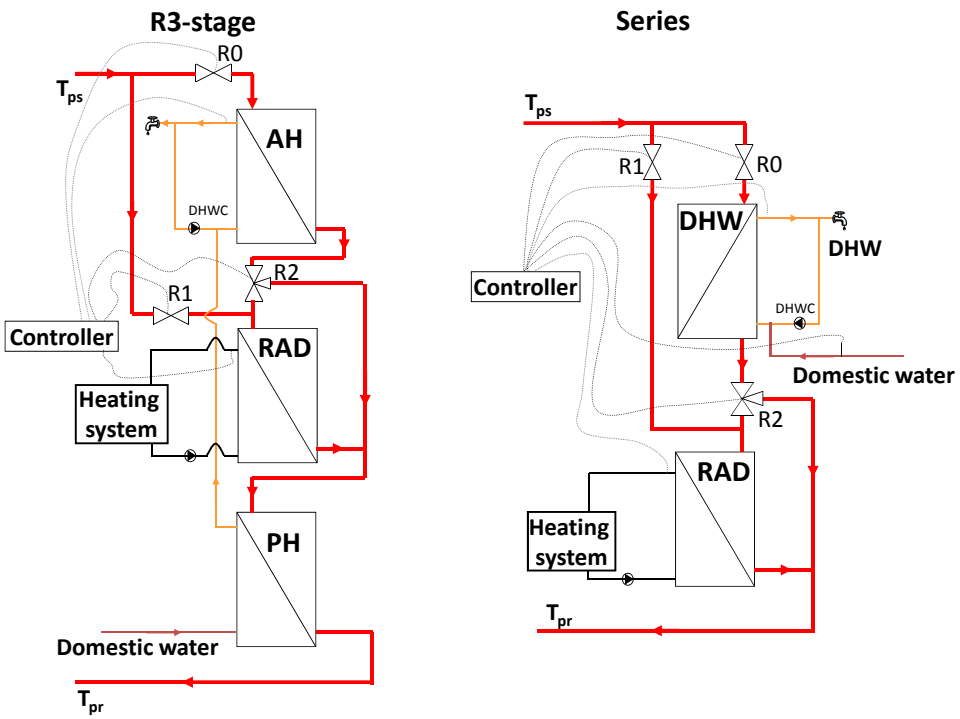

Figure 8. The two alternative connections evaluated in Paper V.

For the R3 connection the valve R0 controls the primary flow through the after heater (AH) in order to ensure proper DHW temperature. After leaving the AH, the primary flow continues, during the heating season, through the space heating heat exchanger (RAD). In case of overheating of the space heating system, valve R2 opens and allows a bypass flow from the AH direct to the pre heater (PH) providing that valve R1 is closed. Otherwise, if the desired space heating temperature is not reached, valve R1 opens too and allows primary flow of T_{ps} to mix with the primary flow leaving the AH. All primary flow passes through the PH.

At first sight, the series connection looks identical to the R3 connection except for the absence of PH. But in contrast to the R3-stage connection, the series connection can alternate between two different operating modes, depending on whether DHW is consumed or not. When domestic hot water is used, the valve R2 will fully open the bypass port, and the substation will work as a regular parallel-connected substation. In this case the radiator heat exchanger is

supplied with district heating water through valve R1. Otherwise, when no DHW is used, the series connection operates according to the same principles as the R3 connection.

The results presented in Paper V show that the R3-stage heat exchanger always results in the lowest primary return temperature. However, for buildings with low consumption of domestic hot water but high DHW circulation losses, e.g. non-residential buildings, the series connection can be competitive. It also turns out that the benefits from the alternative connection principles are greater in the case of low temperature building heating systems.

The calculations of primary return temperature in Paper V were made with real time simulations of a heating and domestic hot water system performed with Simulink® software.

The domestic hot water consumption patterns were based on a statistical model described by Arvastson and Wollerstrand [43]. The patterns are shown in Appendix 1.

Advanced control of hydronic heating system

To achieve good cooling of the primary district heating water, it is important that the radiator system is well balanced. A radiator system can be balanced according to several different principles. The heat output from a radiator system can be controlled in two ways, by adjusting the flow through the radiators or the supply temperature to the radiators.

During the heating season the district heating return temperature is strongly dependent on the temperature levels in the heating system [44]. The design temperatures used in the heating system at the dimensioning outdoor temperature are typically 60/40°C, 60/45°C, 80/60°C, although other systems temperatures do occur. Another relatively commonly used radiator control strategy is the low-flow balancing method, a method based on a strongly reduced flow through the radiators and a higher supply temperature, typically 80/30°C. This results in a lower return temperature from the radiators. This low-flow balancing method has been shown in several studies to give a low primary return temperature (see e.g. [45]).

Typically, the heat output from the radiators is controlled by the radiator supply temperature (T_s), while the flow through the radiators is assumed to be

unchanged. The two basic relationships that describe the heat output from a radiator are included in equation (4):

$$Q = \dot{m}_s \cdot C_p \cdot (T_{ss} - T_{sr}) = U \cdot A \cdot \Delta\theta^n \qquad (4)$$

where $\Delta\theta$ is the logarithmic mean temperature difference between the radiator surface and the surrounding temperature. n is the radiator exponent (typical value: $n \approx 1{,}3$), A (m^2) is the area of the heat transfer surface and U ($W/(m^2 \cdot K^n)$) is the simplified heat transfer coefficient described by Trüschel [37].

By applying equation (4), the temperature program for a radiator can be derived for each and every heat load with a certain radiator flow for a specific radiator with known values of U, A and n. Since the heat output is controlled by varying supply temperature while the flow through the radiator is maintained, the supply temperature program is not linear with the outdoor temperature, despite the fact that the heat load is assumed to decrease linearly with increasing outdoor temperature.

By combining the relationship for the heat output in the heating system and the heat transfer relationship in a heat exchanger, the primary return temperature can be calculated for an arbitrary heat load and primary supply temperature. The following equation (5) describes a simplified relationship for heat transfer in a plate heat exchanger. The equation is described by Wollerstrand in [35].

$$Q = \dot{m}_p \cdot C_p \cdot \left(T_{ps} - T_{pr}\right) = \frac{c_0}{\frac{1}{\dot{m}_p^n} + \frac{1}{\dot{m}_s^n}} \cdot A \cdot \theta T \qquad (5)$$

where c_0, n and A are constants for the heat exchanger. θT is the logarithmic mean temperature difference between the water on the primary and the secondary side of the heat exchanger.

The temperature efficiency for a heat exchanger decreases when the heat load decreases. This is illustrated by Gummérus [34]. This applies for a heat exchanger connected to a space heating system controlled traditionally with a constant secondary flow rate.

Optimized space heating system operation

If the heat output from the space heating system is not limited by being controlled only by the supply temperature to the radiators, the flow through the radiators ($\dot{m}_s$) can also be varied. The temperature efficiency of the radiator heat exchanger can then be improved and optimized for all heating different loads.

This interaction between the DH network, DH substation and the connected buildings was in focus when Frederiksen and Wollerstrand concluded 20 years ago that [46]:

> *"In all cases of heat exchanger design, at all heat loads, and at all primary forward temperatures, an optimal flow-rate exists in the radiator circuit, i.e. a theoretical value giving a minimum return temperature on the primary side".*

This can be achieved by combining equations (4) and (5) to find the combination of secondary supply water temperature and radiator flow that provides the lowest possible primary return temperature for a given heat load and primary supply temperature.

With modern technology it is possible to utilize this insight by controlling not only the space heating supply temperature, but also the rotational speed of the circulation pump and thereby also the mass flow in the space heating system. In Paper I static calculations of lowered primary return temperature for such an approach are presented. Also, a method for a field experiment is presented where the space heating mass flow is reduced stepwise, showing how the primary return temperature is then affected. Based on this method a suitable adaptive control algorithm can be derived for a certain heating system. Theoretical calculations in Paper I show that the annual DH return temperature can be lowered by several degrees Celsius on an annual average basis. Calculations of the lowered return temperature are also made for oversized space heating systems, which are often found in the existing building stock.

In an ongoing project at our division this strategy is currently being implemented in DH substations within some residential buildings in Karlshamn, Sweden. The DH substations are equipped with digital controllers equipped with software add-ins which permit online modification of the

behavior of the control algorithm and online monitoring of the performance. The “agents” used in Karlshamn are described by Johansson in [47].

Impact of district heating temperatures on a CHP station

District heat can be produced in a large variety of production units using different fuels and technology for heat production. An often promoted technology is combined heat and power generation (CHP). Today, 35% of Sweden’s total district heat is produced with CHP technology [48]. The simultaneous electricity production is equivalent to almost 10% of total Swedish electricity consumption in 2008 [48]. The share of national electricity production that arises from CHP varies greatly between different countries. This can be explained by the large variation in the expansion of district heating, as well as political and local conditions. In an international context, it can be mentioned that the share of national electricity production that arises from use of CHP technology is roughly 10% in the US and slightly more than 10% in Germany [49]. The nations with the highest share of electricity produced in CHP stations are Denmark 50%, Finland 40% and Russia 30% [49]. The countries with a high share of electricity produced in CHP stations have one thing in common; the basis for utilization of district heating is large. Within the European Union there is a directive with a clear ambition to expand electricity produced from CHP stations [50]. This is because electricity produced with CHP technology has a significantly higher overall efficiency than regular power plants where only electricity is produced.

The configuration of a CHP station connected to a district heating network can be carried out in many different ways. This study focuses on a back-pressure CHP station with two condensing pressure levels and the district heating system as the only heat sink. This means that the operation of the CHP station is totally dependent on the temperatures and the heat load in the district heating network (see Figure 9). Other configurations of CHP stations are also found. One example is CHP stations capable of using sea water for further expansion in the steam turbine to increase the electricity production and/or the annual operational hours. Another example is to use a re-heater to increase the electricity output.

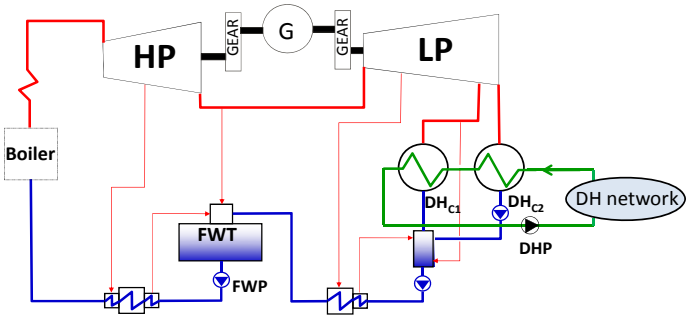

Figure 9. Simplified principle of a backpressure CHP station with two condensers connected to a DH network.

In a CHP station with only one condenser pressure level, the amount of electricity produced is totally dependent on the district heating supply temperature, but for CHP stations with two or more condenser pressure levels the electricity output is also, to some extent, dependent on the district heating return temperature [3].

In order to increase the total efficiency of a CHP station, flue gas condensation can be installed. This allows heat from the flue gases to be utilized. This is at the expense of the electricity production, since the heating demand for the CHP station decreases [51]. In the case of a heat-only boiler, however, flue gas condensation is very appropriate since the overall efficiency increases substantially, and may increase even further if the district heating return temperature is lowered.

Case study

In Paper VI the power-to-heat ratio for an existing CHP station located in Enköping is calculated for different temperature levels in the DH network. The analyzed CHP station is of a common configuration in Sweden, with two condenser pressure levels and using the DH network as the only heat sink. The operational hours of the CHP station are dependent on the present heat demand in the DH network. The aim of the paper is to visualize the benefits from lowered DH temperatures, and especially the DH supply temperature. Even though it is a common opinion that the primary return temperature has only a minor impact on the power-to-heat ratio, this is not always true when looking at the effect on an annual basis. If the freed capacity that occurs due to reduced mass-flow in the DH network when lowering the return temperature

is used to increase the number of connected buildings, the heat load will increase. This will then increase the annual operational hours for the CHP-station.

The model of the CHP station used in Paper VI is described by Genrup [52] and Truedsson [53]. The calculations of the CHP station are performed in IPSEpro®, which is an equation-based static simulation program for designing and evaluating different heat and power production units. The model used for these calculations is an evaluated off design model of a CHP station located in Enköping. The results are analyzed in Matlab®.

Impact of implementation of add-on-fan blowers and optimized space heating control on the district heating primary energy factor

To evaluate the benefits of lowered DH temperatures in terms of changed PEFs for DH, an analysis is made for two cases: 1) implementation of optimized control of space heating system; 2) implementation of add-on-fan blowers. The results of this study are presented in Paper IX. The methods used to lower DH temperatures are presented in Paper I, Paper VII and Paper VIII.

The main heat production in the DH network is based on a bio-fueled CHP station. A bio-fueled heat-only boiler is used as the peak load and base load heat-production plant. The case study of a DH system with a CHP station and heat-only boiler is described in Paper VI. The primary energy factor for the DH network is calculated according to the power bonus method, previously described in the section on the primary energy concept and district heating.

The PEFs for fuel input and electricity used in Paper IX are taken from the EN 15603:2008 standard [10], except for the Nordic electricity mixture which is calculated from statistics from [54].

Calculations of lowered DH temperatures are made for two different rotational speeds of the add-on-fan blowers, with electricity usage of 1% and 0.1% of the dimensioning heat output respectively. Two different operational modes for the add-on-fan blowers are implemented: 1) add-on-fan blowers in operation during the whole heating season; 2) add-on-fan blowers in operation only when reducing the PEF for DH.

The reduction of the PEF for DH is strongly dependent on the approach taken to the substituted electricity. For a marginal electricity approach, that is, replaced electricity produced in coal-fired power stations, the PEF for DH can be reduced by 25-40% by using add-on-fan blowers. This can be achieved provided that both the DH supply and the return temperatures are reduced and the additional electricity use by the add-on-fan blowers is not taken into account.

If the increased electricity consumption from the add-on-fan blowers is considered, the PEF for DH can still be reduced by 20-25% if the fans are only in operation when beneficial. This operating strategy becomes especially crucial in the case of high fan speeds, which consume more electricity.

If the substituted electricity is considered as the European electricity mixture, the impact of changes in PEF for DH is only a few percent. For a Nordic electricity mixture approach, the impact is even less.

If both the DH return temperature and the DH mass flow are lowered, the PEF for DH is not reduced to the same extent as in the previous case. However, still the DH PEF can be reduced by approximately 5% if the substituted electricity is considered as electricity produced in coal-fired condensing power stations. These results could be achieved either by using add-on-fan blowers at the higher rotational speed or by implementing optimized control of the space heating system.

RELIABILITY OF DISTRICT HEATING SUPPLY

The reliability of the heat supply with DH has been analyzed in previous studies and the conclusion is that in general DH can be considered as a reliable technology (see e.g. [55], [56] and [57]). However, failures do occur. In particular, blackouts in the electricity network seriously affect the functioning of DH and the delivery of heat aimed for space heating purposes, which is an issue in focus in this section. Almost all heating systems depend on electricity for proper operation. In a report by Werner and Sköldberg, the need to reduce the electricity dependence of DH was pointed out as an important issue for research [58].

Electricity dependence of district heating

The DH network needs electricity for pumps to distribute DH water and to maintain proper differential pressure in the network. In the DH-connected buildings electricity is used in circulation pumps to distribute the heat, and it is used in the control equipment in DH substations. An exception are space heating systems directly connected to the DH network, where the dependence on electricity can be limited. Jet pumps utilizing differential pressure in the DH network can be used in this case to distribute the heat without electricity [59]. This type of pump is still often used in Russia and Eastern Europe [60], but also Germany and Austria [61]. The concept is studied under Swedish conditions by Olsson [59].

In recent years, severe weather has increased the focus on secure energy supply, especially concerning space heating. In Sweden the Gudrun storm and the both large-scale and long-term power failure it caused, put this issue into focus. In the wake of the storm a number of studies have focused on the security of the electricity and heat supply (see Paper IV and [62] for a literature review).

During a large scale power failure, it is of great importance to secure building heat supply, especially during cold weather conditions. In Sweden, where the majority of the population live in buildings connected to DH, the electricity dependence of DH is an area of interest both in order to avoid evacuation of

inhabitants and to prevent costly freezing damage in buildings. In the project "DH in case of power failures"[2], the possibility for DH connected buildings to receive heat during power failures was in focus. Results from the project are presented in Paper II, Paper III and Paper IV. The author performed the project together with Patrick Lauenburg, and the results are also an important part of his PhD thesis [63].

The following section describes the operation of heating systems in buildings connected to DH in case of a power failure.

Natural circulation in hydronic heating systems during power failure

In case of a power failure, the electric circulation pump of the building's heating system will shut down, along with the whole control system. The space heating control valve in the DH substation will normally keep its current position, by contrast with the DHW control valve, which closes automatically in order to avoid scalding by overheated water.

Up to now, a common opinion has been that buildings connected to DH are totally dependent on electricity and that no heat will be distributed during a power failure due to the lack of circulation. However, old heating systems were often designed for natural circulation, which occurs because the water density varies with temperature. In such a system, the warm water will rise until it enters the radiators, and, after it has been cooled, it will sink. Systems designed for natural circulation have large pipe dimensions and low flow resistance in the boiler [64]. They are also most suitable in buildings with relatively small horizontal extension [64].

Modern heating systems are generally constructed with smaller pipe dimensions, which leads to increased fluid resistance compared with heating systems designed for natural circulation. In an earlier field study regarding the impact of DHW circulation [65], it was revealed that natural circulation was occurring at night, when the DHW circulation pump was turned off, in spite

[2] The project ran between 2007 and 2008 and resulted in a final report *Fjärrvärme vid elavbrott – Slutrapport (District heating in case of power failure - Final report)* [62]

of the relatively narrow pipe dimensions. This was due to unexpected heating of the circuit caused by a leaking control valve for DHW preparation. During the study within the project "DH in case of power failures", natural circulation was found in modern heating systems. This includes so-called one pipe systems with even larger flow resistance than modern two pipe systems. Natural circulation was also found in a heating system with a single DH substation supplying two multi-dwelling buildings where the heating water was delivered through long horizontal buried pipes.

The amount of heat received by the buildings differs depending on their construction date and the layout of the heating systems, but all the multi-dwelling buildings tested received between 40 and 90% of the original heat supply. (See Paper IV where results from the field study are presented.) Paper III describes real-time simulations of buildings connected to DH with and without pump operation. By using the model, power failures and corresponding indoor temperatures can be simulated at different outdoor temperatures and different DH supply temperatures. The real time simulations are performed with Simulink® software.

Even though the results shown in Paper IV indicate a good possibility for many buildings to receive heat during a power failure, obstacles to natural circulation do occur. In Paper II some of these are identified and discussed.

It is important to keep in mind that keeping the DH network in operation during a power failure is crucial for the connected buildings' ability to achieve heat. It is also essential that the heating system has a significant height difference in order to derive natural circulation, and that the DH substation is located low in the system.

An unconventional solution to maintain heat distribution within the building without electricity is to use a turbine-driven circulation pump. This approach is described by Frederiksen [66] and could be suitable for buildings where heat delivery is essential but where conditions for natural circulation are poor.

CONCLUDING DISCUSSION

Lowered district heating temperatures

In this thesis three methods to lower DH temperatures are in focus:

- Increased heat output from radiators equipped with add-on-fan blowers
- Optimized control of hydronic space heating system
- Alternative connection principles in DH substations

All these three methods are suitable for lower DH temperatures for existing building stock since only minor, or no, changes in the hydronic heating system are necessary.

The first strategy is to increase the convective heat output in an existing radiator system by installing add-on-fan blowers. Field studies (Paper VII) and simulations (Paper VIII) of radiators with such blowers show that the heat output can increase by up to 60% with minor additional electric input. If DH is produced in CHP stations, the increased electricity production gained by the use of add-on-fan blowers can exceed the additional electricity used by the add-on-fan blowers on an annual basis.

The second strategy, optimized control of the space heating temperature program, utilizes the fact that the temperature efficiency of heat exchangers varies with the heat load. By controlling both the flow and the supply temperature in the radiator system, an optimum operational point resulting in the lowest possible DH return temperature in the DH substation can be found at an arbitrary operating point. The resulting DH return temperature during such a control has been calculated in Paper I. Use of an optimized space heating control algorithm is especially beneficial for buildings with oversized space heating systems, which is more or less always the case when dealing with older buildings.

Finally, use of alternative, cascading connection principles in DH substations (Paper V) lowers the DH return temperature by several degrees on an annual basis. The comparison of DH return temperatures in Paper V is made between two cascading connection principles, a standard 2-stage connected DH substation and a parallel-connected one. DH companies often desire simple and robust DH substations. With this approach, the parallel-connected substation is gaining ground, especially in conventional buildings. The benefits from using alternative connection principles in DH substations are significant for buildings with low space heating temperature demand and for buildings with low DHW consumption.

District heating primary energy factors

The magnitude of primary energy factors for DH depends on the energy carriers used for heat production in the specific DH network. In the case of cogeneration of heat and electricity, the PEF for DH benefits from the electricity produced, especially if a marginal electricity approach is adopted. In Paper IX primary energy factors for a DH network with CHP are computed using the power bonus method described in European standard EN 15316-45 [12]. The calculations confirm that the PEF is strongly dependent on how the substituted electricity is considered in this case.

The DH system where electricity consuming add-on-fan blowers are used to lower the DH temperatures is a special case from the PEF point of view. If marginal electricity production based on a coal-fired condensing power station is used, the PEF for DH can decrease by over 20% on an annual basis. However, if the substituted electricity is considered as the EU mixture, the benefit is less than 10%, and if the Nordic electric mixture is considered, the benefit is limited to only a few percent.

A clear guideline to determine the PEF for DH with and without CHP stations is necessary if PEFs have to be used as an indicator when choosing a heating system. Such an indicator could be used by local DH companies as a driving force to reduce the primary energy consumption for the DH production as a competitive advantage. If primary energy factors are used, it is important to have public support and understanding of the concept. It must be also emphasized here that it is important to use the local PEF for DH, and not the national average value, since the PEFs differ strongly from one DH network to another.

Electricity dependence of district heating connected buildings

DH is usually considered as a robust heat source, even though the heat delivery relies on the electricity supply in the DH connected buildings. When the project "DH in case of power failures" started in 2007, there was great scepticism about the scope for delivering heat by means of natural circulation in modern space heating systems, separated from the DH network by heat exchangers. However, the study concluded from both the calculations and several field experiments that this is possible. This opens up the prospect of island operation in the local electricity grid, with CHP stations using the DH network as a heat sink in case of power failures.

It may seem that the add-on-fan blowers are in conflict with the ambition to reduce the electricity dependence of DH, as the add-on-fans clearly rely on electricity. However, the radiator will keep a large part of the heat output even when the fan is not in operation. Of course, the indoor temperature within the building will decrease in the event of a power failure, but the process will be slowed down. It is demonstrated that in many cases, the heat supply to buildings will be sufficient to avoid freezing in a hydronic heating system and delay the critical point when it will be necessary to evacuate the inhabitants.

Cavitation in control valves

The experimental study of cavitation in control valves showed that the valves were cavitating under conditions that are present in many Swedish DH networks. The cavitation occurs especially with high valve openings, high water temperature and large differential pressure. The absence of major problems with cavitation within the DH field can be explained by the prevalent use of oversized control valves.

Add-on-fan blower

The performance of the tested add-on-fan blower could be improved by design optimization. The main disadvantage of the blower is that the air flow is not evenly distributed along the radiator. The fans in the setup were regular fans used in computers for cooling. In this application a cross-flow or tangential fan would probably be more efficient.

The electricity demand of the add-on fan blowers is critical when it comes to the benefits for DH and electricity production. Consequently it would be of interest to reduce the electricity demand with locally produced electricity. Two simple principles could be considered: utilization of the photovoltaic effect or of the thermoelectric effect. Applications using the thermoelectric effect have already been applied in commercial products such as the Ecofan for distributing heat when using wood-fired stoves [67]. The available temperature difference is, of course, in that case higher, but a Czech company already has in its product catalogue an air-coil heater that operates based on this principle [68]. Another approach to reduce the electricity dependence is to use the photovoltaic effect. This approach is present in fan applications for forced ventilation of cars, caravans and boats. The disadvantage of the photovoltaic effect is the dependence on light for operation, if electricity storage is not installed.

The performance of the blower has only been tested for two simple types of radiators. It would be of interest to investigate how the additional airflow affects other types of common radiators, e.g., with convection panels.

To ensure the large market share for DH in the Swedish building stock, new marketing must be considered. Such a market is district cooling. It is not unrealistic to think that the customer demand for cooling will increase even within the residential buildings sector. However, to introduce a cooling system within the existing building stock may be too costly. If the existing heat distribution system could be used, not only for heating, but also for cooling,

the installation costs would be kept at a reduced level. In Sweden, there are at least two buildings where radiators are, or were, used for both space heating and cooling [69] and [70]. The add-on-blower application will increase the potential for cooling with already existing radiator systems. This could offer a great advantage in the competitive heat market.

Optimized space heating system operation

The optimized space heating control algorithm is at present in the final stage of development. Remotely controlled field experiments are being performed in Karlshamn, Sweden. An adaptive control algorithm is being used to find the optimum operating parameters independent of heat load and with respect to present DH supply temperature. The project is being performed at our division in Lund.

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APPENDIX 1

Domestic hot water consumption for apartments used in Paper V

The domestic hot water (DHW) flow patterns are simulated using a computer program developed by Wollerstrand, based on the statistical model for DHW usage presented by Arvastson and Wollerstrand (1997).

The simulation algorithm for DHW consumption is an improved model based on detailed field measurements performed in Sweden by S. Holmberg. Separate probability profiles are available for tapping occurrence, tapping size and tapping duration. Evenly distributed random numbers (Monte Carlo method) are used to simulate the tappings.

For calculations in Paper V the average consumption of DHW is 183 l/apartment and day for all building sizes. In Figures 1 and 2 below, the DHW consumption is plotted for different time periods and sizes of residential buildings. The flow patterns are presented for a time period of 24 hours, a week and a month (31 days) for four different residential building sizes (15, 30, 60 and 120 apartments).

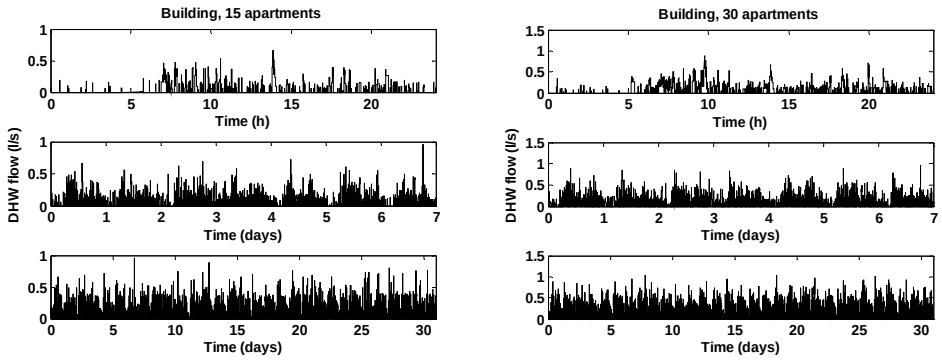

Figure 1. Domestic hot water consumption for building with 15 apartments (left) and 30 apartments (right)

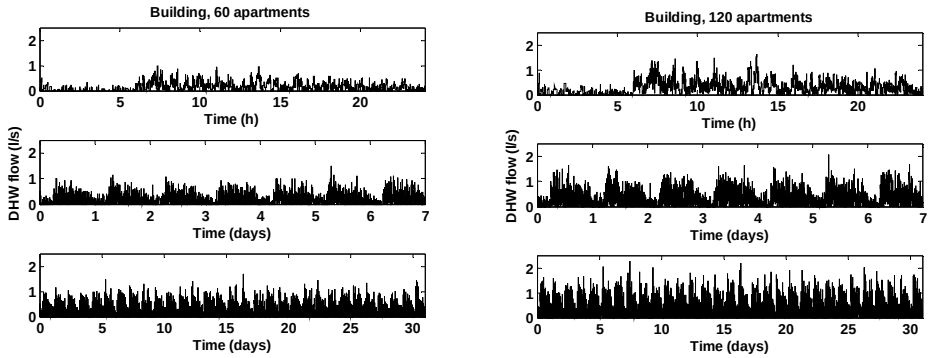

Figure 2. Domestic hot water consumption for building with 60 apartments (left) and 120 apartments (right)

Reference

Arvastson, L., Wollerstrand, J., *On sizing of domestic hot water heaters of instantaneous type*, 6th International Symposium on Automation of District Heating Systems, Reykjavik, Iceland, 1997

I

OPTIMIZED SPACE HEATING SYSTEM OPERATION WITH THE AIM OF LOWERING THE PRIMARY RETURN TEMPERATURE

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ABSTRACT

The paper presents results from a study aiming at minimizing the primary return temperature from a district heating (DH) substation by optimizing the control algorithm for a space heating system connected to the DH grid via a heat exchanger (HEX).

As shown in a previous study, an optimal (i.e. minimum) DH return temperature exists for each and every heat load. By varying the radiator flow and the radiator supply temperature this optimum can be found.

A space heating system is traditionally designed for a constant circulation flow rate combined with a suitable control curve for the space heating supply temperature as a function of the outdoor temperature. Optimal choice of the control curve varies from case to case and is an issue both we and others have dealt with in previous work. In the paper, theoretical control curves for optimal control of the space heating system in order to minimize the DH return temperature has been derived by calculating how supply temperature and circulation flow should be varied with the heat load. The estimated gain was found to vary strongly, depending on the actual conditions; however, assuming realistic conditions, it was seen that the gain can be as much as a reduction of 6°C of the DH return temperature from the radiator HEX on yearly average.

This paper presents theoretical knowledge and shows results supported with practical experiments. Based on these results a method for adaptive choice of parameter values for an optimized controller can be developed. One of the advantages with such an algorithm is that it will automatically adapt to changed conditions, e.g. variation in primary supply temperature.

Keywords: District heating, Space heating, Control System, Low Return Temperature

INTRODUCTION

The following work deals with the operation of hydronic space heating systems indirectly connected to district heating (DH), how to choose temperature program and circulation flow and how this affects the DH return temperature. A new method using a variable, load dependent flow rate will be described and analyzed.

Lowered return temperatures, or increased cooling, of DH water is always a relevant topic in DH technology. The efficiency of both heat production and heat distribution will be higher if the energy in the circulated DH water is used to an extent as large as possible. Many studies have shown that the return temperature mainly is determined by the parameters in the house-internal systems, for example supply and return temperatures and circulation flow in the space heating system.

Traditionally, hydronic heating systems work with a (more or less) constant flow rate. However, in previous work [3], [4] and [6] it is shown that by varying the flow rate, depending on the heat load, it is possible to lower the DH return temperature.

OPERATION OF THE RADIATOR SYSTEM IN GENERAL

Space heating systems dealt with in this work is conventional 2-pipe radiator systems, connected to a DH substation, see Figure 1. Note that we exclusively deal with so-called indirect connections, i.e. a heat exchanger (HEX) hydraulically separates the DH and radiator systems, which is the predominant solution in Sweden.

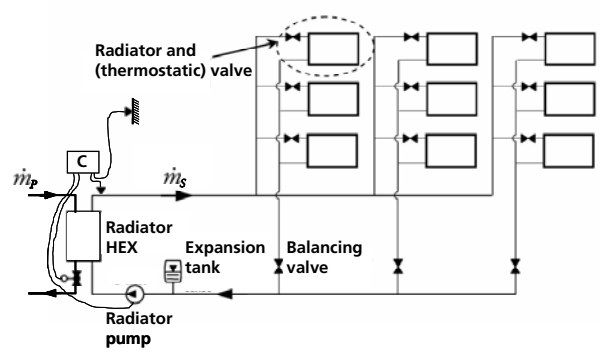

Figure 1 Schematic picture of a 2-pipe radiator system in a building connected to DH.

Traditionally, the heat output is controlled by varying the supply temperature to the radiators according to a control curve based on the outdoor temperature, the building's heat losses and the radiators' thermal characteristics. The circulation flow is not regulated in the substation and although it might be throttled by thermostatic valves in radiators, the flow can be considered to be constant. Frederiksen and Wollerstrand already in 1987 [3]

showed that in every radiator system, for every heat load and DH supply temperature, there is a flow rate and a supply temperature that gives the lowest DH return temperature.

Regarding the temperatures in the radiator system, there have been different recommendations. Today, generally lower secondary temperatures are used (60/45, 60/40 and 55/45°C), while traditionally higher temperatures (e.g. 80/60°C) has been prevailing.

Radiator systems are almost always oversized, often up to 100 per cent or more, see for example [5], [7] and [8]. What is the consequence of an oversized radiator system? If no actions are taken, the indoor temperature will be too high. A system with well-functioning thermostatic valves can, at least in theory, throttle flows and achieve correct indoor temperature. Other possibilities are that the tenants by themselves will close the radiator valves, complain to the caretaker – who can change the secondary temperature program – or simply open a window.

Figure 2 shows three different graphs. Each one shows a traditional 80/60°C system (red lines) together with a modified temperature program (orange lines) to compensate for an oversizing of 100 per cent of the system (HEX, radiators and flow rate). The vertical line in each graph indicates the design outdoor temperature, T_{DOT}, in this case −15°C. An oversized system can be considered as a correctly designed system that operates at a wider outdoor temperature range than the actual, by moving T_{DOT} to the left. According to this reasoning, the secondary temperature program can be converted to the new temperature range (which is doubled in the case with 100 per cent oversizing). Assuming that heating is needed up to 17°, the original temperature range is 17 − (−15) = 32°C and the 100 per cent oversized range becomes 2·32 = 64°C and the new T_{DOT} = 17 − 64 = −47°C. In other words, the system is able to achieve the desired indoor temperature at −47°C.

Having 80/60°C at −47°C, we get 55/45°C at the real T_{DOT} = −15°C. This temperature program is shown in the first graph and corresponds to the case when the supply temperature is lowered, without any other action taken. In the second graph, the idea is to keep the temperature drop of 20°C at T_{DOT}. The flow has therefore been adjusted (the original pump is oversized) and we get a 60/40°C system instead. This alternative gives a lower return temperature and reduced risk of noise in the system. Finally, the third graph shows what is usually termed a low-flow system. The method got some attention in the 70s, see for example [1]. Theoretically, this would be the case if perfectly functioning thermostatic valves were used. In practice, the system has to be balanced hydraulically to avoid imbalance and overheating of parts of the building.

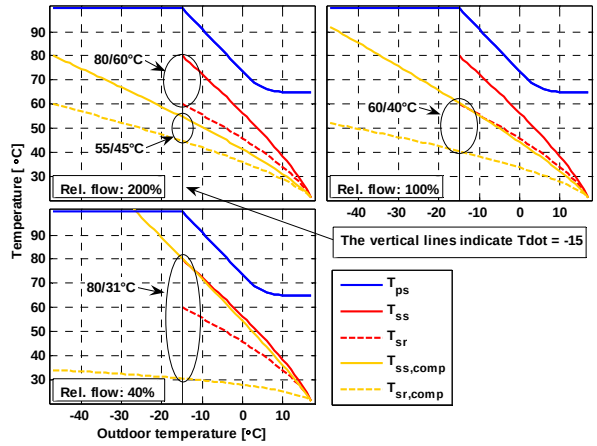

Figure 2 Comparison between secondary temperatures with no oversizing (red lines) and with 100 per cent oversizing (orange lines), compensated in three different ways.

The advantages with such a low circulation flow rate are that the pressure drop will be relatively small, and that all radiators will have approximately the same pressure differential, which simplifies design and balancing of the radiators [2]. Further, there is a reduced risk of noise and less pumping energy is required. Thermostatic valves will have larger authority and there will no longer be any need for balancing valves on individual risers.

The high secondary supply temperature used at low circulation flow makes the system more sensitive to changes in the primary supply temperature. If the primary supply temperature is decreased, the control valve opens and increases the primary flow through the HEX. In such case, the primary return temperature will increase.

OPTIMIZATION OF THE RADIATOR PROGRAM – THEORY

How is the primary return temperature, T_{pr}, affected by a changed radiator flow rate and supply temperature? The heat flow from the primary water, via the secondary water to the indoor air can be described by:

$$\dot{Q} = \dot{m}_p \cdot c_p \cdot \left(T_{ps} - T_{pr}\right) = \dot{m}_s \cdot c_p \cdot \left(T_{ss} - T_{sr}\right) = \\ = K_{rad} \cdot LMTD_{rad}^{m} \tag{1}$$

The logarithmic temperature difference, $LMTD_{rad}$, between the radiator surface and the indoor air is defined as:

$$LMTD_{rad} = \frac{\left(T_{ss} - T_{sr}\right)}{\ln\left(\dfrac{T_{ss} - T_i}{T_{sr} - T_i}\right)} \tag{2}$$

The heat transfer capacity from the radiator surface is practically constant, i.e. it is not affected by the magnitude of the circulation flow through the radiator. This is due to the fact that the heat transfer resistance between the water in the radiator and the inside of the radiator wall is at least ten times lower than the resistance between the radiator surface and the surrounding air. However, a reduced flow means that every unit of water will be in the radiator for a longer time, and hence the cooling of the water will be larger. The largest possible theoretical temperature difference is limited by the primary supply (T_{ps}) and indoor air (T_i) temperature.

In contrast to the radiator, the performance of the radiator system HEX is dependent on variations in the flow. This is a water-to-water HEX where the primary and secondary heat transfer coefficients are of similar magnitude. A lower flow has a negative influence on the heat transfer which results in a reduced cooling of the primary water. How could a lower radiator flow still be of interest?

The transferred heat in the radiator HEX is given by the following equation:

$$\dot{Q} = k \cdot A \cdot LMTD \tag{3}$$

The overall heat transfer coefficient is defined as the sum of all the resistances from the primary to the secondary side of the HEX. The most significant resistances are the convective (between water and HEX wall material). The conductive resistance in the wall material can generally be neglected.

Because of the relatively moderate temperature variations in a DH substation the convective heat transfer coefficient can be assumed to be a function of the flow only, according to: (cf. [9])

$$\alpha = c_0 \dot{m}^n \tag{4}$$

Using (4) the overall heat transfer coefficient, k, can be written as:

$$k = \left(\frac{1}{\alpha_p} + \frac{1}{\alpha_s} \right)^{-1} = c_0 \left(\frac{1}{\dot{m}_p^n} + \frac{1}{\dot{m}_s^n} \right)^{-1} \tag{5}$$

The last substitution assumes symmetrical flow channels on primary and secondary sides of the HEX. The equation tells us that a lower flow means lowered heat transfer capacity. However, it is possible to show that, if one flow is significantly larger, the impact of a changed magnitude of the lower flow is reduced.

Consider a typical symmetrical plate HEX in a DH substation. In a high flow system, e.g. an 80/60°C system oversized 100 per cent and operated as a 55/45°C system, the temperature drop at DOT is 10°C, while the primary side temperature drop can be 50°C or more. The relation between the primary and the secondary flow is then inversely proportional to the relation between the respective temperature drops. Now, compare the influence on the overall heat transfer coefficient using 10 and 20°C (half the flow) temperature drop, respectively:

10°C: $k \approx 1 / (50^{0.7} + 10^{0.7}) = 0.049$
20ºC: $k \approx 1 / (50^{0.7} + 20^{0.7}) = 0.042$

Consequently, halving the flow only gives a moderate decrease (14%) of the heat transfer capacity. It is not until we get approximately the same flow on each side that the decrease in heat transfer capacity becomes considerable. On the other hand, a decreased flow results in temperature changes in the radiator system.

We must therefore study the parameters' total influence on the heat transfer, which most easily is done numerically.

In the first graph in Figure 3, it can be seen that the overall heat transfer coefficient, k, in the HEX is continuously decreases with an increased temperature drop in the radiator system. This is caused by a decreasing heat transfer coefficient on the secondary side of the HEX due to the smaller flow. At the same time, the *LMTD* increases. Note that the primary flow is varied in order to keep the level of transferred heat constant which means that the product $k \cdot LMTD$ is constant.

When the secondary flow rate decreases, the primary return temperature begins to decrease until it reaches a minimum level. After the optimum, it starts to increase again. The minimum primary return temperature is achieved at a secondary temperature drop of 40-45°C.

This HEX has been designed for a secondary flow significantly larger than the required primary flow. Because of this, and the fact that the HEX is symmetrical, the heat transfer on the primary side is relatively small. If we know that the radiator system will work with a high temperature drop, the HEX can be designed differently. The next graph shows the situation when the HEX has been made narrower but longer, with the same heat transfer area, to better suit the new flow rates. This can for example be made by a so-called two-pass HEX, i.e. a long HEX "folded" in the middle.

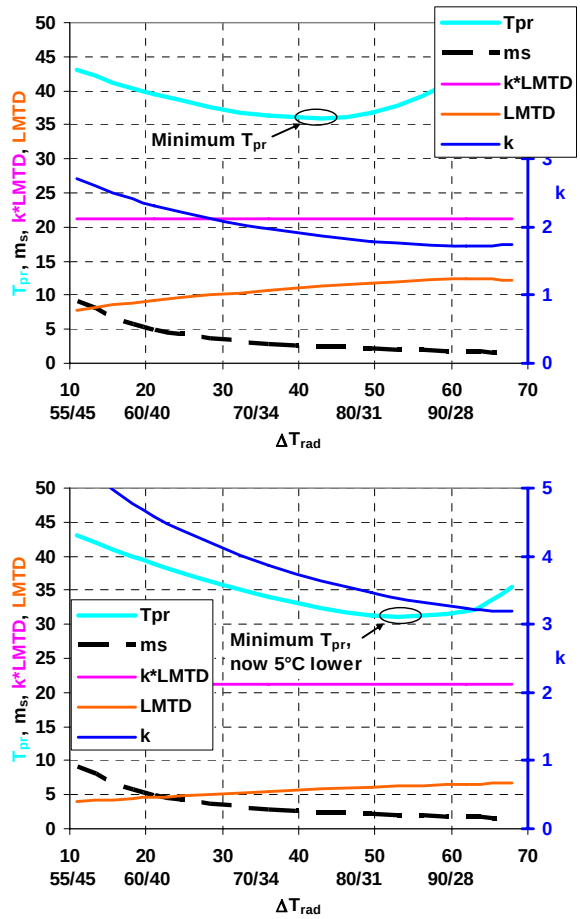

Figure 3 The graphs show the primary return temperature for different secondary flow rates and corresponding temperature programs (for constant heat supply). The upper graph shows a regular HEX and the lower graph shows a HEX which is made longer (constant heat transfer area).

The overall heat transfer coefficient is now significantly increased and the primary return temperature has been decreased several degrees. Its minimum is now 5°C less and has moved to a secondary temperature drop of 50-55°C (se the lower graph in figure 3).

Optimized temperature program

The design conditions for the HEX are normally given at full load, T_{DOT}. However, space heating systems are almost always working at part load, which makes the HEX oversized from a design point of view. This means that the highest possible cooling of primary water is achieved by increasing the radiator supply temperature while the circulation flow rate is reduced; see the dashed lines in Figure 4 showing the optimal temperatures when the radiator flow is allowed vary. As a reference, the grey line shows the primary return temperature for a high flow system (55/45°C).

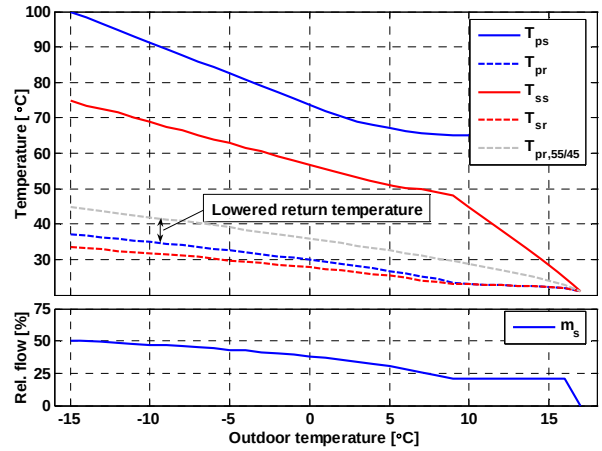

Figure 4 A 100 per cent oversized hydronic heating system compensated by an optimized temperature program (variable flow). The primary return temperature for a high flow system (55/45°C) is shown for comparison (grey lines).

To be able to quantify the comparison between the different ways to operate the space heating system, one must take the duration of the outdoor temperature into account. The resulting weighted, primary return temperature for the different cases is shown in Table 1. For each case the flow rate relative to the reference case (0 per cent oversizing, standard 80/60°C program, bold figures) is shown. Above the horizontal grey line there are two cases with no oversizing of the heating system, designed for 80/60°C. Under the line there are four different cases where the system is 100 per cent oversized, originally also designed for 80/60°C. In these cases some kind of adjustment has been made in order to compensate for the oversizing and achieve correct indoor temperature.

The first column in the table indicates the oversizing of the system, the second column describes the type of radiator program. The third column represents the relative secondary flow rate, compared to the reference case. In the fourth column the reduction in the primary return temperature in °C, compared to the reference case, is shown.

With a low secondary flow rate or with optimized flow the original HEX operates with a flow substantially lower than the design flow. As explained in previous section, all radiator programs that involve a reduced flow rate gain primary cooling with a prolonged HEX. In column five the reduction in the primary return temperature for a 100 per cent prolonged HEX is given, compared to the reference value.

Table 1 Difference in yearly weighted primary return temperature from the space heating HEX relative the reference case (bold) [°C]. For each temperature case the magnitude of the circulation flow relative the reference case (= 1) is given. Oversizing means that both radiators and the space heating HEX are oversized.

Over-sizing [%]	Type of radiator program	Relative flow	Return temperature reduction with HEX:	
			Original	Prolonged 100 %
0	standard, 80/60°C	**1**	**44.9**	−0.8
0	optimized	(0.2-1)	−1.8	−5.8
100	55/45°C (high flow)	2	−10.2	−10.3
100	60/40°C (normal flow)	1	−12.2	−12.3
100	80/32°C (low flow)	0.4	−13.4	−14.6
100	optimized	(0.2-0.5)	−16.1	−18.1

Four major conclusions can be drawn from the table:

- An oversizing in itself can give a large gain in primary cooling provided that some kind of compensation is made for the system to work properly, i.e. correct indoor temperature
- By optimizing the system (by a varying flow rate) the primary return temperature can be further decreased
- An extension of the HEX is further reducing the primary return temperature in systems with a low secondary flow rate
- Regardless of the degree of oversizing, the combination of an optimized program and a prolonged HEX gives a substantial reduction in the primary return temperature

The lowest acceptable level of the circulation flow rate during the optimization of the radiator circuit was set to 20 per cent of the original flow. There is no specific reason for this level and it might in practice be suitable with another level. To investigate how sensitive the results are with respect to this level, the calculations were performed using 40 per cent instead (corresponding to a low flow system). With no oversizing, the result was −1.6°C (compared with −1.8°C) and with 100 per cent oversizing −15.6°C (compared to −16.1°C). The influence on the yearly average is consequently rather small when changing the level for the minimum flow from 20 to 40 per cent.

FIELD STUDY

To investigate theories in practice, field studies has been performed in a multi-dwelling building containing 20 flats. The radiator system in the building is oversized and the original temperature program, 80/60°C, is adjusted to approximately a 55/45°C program.

The aim with the field experiments, apart from verifying the theories, is to develop a method for finding the minimum primary return temperature. The method can later on be used for developing an adaptive control algorithm.

The experiments are performed on a conventional DH substation consisting of a domestic hot water circuit and a radiator circuit.

A reduced radiator circulation flow rate means a reduced heat supply. To prevent this, the experiment starts with locking the control valve, which regulates the heat supply to the radiator circuit, in its current position. By locking the primary control valve the primary flow rate can be considered as constant. Therefore, the transferred heat only depends on changes in the primary return temperature, which are small compared to the total temperature drop. As shown in the previous section, the return temperature is depending on the secondary circulation flow rate.

Obviously, both primary differential pressure as well as primary supply temperature can vary, even if these variations are quite slow. If needed, the primary flow can be adjusted to compensate for a changed differential pressure. In case of large variations of the primary differential pressure and/or supply temperature, the test will have to be restarted later.

Another problem is how to measure the heat supply to the radiator circuit, if there is no secondary energy meter (which normally is the case). The method used here was to temporarily close the control valve to the domestic hot water circuit. Then, the primary flow rate to the radiator HEX is registered on the flow meter of the primary heat meter belonging to the substation. With temperature measurements on supply and return, primary and secondary side, both heat supply and secondary circulation flow rate can be calculated from the energy balance:

$$\dot{Q}_{rad} = \dot{m}_p \cdot c_p \cdot \left(T_{ps} - T_{pr}\right) = \dot{m}_s \cdot c_p \cdot \left(T_{ss} - T_{sr}\right) \qquad (6)$$

This procedure is repeated at regular intervals during the test.

Test 1

During Test 1, the outdoor temperature was about 7.5°C, the heat load 33 kW and the primary supply temperature varied from 87 to 90°C. The secondary flow was adjusted using a shut-off valve

located after the circulation pump. The result from the test is shown in Figure 5.

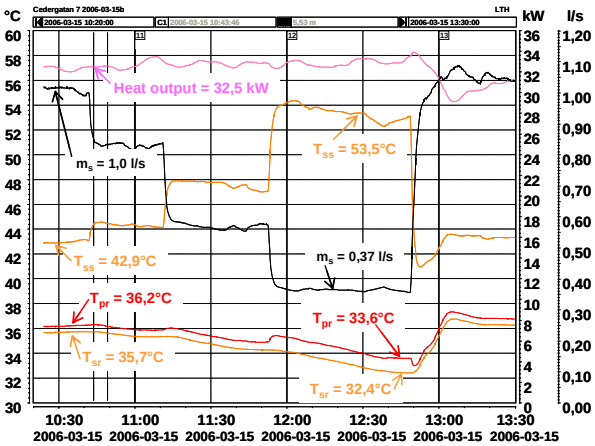

Figure 5 Test 1. Decrease of the primary return temperature at reduced circulation flow rate.

The circulation flow was decreased in three steps, beginning at 10:40. Each flow change results in a sudden increase of the outgoing, secondary temperature. The secondary return temperature remains constant for a couple of minutes, due to transportation times, after which it begins to slowly decrease since the temperature drop in the radiators increases. At first, the primary return temperature increases a little, and then starts to decrease along with the secondary return temperature. The small increase depends on poor heat transfer in the HEX, which sets in immediately, while the following decrease depends on the increased secondary temperature difference, which changes more slowly depending on thermal inertia and transportation times in the radiator system.

The difference between the primary and the secondary return temperature gets bigger when the circulation flow becomes lower, because of the deteriorated heat transfer.

The circulation flow was reduced by 17, 43 and 64 per cent during the test and resulted in a lowered primary return temperature; 2.5°C at the most, see Table 2.

Table 2 Results from Test 1. Decrease of the primary return temperature with lowered circulation flow rate.

m_{rad} [%]	T_{ss}	T_{sr}	ΔT_s	T_{pr}	ΔT_{pr}	Time to steady state
100	43.1	35.7	7.4	36.2	**0**	-
83	44.1	35.3	8.8	35.8	**−0.4**	~30
57	47.0	34.2	12.8	34.9	**−1.3**	~45
36	52.8	32.4	20.4	33.6	**−2.6**	~60

The last column indicates the time it takes to achieve stable conditions in the system. As seen, it takes some time to find an optimum, especially when the flow rate is low. This means that there is a risk for significant variations in primary differential pressure and supply temperature.

Test 2

In the second test, the idea was to speed up the optimization by utilizing the fact that the secondary return temperature will decrease if the flow is decreased in a circuit with a constant heat input. Taking this into account, one does not have to await a stationary secondary return temperature, but only long enough to see if the primary return temperature is decreasing. If so, the secondary flow can be further decreased.

The outdoor temperature during Test 2 was 7.3 ± 0.2°C, the heat supply was 30.7 kW and the primary supply temperature was 83.9 ± 0.1°C. The flow was reduced by variation of the rotation speed on the front panel of the pump. Some additional throttling using a shut-off valve was required to achieve a flow that was small enough. The result of the test is shown in Figure 6.

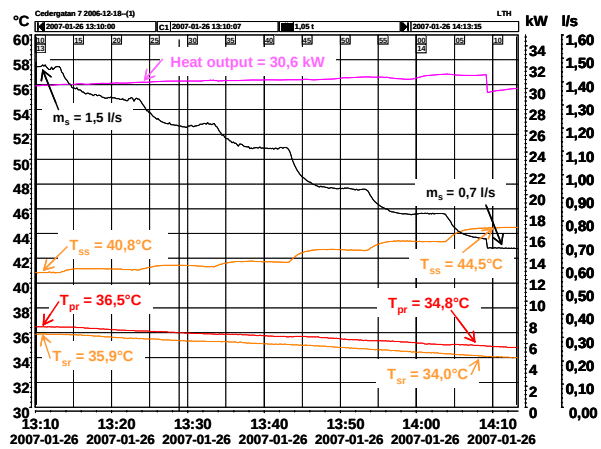

Figure 6 Test 2. Decrease of the primary return temperature when the circulation flow is reduced in shorter time intervals.

The most obvious difference compared with Test 1 is that both the primary and secondary return temperature decrease continuously. A definite determination of the lowest primary return temperature will require the radiator circuit to stabilize. If the primary return temperature should increase after a flow reduction, the optimal operational point has been missed and the flow needs to be increased again.

DISCUSSION AND FUTURE WORK

The presented results show good possibilities to improve the cooling of primary water by optimizing

the radiator circulation flow and supply temperature for all heat loads.

For a practical application, the rotation speed of the pump and the supply temperature set-point can be adjusted on-line by a modified algorithm in the radiator controller.

An adaptive control algorithm also gives the possibility to identify a changed optimum. For example, a lowered primary supply temperature (which is very beneficial for the DH system if possible) can have a negative influence on the primary return temperature in a low flow radiator system. In such a case, it is desirable if the controller can identify the new optimum. Other factors affecting the most efficient operation of the system is for example fouling of the HEX or changed heat losses from the building.

To be able to establish a concrete application, more field tests in buildings with various degrees of oversizing, and at different outdoor temperatures, needs to be performed.

Since automation is used in most buildings today, more sophisticated control is possible to implement.

Another important conclusion is that the potential for the reduction in the primary return temperature can be even larger if the HEX is prolonged. When replacing or making a new installation of a HEX this could be taken into account.

ACKNOWLEDGEMENTS

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NOMENCLATURE

Abbreviations

DOT	Design outdoor temperature
DH	District heating
HEX	HEX

Variables

A	Area, m^2
α	Heat transfer coefficient, W/m^2K
c_0	Constant
c_p	Heat capacity, J/kgK
LMTD	Logarithmic mean temperature,°C
k	Overall heat transfer coefficient, W/m^2K
K	Heat transfer constant, W/K
$\dot{m}$	Mass flow, kg/s
m	Radiator exponent, -, also mass flow
n	HEX exponent, -
$\dot{Q}$	Heat flow, kW
T	Temperature,°C

Subscripts

0	Design condition
i	Indoor
p	Primary (side)
rad	Radiator (system)
s	Secondary (side)

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II

OBSTACLES FOR NATURAL CIRCULATION IN HEATING SYSTEMS, CONNECTED TO DISTRICT HEATING VIA HEAT EXCHANGERS, DURING A POWER FAILURE

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ABSTRACT

The largest threat during a power failure of long duration is interrupted space heating, both in residential buildings and within sensitive public activities. Almost all heating systems are dependent on electricity, not only electric heating systems. Thus, hydronic heating systems need electricity for pump operation, control equipment, burners etc. Since district heating (DH) is a market-dominating type of heating method in Scandinavia (with almost 90 per cent of the heat market for multi-dwelling buildings in Sweden) it is of great importance to investigate the potential for buildings connected to DH to receive heat during an electric power failure.

In Sweden, heat from the DH water is transferred to the space heating radiator system via a heat exchanger (HEX). During a power failure the pump within the building, used for water circulation in the radiator system, as well as the control valve for regulating the DH flow, will be out of order.

In previous papers, results from field studies and simulations have shown that a significant amount of the original heat load in a building remains during a power failure. The phenomenon that makes this possible is natural circulation, a mechanism that was used for circulation in old radiator systems.

A modern heating system has several obstacles for natural circulation to arise. In this paper, problems with very high and/or long buildings, 1-pipe systems, by-pass connections, and reversed HEXs (installed upside-down) are analysed.

Keywords: District heating, Power failure, Space heating system, Natural circulation, Field study

INTRODUCTION

During a large scale power failure a lot of important functions in a modern society are affected. Heat supply is one important function during severe weather conditions. Almost all heating systems, not only electric heating systems, are dependent on electricity in order to operate. Electricity is used for control equipment and distribution of water in the heating system. These devices will of course not be able to work during a power failure.

District heating (DH) is the dominating heating system in Scandinavia, covering about 50% of the total heat market in Sweden (86% for multi-dwelling buildings, 70% for non residential buildings and 10% for one- and two dwelling buildings) [4].

Buildings connected to the DH grid in Sweden are connected via a heat exchanger (HEX), so-called indirect connection. This paper will also solely be focusing on indirect DH connections and no conclusions regarding the upholding of heat supplies during power failures with direct connections will therefore be made. However, in many countries direct connections are common ([5]) and to comment upon this fact, one must keep in mind that the use of HEXs is a definite obstacle for the heat transfer between the DH grid and the secondary systems when no electricity is available. In this sense, the paper deals with the “worst case” and direct connections are left to future work.

In an indirectly connected building, electricity is used for controlling the heat output to the building by regulating the DH flow through the HEX and for pumps for distributing the water in the secondary system to the radiators. If the power supply fails, the circulation pumps will stop and the control valve for the radiator system will stop in its current position, allowing for the same DH flow to pass through the HEX, provided that the DH grid still is in operation.

In very old radiator systems the distribution was not dependent on a pump, but on the density differences between hot and cold water and the height of the system, so-called natural circulation. Characteristic of these systems is small flow resistance and a high secondary supply temperature to the radiator system. In modern heating systems, designed for pump operation, there are many components causing additional pressure losses such as HEX, balancing valves, much smaller pipe dimensions etc. Nevertheless, as demonstrated in previous papers ([1], [2]) the potential for natural circulation in a modern hydronic heating system connected to DH is not negligible. The potential for natural circulation depends not only on the type of heating system and its components, but also on the height and the length of the building as will be shown in the paper.

The heat production in a DH grid is often incorporated in a combined heat and power, CHP, station, which means that power and heat is generated in the same unit. In a CHP station the DH grid is used as a heat sink for the power production. Many CHP stations have no other available heat sink.

To be able to establish island operation, which means local production and utilization of electricity during a large scale power failure on the national

grid, the CHP station will still need cooling for its operation. To fulfil this demand during island operation, when only limited supplies of electricity are available, the buildings connected to the DH grid must be able to receive heat.

Apart from giving possibilities for island operation, the society as a whole can benefit if evacuations of citizens can be avoided or delayed during a long power failure during harsh weather conditions.

Objective of the study

In areas were DH has a large market share, the possibility to uphold heat supplies during a power failure of long duration is of great value for the society. The overall objective with the project is to secure heat supplies during a large scale power failure. The objective of this paper is to investigate the obstacles for natural circulation to arise in space heating systems, originally designed for pump circulation, without electric power.

HEATING SYSTEMS CONNECTED TO DH

In a DH connected building, the DH is used for provision of domestic hot water, and space heating. Space heat can be distributed by a hydronic radiator system and, especially in public buildings, via a forced ventilation system. Heat is transferred from the DH grid (primary side) to the heating systems (secondary side) in the DH substation. In this work we have been focusing on so-called indirect DH connection which is the prevalent technique in Sweden. This means that primary and secondary systems are hydraulically separated by HEXs. Fig. 1 below shows an overview of a simple substation with a domestic hot water (DHW) circuit and a radiator (Rad) circuit.

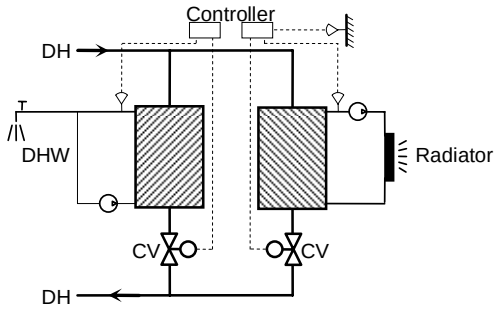

Fig. 1 Overview of a DH substation with a DHW and radiator circuit.

The DHW circuit in the picture is of the instantaneous type, which does not include a storage tank. However, this is just an example of a substation layout and the DHW system will not have any influence on the natural circulation in the heating system.

The substation in the picture is a so-called parallel coupled substation. Another very common design is two-stage connection, where cascading of the HEXs is employed to lower the primary return temperature. If the building has a ventilation system with forced air, heat is required to the air handling units. There are various ways to connect these units. One is to have an additional circuit connected in parallel to the DHW and radiator circuits, see Fig. 2.

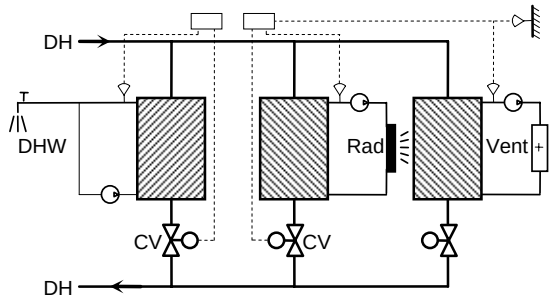

Fig. 2 Substation with a ventilation circuit connected in parallel with DHW and radiator circuits.

Another common variant is to combine the hot water to the radiators and air handling units in one circuit, often termed hot water (HW). Because the radiators and air handling units have different demands regarding temperature levels, by-pass connections are used, both for the radiator and ventilation circuit. The by-pass allows the secondary circuits to operate at individually lower temperature and pressure levels. Fig. 3 below shows a HW circuit with a common type of by-pass connection.

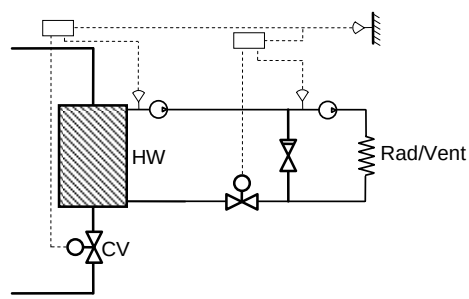

Fig. 3 HW circuit with a by-pass connection.

In most buildings the radiator system represents the largest part of the space heating supply, and in almost all residential buildings for the whole space heating supply. We will therefore mainly be focusing on the radiator system. Besides, all fans operating in the ventilation system will stop without power. Although, the by-pass connections are an essential part in the system and will be dealt with further on.

Fig. 4 below shows a simplified picture of a building with a space heating system connected to DH. DHW and possible forced ventilation have been omitted.

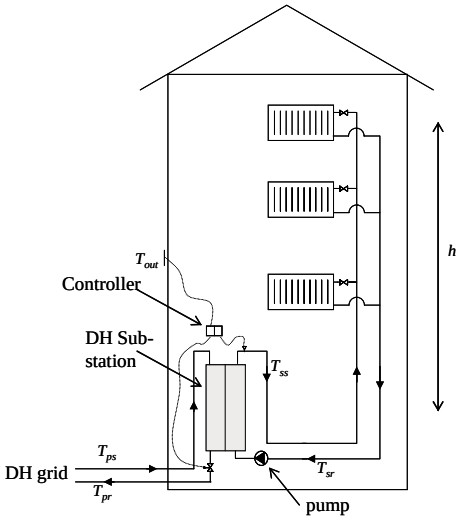

Fig. 4 Building with a hydronic radiator system connected to DH.

The heat is transferred from the HEX via a piping system to the radiators. Normally, a 2-pipe system is used, i.e. one supply and one return pipe, which means all radiators are connected in parallel. However, during the 60s and 70s, when a large number of residential buildings were built in Sweden, a lot of systems were built with a 1-pipe system, where the radiators in one ore more flats are connected in series.

Traditional control of heat supply is based on variation of the supply temperature to the radiators, depending on the outdoor temperature. The supply temperature set point is adjusted to compensate for the buildings' thermal inertia. The requested supply temperature is obtained by regulating the DH flow through the HEX using a control valve, typically manoeuvred by an electric actuator.

Assuming that the DH grid is still in operation during a power failure, the heat supply itself is not a problem (as would be the case with electrical heating, a heat pump or a boiler). However, all pumps, fans and control equipment need electricity to be able to distribute the heat in the building. The circulation pump in the system will stop. When the control equipment fails, the actuators will not receive any control signals and will in general stop in its current position. However, actuators in tap water systems are often equipped with a spring-return mechanism to avoid the risk of scalding from extremely hot tap water. Dampers in air handlers are generally equipped with a spring-return mechanism to avoid freezing. In space heating systems, however, spring-return is not used because it involves a higher cost and is not required. If the DH grid is in operation, this means that there will still be a flow through the HEX. The question is whether any heat can be transferred into the building.

NATURAL CIRCULATION

This section will describe the mechanism for natural circulation and present results from field studies and computer calculations.

In parallel to field studies a theoretical computer model, of modern buildings operating with natural circulation during a power failure, has been developed. Results from computer simulations, theoretical relations and resemblance between simulated results and field studies are presented in detail in previous papers ([1] and [2]).

Mechanisms

Natural circulation is a well-known phenomenon, which occurs due to the density difference between hot and cold water. Very old heating systems were designed for natural circulation with low flow resistance in pipes and in the heating source. Fig. 5 shows a heating system designed for natural circulation (to the left in the picture) and a system designed for pump circulation (to the right).

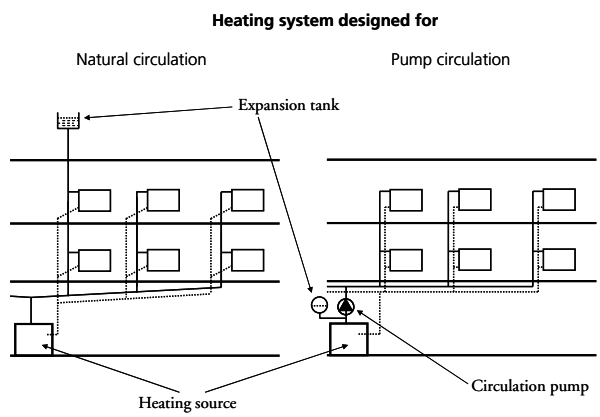

Fig. 5 Hydronic radiator system designed for natural circulation (left) (modified after [3]) and pump operation (right)

Note that there is a slight inclination of the horizontal pipes in the system built for natural circulation. This helps to distribute heat in horizontal direction.

The differential pressure that causes natural circulation is, as mentioned before, dependent on the density difference between secondary supply and return temperature, and the height of the system, see Eq.(1):

$$\Delta p = g \cdot h \cdot (\rho(T_{ss}) - \rho(T_{sr})) \qquad (1)$$

A modern heating system has, as already mentioned, additional pressure losses. However, Eq.(1) is still applicable. If DH water is passing through the HEX during a power failure, the temperature in the HEX will be substantially increased on the secondary side, since the circulation pump no longer is in operation. The

increased temperature leads to an increased temperature difference in the secondary system. In combination with the height of the substation a differential pressure will arise, possible to cause a circulation. Fig. 6 below shows the result from a field experiment. The upper diagram shows temperature levels in the HEX, blue lines indicate primary side and red lines indicate secondary side. Solid lines are used for supply and dashed lines are used for return temperatures. The diagram in the middle shows the outdoor temperature and the lower diagram shows relative heat output (red) and secondary circulation flow (blue). The "power failure" begins at 9:40.

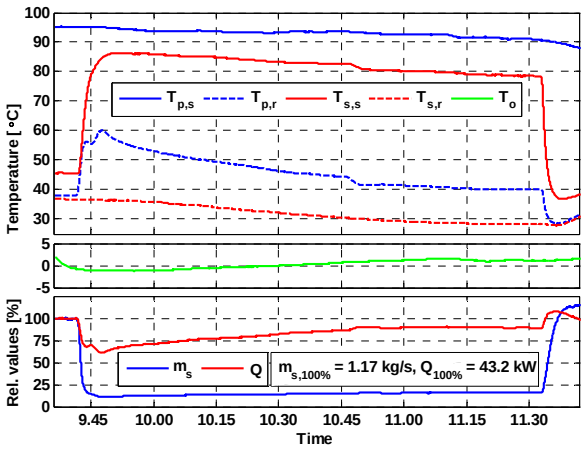

Fig. 6 Field experiment with natural circulation.

When the pump stops, the flow decreases causing the secondary supply temperature to rise to almost the same level as the incoming DH water. Since the secondary temperature difference is large, the heat output is about 90% even though the secondary mass flow is reduced with more than 80%.

The secondary temperature difference is of great importance to the potential for natural circulation. Therefore, it is possible for the DH company to increase the primary supply temperature in the DH grid and thereby increase the heat output. The influence of a changed primary supply temperature is discussed in [1].

In total, 14 different objects have been tested for natural circulation, including one-, two- and multi-dwelling, schools and nursing homes. Almost all buildings will receive a considerable amount of heat by means of natural circulation. However, as seen in Fig. 7, the relative heat output varies quite a lot between the objects, depending on the conditions in each system.

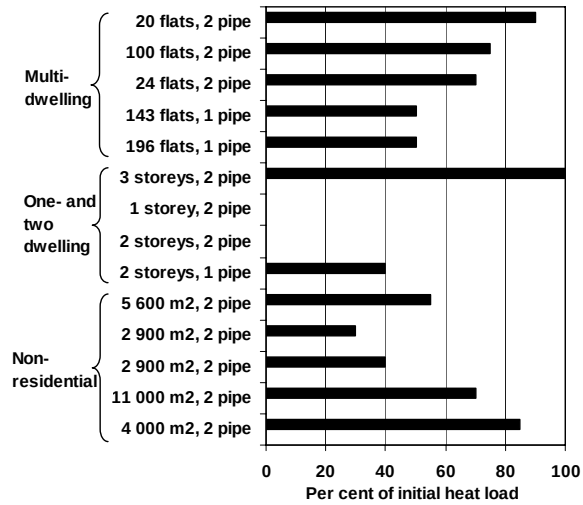

Fig. 7 Buildings tested for power failure. The column shows the heat supply (in relation to the initial) that was achieved during the test.

Limitations

There are several limitations for the potential for natural circulation in a modern heating system.

A major difference between pump circulation and natural circulation in a modern heating system is that in case of natural circulation the differential pressure from the circulation pump is substituted by several pressure differentials from virtual pumps, each based on the temperature difference at its location in the heating system, see Fig. 8.

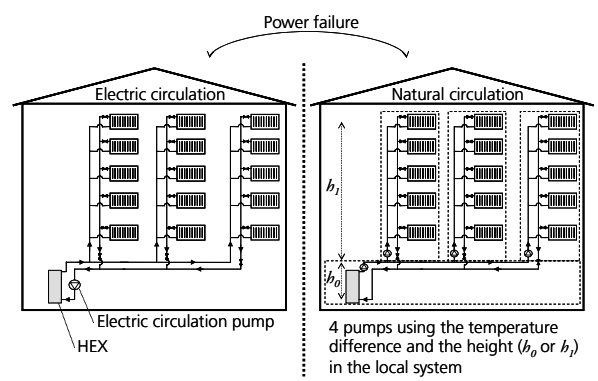

Fig. 8 Schematic picture of pressure differentials in the system with pump circulation (left) and natural circulation (right).

In the figure the circulation pump is substituted by one virtual pump in the horizontal distribution, based on the temperature difference of the incoming and outgoing secondary water and the height of the horizontal system (h_0), which basically is the height of the substation [6]. A virtual pump is also found in each riser, provided that there is a temperature difference between supply and return pipe. The

height of the risers (h_1) determines the differential pressure. Naturally, these virtual pumps are always present, also during pump operation. However, since the pressure differential from the (real) pump is significantly higher (at least ten times higher, generally more) than the pressure differential due to density differences, the influence of this effect is generally negligible.

Due to the fact that the horizontal distribution has a smaller "pump" (because of the lower height) than the risers, it is possible that the natural circulation flow in horizontal direction will not reach all risers.

To study the influence of the different heights (h_0 and h_1) it is convenient to handle the system as a circuit diagram. Fig. 9 shows the circuit based on a (simplified) system with four risers. Each valve in the circuit is labelled with a k_v value and is equivalent to the pressure losses in pipes, radiators and radiator valves, balancing valves and HEX. The differential pressures emanating from temperature differences are symbolized by pumps.

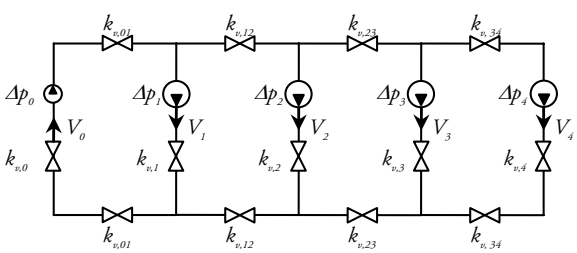

Fig. 9 A radiator system represented as a circuit diagram.

Now we use the well-known relation, applicable for turbulent flows:

$$\Delta p = \left(\frac{\dot{V}}{k_v} \right)^2 \qquad (2)$$

Note that even for natural circulation, the flow can generally be considered as turbulent. For the circuit, an equation system can be set up, and from this we can find the different flows (V_0, V_1, V_2, V_3 and V_4) for different values on the pressure differentials and the k_v values. Using realistic parameter values (corresponding to the system shown in Fig. 6), we get the result shown in the first diagram of Fig. 10. The grey bars show the relative flow in each riser at pump operation (V_0 is the total flow and hence it is 100%). The black bars show the corresponding flows during natural circulation. The textbox indicates that we assume the risers to be five times as high as the substation and that we have a temperature difference, and hence a pressure differential, in all risers. If we look at the second diagram (upper right) the conditions are slightly changed, the height ratio is the same, but now we assume that the hot water temperature typical for natural circulation has not reached the last riser (which is due to the insufficient differential pressure in the horizontal distribution circuit). Accordingly, this riser gets a significantly lower flow. In the last diagram, the pressure conditions are the same as in the previous but the height ratio is changed. We now assume the risers to be ten times higher than the substation. The result is that we end up with a *negative* flow in the last riser.

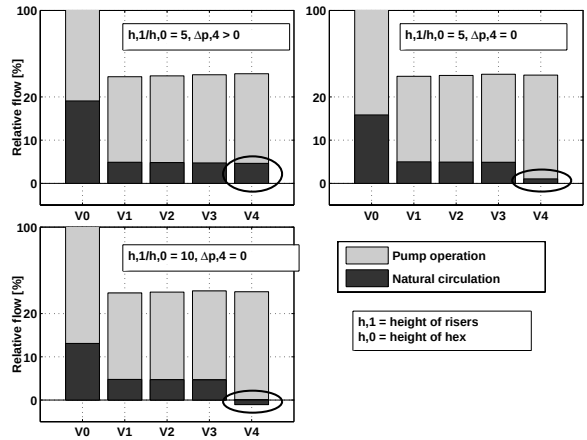

Fig. 10 Relative circulation flows for different values of h_0 / h_1 and Δp_4. Grey bars correspond to pump circulation and black bars correspond to natural circulation.

The fact that the flow might change direction in some risers far from the substation was documented in one of the studied objects. In Fig. 11 we see supply and return temperatures on six risers situated around the point in the system beyond which the natural circulation could not reach. The house was an older ten-storied residential building. When the "power failure" begins around 9:30, all risers at first receive a higher temperature. However, later on we see that the return temperature becomes higher than the supply temperature in these risers, indicating a flow in negative direction.

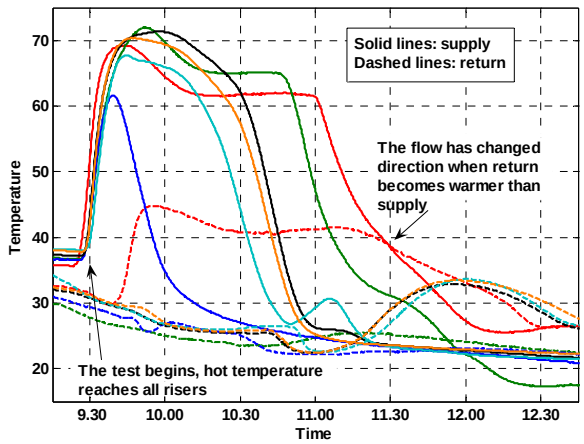

Fig. 11 Supply (solid lines) and return (dashed lines) temperatures on six risers. Each riser is represented by one colour. The pump is turned off at 9:30 and after 11:30 the flow has reversed or stopped in all risers.

Knowing that we might get a negative flow in some risers, one can wonder what the consequence will be. The negative flow in itself is hardly a problem. However, the results show that in some buildings, although the natural circulation is working well in most parts of the building, others might not receive any heat at all. This fact must be kept in mind, especially when dealing with high and/or horizontally extended systems.

As mentioned earlier, some heating systems are constructed as 1-pipe systems, which was popular during the 60s and 70s in Sweden. 1-pipe systems are typically fitted in very large buildings, often with more than one building integrated in the same radiator system. Several radiators, sometimes 10 or more, are connected in series and the systems are characterized by substantial pressure drops. An overview of a test in such a system is showed in Fig. 12, with red areas indicating where natural circulation was established.

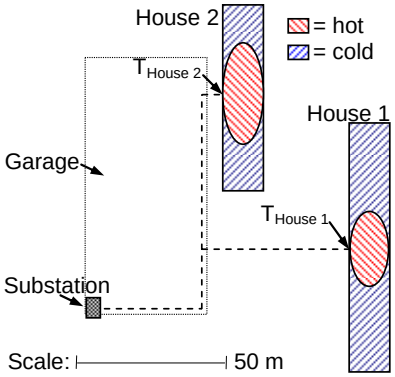

Fig. 12 Overview of a tested 1-pipe object. Red areas indicate were natural circulation is established.

As shown in the figure, the distribution pipes from the substation to the houses are almost 100 metres. Still, the flow manages to reach both houses and approximately half of the risers. Fig. 13 shows the supply and return temperatures in the radiator system measured at the substation and at the entrance to each of the two houses. We see that it takes about an hour for the flow to transport through the distribution pipes, and when a flow is stabilized, the temperature loss amounts to 7-10°C.

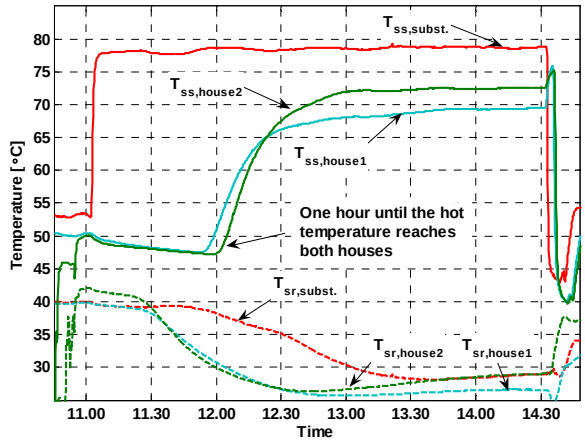

Fig. 13 Supply (solid lines) and return (dashed lines) temperatures at the substation and at the entrance to the two buildings.

It is not entirely obvious whether it is the 1-pipe system or the horizontally extended system, which constitutes the largest limit for natural circulation. The heat load estimates to about 50% and reaches approximately half the system, i.e. the heat load in a section of the system is sufficient once the circulation reaches the section. Because the circulation does not reach large parts of the system, it is likely that the effect earlier described, that the system must not be too high or long, that mainly inhibits the circulation.

Since the differential pressure is limited without electricity the system is sensitive to outer disturbances such as hydraulic short circuits. Short circuits could occur in systems when a by-pass connection is used. By-pass connections are commonly used for reducing the HW circuit temperature. There are several different types of by-pass connections. The most commonly used type in DH connected systems is shown in Fig. 14.

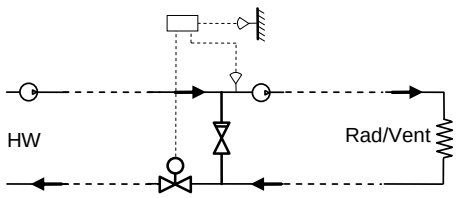

Fig. 14 Overview of a by-pass connection.

When power fails the pump in the ventilation/radiator circuit stops. The control valve stops in its position allowing HW to reach the circuit. Since this circuit has a larger height the differential pressure for natural circulation is bigger than in the HW circuit. This might cause an opening of the check valve allowing return water to mix with supply water. This results in a reduced supply temperature to the ventilation/radiator circuit. The result from a field experiment with a by-pass connection during a power failure is shown in Fig. 15.

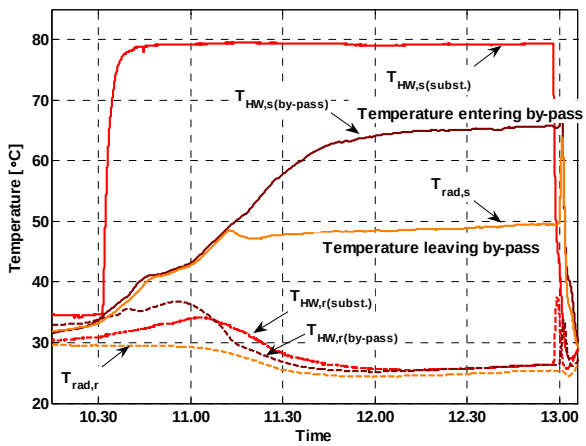

Fig. 15 Supply (solid lines) and return (dashed lines) temperatures at the substation and at a by-pass connection.

As can be seen in the figure the temperature after the bypass valve is significantly lower than the incoming HW temperature before the by-pass connection. Still, the temperature leaving the by-pass connection is sufficient for a substantial natural circulation to arise.

When assessing the potential for natural circulation in a system with by-pass connections, one can expect that they limit the circulation. It is desirable if one can make it possible to close the check valves, manually or automatically.

Normally the primary and the secondary supply pipes are connected at the top of the HEX. During the field studies, a type of substation was discovered that had the supply pipes connected at the bottom. This has no effect on the normal operation of the substation. However, in case of a power failure the water on the secondary side of the HEX becomes very hot. Without the pump in operation, the hot water will rise and leave the HEX at the top, resulting in a changed flow direction. Now, the HEX will work as a co-current HEX instead of a counter-current HEX. See Fig. 16 below.

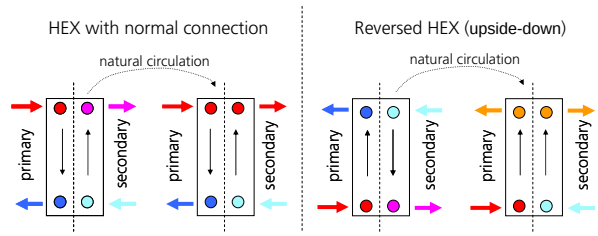

Fig. 16 Common outline of a radiator HEX (left) and the reversed type (right).

The consequence of this type of connection was studied in a field experiment, see Fig. 17.

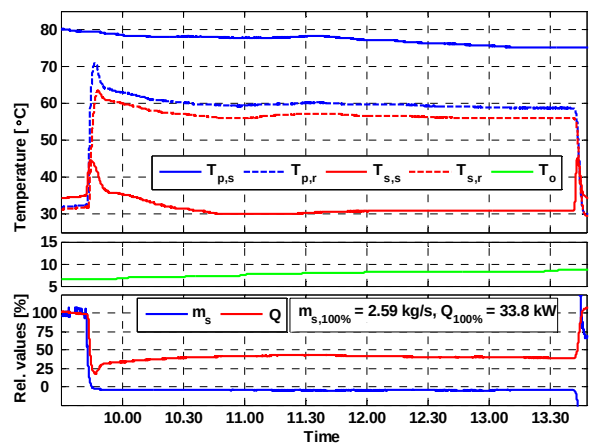

Fig. 17 Result from a natural circulation test in a system with a reversed radiator HEX. See Fig. 6 for notations.

When the HEX is working as co-current HEX the secondary supply temperature is limited by the primary return temperature (instead of the primary supply temperature). This means that the secondary temperature difference, and consequently the pressure difference, is limited. Also, the secondary flow has changed its direction, which most likely will have a negative influence on the heat output in the radiators.

Another disadvantage is that the primary return temperature becomes very high. This connection type is not to recommend due to its limited potential for natural circulation and its high return temperature during a power failure.

DISCUSSION

Although almost all hydronic heating systems today are designed to operate using an electric

pump, natural circulation is likely to arise to some extent in most systems connected to DH. This fact is of great interest for DH companies as well as for municipalities and owners of property.

A modern heating system has several obstacles for natural circulation to be able to develop. In this paper, problems which occur in very high and/or long buildings, 1-pipe systems, by-pass connections, and reversed HEXs (installed upside-down) have been shown.

It is concluded that even though heat will not be distributed evenly in a whole building, the parts that do receive heat, most likely will get enough.

With the lower temperature levels in the heating system received from by-pass connections and a reversed HEX, the potential heat output is significantly decreased.

When assessing the possibilities for a specific building to receive DH during a power failure utilizing natural circulation, the obstacles discussed in this paper should be considered.

To facilitate for a heat load as large as possible during a power failure, some pitfalls should be avoided. By avoiding reversed HEXs and by-pass connections, as long as possible, this can be fulfilled. When by-pass connections are used, it is desirable if the check valve can be closed, manually or automatically, during a power failure in order to facilitate natural circulation.

ACKNOWLEDGEMENTS

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NOMENCLATURE

Abbreviations

CV Control valve
DH District Heating
DHW Domestic Hot Water
HEX HEX
HW Hot Water circuit
Rad Radiator circuit

Variables

g Gravity force, m/s^2
h Height, m
k_v Valve capacity, m^3/(s·bar$^{1/2}$)
$\dot{m}$ Massflow, kg/s
p Pressure, Pa
$\dot{Q}$ Heat flow, W
T Temperature, °C
$\dot{V}$ Volume flow, m^3/s
ρ Density, kg/m^3

Subscripts

o Outside
p Primary
r Return
s Secondary, supply

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III

Modelling Space Heating Systems Connected to District Heating in Case of Electric Power Failure

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Abstract

Recent year's extensive power failures have put focus on the importance of secure local production and distribution of energy. Since district heating (DH) is the dominating heating system in Scandinavia it is of great importance to investigate the possibility for buildings connected to DH to receive heat during an electric power failure.

By using computer models, buildings' heating systems can be simulated, and by comparing the results with field studies the model can be evaluated. The model shows good resemblance with the field study.

By using a model the influence of different parameters can be studied. The influence of, e.g., changed primary supply temperature and outdoor temperature can be studied.

Keywords: Power failure, District heating, Computer modelling, Space heating system

Nomenclature

Variables

A	Area, m^2
c_p	Specific heat capacity, J/kgK
g	Gravity force, m/s^2
h	Height, m
k_v	Flow capacity, $kg/(s \cdot Pa^{1/2})$
$LMTD$	Logarithmic mean temp. diff., °C
m	Mass, kg
$\dot{m}$	Mass flow, kg/s
n	Radiator exponent
p	Pressure, Pa
$\dot{Q}$	Heat flow, W
t	Time, s
T	Temperature, °C
U	Overall heat transfer coeff., W/m^2K
α	Heat transfer coeff., W/m^2K
ρ	Density, kg/m^3

Subscripts

br	Branch
eq	Equivalent
exp	Experimental
hor	Horizontal
i	Indoor
int	Internal
N	Number of branches
nat	Natural circulation
o	Outdoor
p	Primary
r	Return
rad	Radiator
s	Secondary
s	Supply
sim	Simulated
vent	Ventilation
ver	Vertical
w	Wall
∞	Stationary values (after long time)

1. Introduction

In case of a large-scale power failure many important functions in the society are interrupted. Space heating is such an important and critical function. Obviously, heating systems using electricity (either directly or indirectly via e.g. a heat pump) cannot operate. Systems using oil, natural gas or pellets will not work either since electric power is demanded for the operation of burners.

District heating (DH) is the dominating heating system in Scandinavia. It comprises central heat production and distribution by hot water circulated in a grid constituted of buried pipes. The concept allows for making use of waste heat, e.g. from industries or thermal power stations.

Buildings connected to DH are also assumed not to be functioning in case of a power failure. Even if the DH supplier has back-up power in order to maintain production and distribution of heat, the buildings are gen-

erally assumed not to be able to receive any heat when the house-internal heating system is dependent on electric power to pump and control equipment. Recent studies have, however, proved that this does not have to be the case.

Extensive power failures in recent years have produced large efforts on the possibility to produce and distribute electric power locally during a breakdown on the national power grid, so-called island operation. Many power stations can produce both heat and electric power, so-called cogeneration or CHP (combined heat and power). In this case, the DH grid functions as cooler to the power generating process. To be able to produce electricity during island operation, CHP stations still need cooling. If it is still possible to use the DH grid as a cooler, the power station could still operate. However, then the end users must be able to receive district heat during a power failure.

The society can also benefit if heat supplies is functioning to an extent as large as possible. If, e.g., evacuation of citizens could be avoided or at least delayed, this could benefit not at least elderly people and persons receiving health care.

The objective of our study is to investigate what will happen in buildings connected to DH in case of a power failure. Is it possible, if the DH network still is functioning, that natural circulation will arise in modern space heating systems, and to what extent? Natural circulation was used many years ago, before pumps were introduced in space heating systems, and is based on the difference in density between hot and cool water. I.e., hot water from the heat source will rise in the vertical supply pipes and then, when heat is emitted in the radiators, the cold water will descend back to the heat source.

Computer simulations are a useful and necessary tool and good complement to field experiments in the process to establish how different heating systems and buildings are able to receive heat.

2. Objective of the study

In the study, a number of buildings of various sizes, age and type have been examined through practical field studies. To be able to study different outer circumstances and the influence of different parameters, e.g. the outdoor temperature, the field studies have been complemented with computer simulations using MATLAB's toolbox Simulink. The simulations are a useful instrument when estimating how much district heat that could be distributed during a power failure. It is also possible to study the influence from various parameters that might be difficult to capture in reality, e.g. the possible gain from an increased DH supply temperature.

The objective of the paper is to show possibilities to simulate natural circulation in space heating systems and the dynamic influence on the indoor air temperature. The agreement between results from simulations and field experiments will be discussed.

3. Space heating systems connected to DH

Figure 1 below describes a waterborne radiator space heating system connected to a DH grid. The water from the DH grid (primary water) is led into a heat exchanger (HEX) where the radiator water (secondary water) is heated. The secondary supply temperature, T_{ss}, is regulated by a controller which in turn regulates the DH flow passing through the HEX. The set point for T_{ss} is based on the current outdoor temperature and the building's time constant (a sudden change in outdoor temperature should not momentarily be compensated for due to the thermal inertia of the building shell). The circulation in the radiator system is achieved by an electric pump. In order to receive proper indoor temperature in the whole building, valves are used to balance the flow between risers and radiators in different parts of the system.

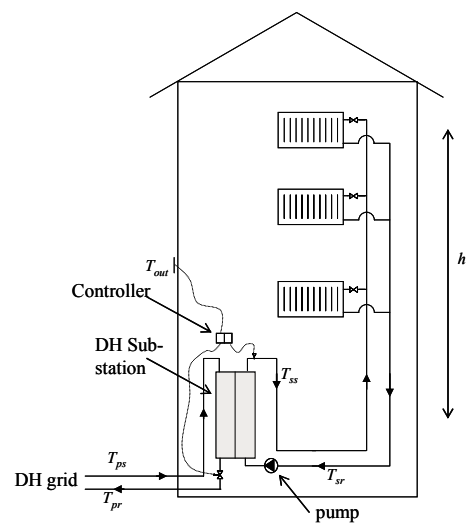

Figure 1 A building with space heating system connected to a DH grid

4. Computer modelling

The building model is based on existing models of HEX, control equipment, actuators, valves, connection pipes (taking into account pressure drop, heat loss and time delay), radiators and building. The theory and function of these components have been described in detail by a number of authors, for example [1], [2] and [4].

In this work, these components have been put together in order to constitute a complete building model with DH substation, space heating system and building shell. To be able to meet the objective of this work, the model has also been modified and improved on several points. The space heating system has been extended to comprise four risers and three storeys, making it possible to study the heat distribution in the building during natural circulation. The flow distribution with pump operation is built on a method based on an analogy to Kirchoff's circuit laws. In case of natural circulation the problem must however be attacked in a different way, which will be described in section 4.1. In order to study the dynamics of the indoor air temperature, in case of a complete or partial break in heat supply, the modelling of the building shell has been modified, which will be described in section 4.2.

The overall structure of the complete model is shown in Figure 2 below.

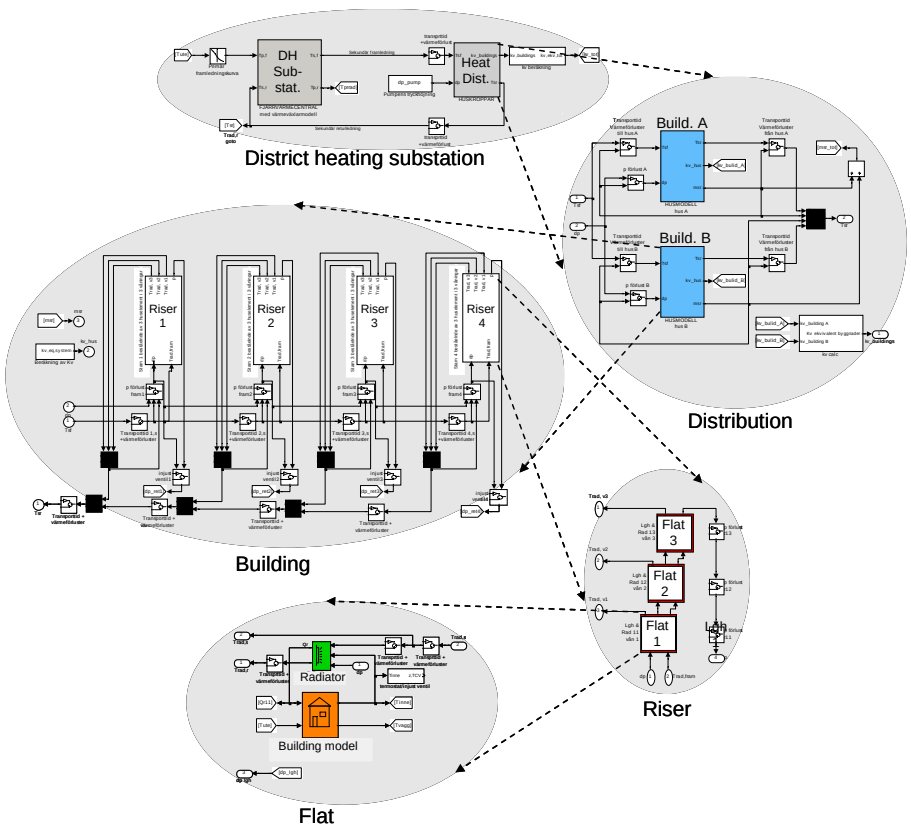

Figure 2 Overview of the computer model of the system including substation, space heating system and building

4.1 Basic parts of the model

The first step in the energy transfer from the DH grid to the building takes place in the DH substation where heat is transferred via a HEX. A control valve adjusts the DH flow in order to achieve a correct outgoing temperature on the secondary side of the HEX, i.e. the radiator supply temperature. A traditional way to handle the dynamic thermal behaviour of the HEX is to divide it into a number of sections. For each section energy balances can be stated for both primary and secondary side flows, and for the wall separating them. After differentiating the following equations are obtained:

$$\frac{\partial}{\partial t}\left(T_{s,out}\right)=\frac{1}{m_s\cdot c_{p,s}}\left[\dot{m}_s\cdot c_{p,s}\left(T_{s,in}-T_{s,out}\right)-\alpha_s\cdot A\left(\frac{T_{s,in}+T_{s,out}}{2}-T_{wall}\right)\right] \tag{1}$$

$$\frac{\partial}{\partial t}\left(T_{p,out}\right)=\frac{1}{m_p\cdot c_{p,p}}\left[\dot{m}_p\cdot c_{p,p}\left(T_{p,in}-T_{p,out}\right)-\alpha_p\cdot A\left(\frac{T_{p,in}+T_{p,out}}{2}-T_{wall}\right)\right] \tag{2}$$

$$\frac{\partial}{\partial t}\left(T_{wall}\right)=\frac{A}{m_{wall}\cdot c_{p,wall}}\left[\alpha_p\left(\frac{T_{p,in}+T_{p,out}}{2}-T_{wall}\right)-\alpha_s\left(\frac{T_{s,in}+T_{s,out}}{2}-T_{wall}\right)\right] \quad (3)$$

The equations are easily implemented in Simulink. The temperature profile is assumed to be linear, while the theoretically correct profile is logarithmic .The logarithmic mean temperature difference, *LMTD* is defined as:

$$LMTD=\frac{\left(T_{p,s}-T_{s,s}\right)-\left(T_{p,r}-T_{s,r}\right)}{\ln\left(\frac{T_{p,s}-T_{s,s}}{T_{p,r}-T_{s,r}}\right)} \quad (4)$$

As mentioned, an arithmetic mean temperature difference has been used. The advantage with the later is that the heat transfer can be directed in both ways, i.e. from primary to secondary side and vice versa. Such situations can occur for short periods in the radiator HEX. This means that we must use a sufficient number of sections in the model. On the other hand, an increasing number of sections will increase the computational time. A suitable choice is to divide the HEX into 3-5 sections, [5]. The method involves a number of assumptions such as uniform temperature in each section, no conduction in the water flow direction or in the length direction of the HEX wall. The heat transfer coefficient, α, is approximated to be a function of the flow only, ignoring minor influence from temperature dependant quantities. The thermal resistance in the wall material can be ignored due to the thinness of the plate. All these assumptions ease the mathematical description, but investigations still have proved a good resemblance to real data. [1], [7].

The piping system model includes three features, pressure drop, heat loss and time delay. The pressure drop due to friction in pipes, valves, radiators and HEX is essential when calculating the flows in the system. The time delay could be of some interest at normal (pump) operation, while the heat loss is of minor interest. While simulating natural circulation, however, all of these features become essential. The driving force from natural circulation is rather small, leading to a substantially smaller circulation flow and, consequently, a low flow velocity. Especially when studying larger buildings, transportation times and heat losses can be considerable and essential for the operation of the system.

The pressure loss calculations will be discussed later on in connection to flow calculations.

The next level of heat transfer in the system takes place in the radiators. Basically, the heat transfer here works in the same way as in the HEX. The model can, however, be simplified on some points. On both sides of the HEX the medium is water, while in the radiator we have water on one side and air on the other. The air has a substantially lower heat transfer coefficient than water and is the absolutely dominating resistance in the radiator heat transfer. The overall heat transfer number, *U*, can therefore be assumed to be constant. We are interested in calculating the return temperature and the emitted heat from the radiator. The air temperature can be regarded as constant in the calculations. In the radiator, we can expect that the heat transfer always will be directed from the radiator water to the air. Therefore, we now can use the *LMTD* calculation (as shown in (4)) and do not have to divide the radiator into sections. We then arrive with the following two equations:

$$\frac{\partial}{\partial t}\left(T_{rad,out}\right)=\frac{1}{m_{rad}\cdot c_{p,rad}}\left[\dot{m}_s\cdot c_{p,s}\left(T_{rad,in}-T_{rad,out}\right)-U_{rad}\cdot A\cdot LMTD^n\right] \quad (5)$$

$$\dot{Q}_{rad}=\dot{m}_s\cdot c_{p,s}\left(T_{rad,in}-T_{rad,out}\right) \quad (6)$$

Note that the *LMTD* is raised to the exponent *n*. This is an approximation to the fact that only part of the heat transfer from the radiator is due to convection, the rest is due to radiation. The contribution from radiation (which normally amounts to 30-50 per cent of the heat transfer) is also a function of the temperature difference between the radiator's surface and the surrounding walls. The complex relation, which also includes physical quantities, can be simplified by including the radiation in (5) by introducing the so-called radiator exponent, *n* [6].

A common method to model the dynamics of the building's heat losses is described for example in [1] and [3]. The basic idea is more or less the same as with the HEX and the radiator. The wall is treated as a homogenous material with a homogenous temperature described as:

$$\frac{\partial}{\partial t}\left(T_w\right)=\frac{1}{m_w\cdot c_{p,w}}\left[\alpha_i\cdot A\left(T_i-T_w\right)-\alpha_o\cdot A\left(T_w-T_o\right)\right] \quad (7)$$

With the wall temperature known the heat flow from the indoor air to the wall can be calculated from:

$$\dot{Q}_w=\alpha_i\cdot A\left(T_i-T_w\right) \quad (8)$$

The indoor air temperature can finally be calculated from:

$$\frac{\partial}{\partial t}\left(T_i\right)=\frac{1}{m_i\cdot c_{p,i}}\left[Q_{rad}+Q_{int}-Q_w-Q_{vent}\right] \quad (9)$$

Q_{int} is internally generated heat from humans and electric equipment like TV's, refrigerators etc. Q_{vent} is the unavoidable heat loss from ventilation of the indoor air.

The heat carrier from the HEX to the radiator is the circulating water in the system. The flow depends on the available pump head and the flow resistance in the system. For fully turbulent flows and limited temperature variations, the relation between flow and differential pressure in a component can be approximated as:

$$\dot{m} = k_v \sqrt{\Delta p} \tag{10}$$

It is reasonable to assume that the flow is turbulent, even when dealing with natural circulation. If a laminar flow should occur, the pressure drop will be so small, in absolute numbers, that the influence will be negligible. The factor k_v describes the flow capacity of a component (to be precise, the flow capacity at 1 bar differential pressure). A k_v value can be calculated for all components, a pipe section, valve (which has a variable k_v value depending on the opening degree), HEX or radiator, as long as the pressure loss for a specific flow is known. For a system with many components (connected in parallel or series), equivalent k_v values can be calculated from:

$$k_{v,eq,parallel} = \sum_{i=1}^{N} k_{v,i} \ , \quad k_{v,eq,series} = \frac{1}{\sqrt{\sum_{i=1}^{N} \left(\frac{1}{k_{v,i}} \right)^2}} \tag{11}$$

Let us now consider the hydronic radiator system as a circuit diagram, see Figure 3. This analogy to electric circuit laws is described in [4] for a domestic hot water system. For branch i, k_v values are indicated for supply and return pipes to the branch and to the radiator, respectively, and for the radiator itself, including its control valve which can be controlled manually or by a thermostatic head. In fact, every branch in the model consists of three storeys, although only one is indicated in the figure. However, each branch in turn comprises a system just like the one shown in the figure.

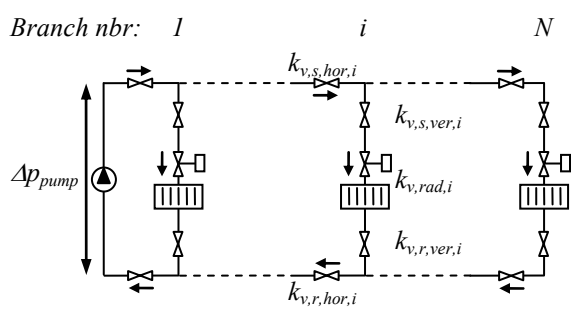

Figure 3 Schematic picture of a space heating system

Equation (11) can now be used to simplify the system by replacing flow capacities, in series or in parallel, by equivalent capacities which in turn can be simplified. The procedure is shown in Figure 4 below.

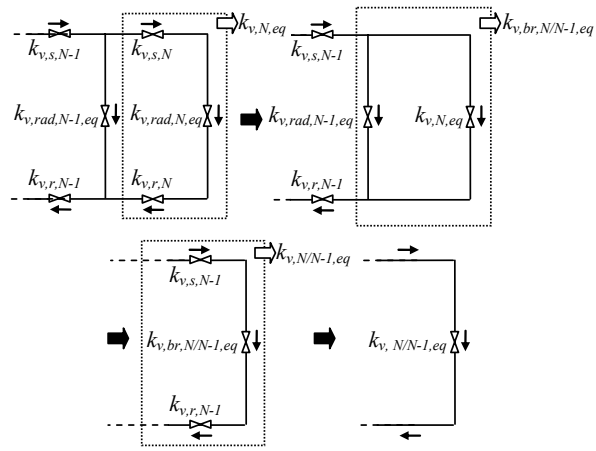

Figure 4 Simplification of the circuit scheme of the space heating system

Finally, we arrive with just one equivalent k_v value for the whole system, and (10) can be used to calculate the total circulation flow, $\dot{m}_s$, in the system. Then, the flow in each part of the system can be calculated by once again going through the system, but now in the opposite direction. We know the available differential pressure for the first branch (after the pump) and the equivalent k_v value and (10) gives us the flow through branch 1. We can then carry on by subtracting the pressure loss from branch 1 and repeat the calculation for branch 2, and so on.

Nowadays, the use of speed controlled pumps is a rule rather than an exception. In the model the pump pressure head is therefore assumed to be constant.

4.2 Improvements and expansion of the model

In this work, the model is used to facilitate the studies on how space heating systems connected to DH will behave without electric power. In order to do that the model needs to be modified. The first step is to introduce the driving force from natural circulation.

4.2.1 Natural circulation

This mechanism is based on the difference in density between hot and cold water. Many years ago, before pumps were introduced, heating systems were designed to operate with natural circulation. The differential pressure originating from natural circulation can be derived from:

$$\Delta p_{nat} = g \cdot h \left(\rho_r - \rho_s \right) \tag{12}$$

The height, h, in the system is measured between the heat source (HEX) and the heat sink (radiator). This force is of course present also in a pump operated system, but is negligible compared to the pump's pressure

head, and can be neglected. When designing a natural circulation system, large pipe diameters must be used in order to minimize pressure drops. As seen in (12) a high building and a high supply temperature support natural circulation. The question is how this works in a system designed for pump operation.

The problem is that the Kirchoff method will have limitations when the pump head is lost. As a first approach, the differential pressure from the pump can be substituted with a calculated differential pressure from (12), based on supply and return temperatures at the HEX and the system's height. This is, however, a simplified model, since the differential pressure actually arises at every riser in the system, i.e., where the height actually is situated. When testing natural circulation in large buildings, it turns out that some risers, situated far from the HEX, might not obtain any circulation, or it might take a long time until circulation arises. Also, the magnitude of the circulation tends to be exaggerated when the temperature difference at the HEX is used for calculations in all risers. During natural circulation, the flow velocity is drastically reduced which means that heat losses will be significant and the supply temperature at the risers will be more or less reduced, a fact that gives a reduced driving force from natural circulation.

The mechanism for natural circulation could be compared to having a small "pump" at every riser. In our case, the model has four risers which mean that we have four such "pumps" or sources for differential pressure. With four sources instead of one, we must consider four circuits when calculating the flow distribution. The flow in each circuit is calculated individually and the total radiator flow corresponds to the sum of the four circuit flows. The difference between calculating flows in the system driven by pump or by natural circulation, respectively, is shown in Figure 5.

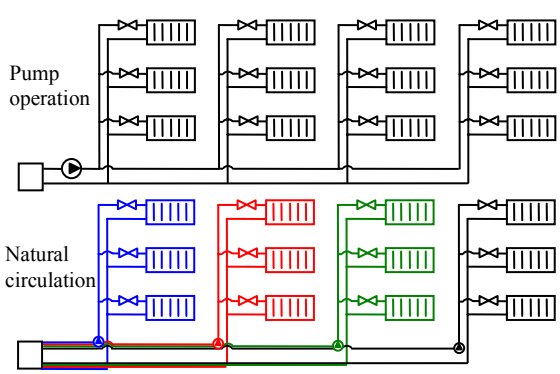

Figure 5 Difference between pump and natural circulation regarding flow calculations

Regarding the implementation of natural circulation in Simulink, a smooth way to handle the to operational modes, pump or natural circulation, is to introduce a control variable which has the value 1 as long as the power supply is OK (pump operation) and 0 when the power fails (natural circulation). The control variable will then govern the switching between the operational modes and, consequently, flow calculations. The control equipment is also affected by a power failure, and this is easily taken care of using the control variable to freeze the position of the DH valve in its current position.

4.2.2 Building shell model

The model of the building shell showed in section 4.1 has proved to be sufficient when modelling the operation of space heating systems during "normal" conditions or static calculations, i.e. where the transient course of the indoor air temperature is of minor interest. However, dealing with power failures must be considered as rather extreme conditions. The existing model showed problems using the conventional building model where the wall is modelled as one section with a homogeneous temperature. Let us consider an example: The outdoor temperature is -10°C and the indoor temperature is 20°C. The temperature throughout the wall will then be (20 – (-10)) / 2 = 5°C. If the heat supply is cut off (or substantially decreased) the indoor air temperature will drastically approach the wall temperature, 5°C. This is not realistic since the wall will have a temperature gradient where the inside surface is just a few degrees cooler than the indoor air temperature.

Figure 6 below shows data from an experiment carried out in a small office at our department. The radiator and ventilation in the room was shut of and an electric radiator, equipped with a timer function, was installed. Several temperatures were measured in the room, among them indoor and outdoor air, the temperature on the inside wall and on the inner surface of the outer wall. The figure shows what happens when the heat supply is cut of. During the test the outdoor temperature was about 5°C.

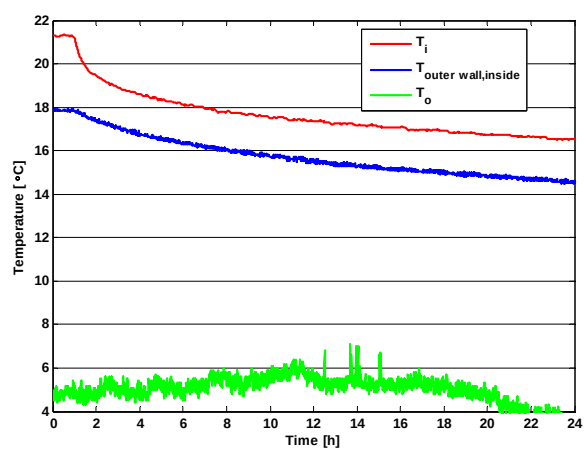

Figure 6 Results from an experiment where the heat supply to a room is cut off

As can be seen, the indoor temperature initially approaches the inside surface temperature of the outer wall. Figure 7 below shows a simulation corresponding to the experiment above. The figure comprises results from, on one hand, the existing model (the curve indicated with "◊") and, on the other, the modified model (the curve indicated with "o"). With the one-section model the wall temperature lies exactly between the indoor and the outdoor temperature. As already mentioned, the indoor temperature drastically approaches the wall temperature and estimates to 7°C already after 24 hours.

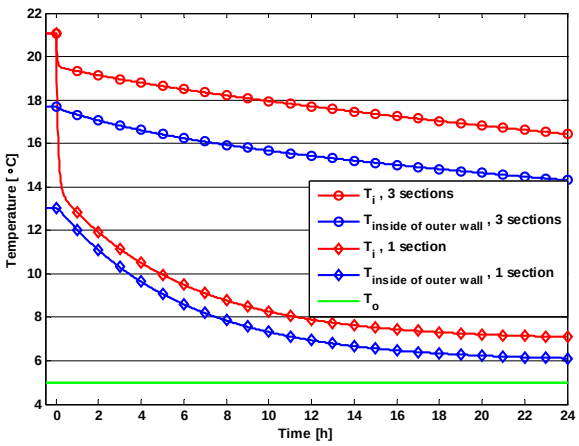

Figure 7 Simulation using the one-section and the three-section building wall model, respectively, where the heat supply to a room is cut off

To improve the modelling of the indoor temperature the building model has been modified according to Figure 8. To the left is the simple, one-section, model and to the right is the new, three-section, model. By using more sections, different wall materials can be used. Here, the two innermost sections are assumed to be made of concrete, which has a high conductivity but a large mass, and the outermost section is the insulation, which has a low conductivity but a small mass. Both materials are essential. The concrete will assure that we have an inside wall temperature relatively close to the indoor temperature and make sure that the building has a realistic thermal inertia. The insulation will assure that we get the correct heat loss and temperature drop through the wall. The choice to use three sections is a compromise; more sections give higher accuracy but longer computational times.

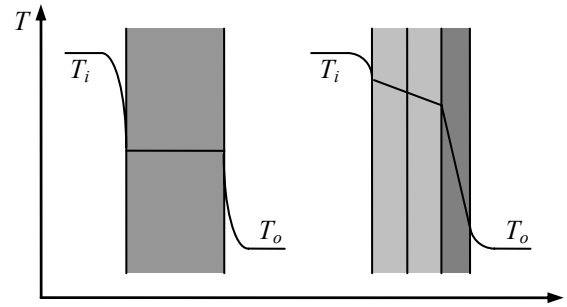

Figure 8 Schematic picture of the one-section (left) and the three section (right) wall model

Now, using the new three-section model, the simulation was run once again. The simulation result, already shown in Figure 7, is now much more in accordance with the results of the real experiment (Figure 6).

4.3 Assumptions for the model

Regarding the model, a number of assumptions have to be made in order to limit the complexity of the model and the computational time.

Most of the assumptions are already discussed, such as *LMTD* versus arithmetic mean value calculation in the HEX model and number of sections in the wall model. Another essential assumption is that there is no heat transfer between the flats. Without this assumption the model would be extremely complex and demand large computational power. The consequence of the assumption is that if the radiator flow is unbalanced the indoor temperature in the flats will not be accurate due to the fact that heat conduction through the walls between the flats is neglected. However, the average temperature in the building would still be correct. This is only relevant if the flow is unbalanced for a long time since heat transfer between the flats is very slow.

The supply temperature in the DH grid is a function of the outdoor temperature. In this study the temperature profile for Malmö is used for the simulations, see Figure 9.

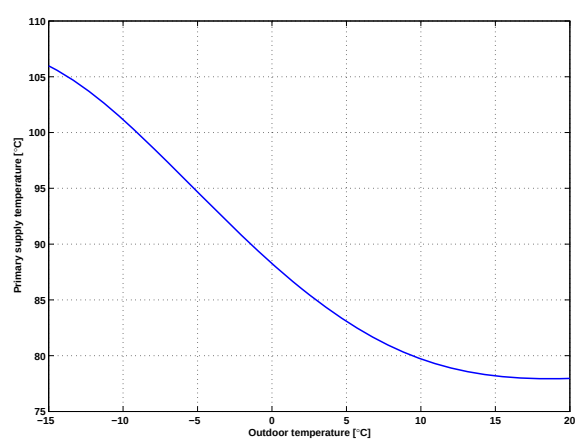

Figure 9 The supply temperature in the DH grid

5. Results

In order to evaluate the accuracy of the model, field studies and simulations with similar outer conditions have been compared. The model has thereafter been used for parameter studies.

5.1 Model evaluation

For evaluation of the model a comparison between a real field study and simulations are made in Figure 10. The studied object is a building located in Malmö. The building was built in 1952 and consists of 20 residential flats distributed on three floors. Before the time 0 in the diagram, the operation is normal and we have 100 per cent circulation flow and heat output. At the time 0, a power failure occurs and the circulation flow decreases drastically. As a consequence, the radiator supply temperature increases substantially. Finally, we end up with a substantially higher temperature drop in the space heating system, which leads to a rather high heat load even with natural circulation.

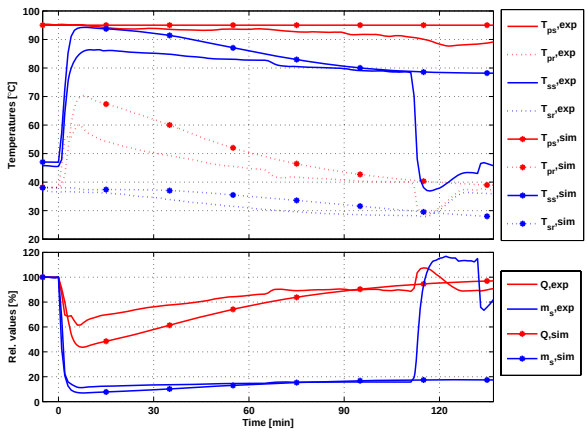

Figure 10 Comparison between field study and simulation

As seen, the simulation shows good resemblance with the field experiment. Even if there is some difference in the transient phase good resemblance is shown both initially (before the power failure) and when the natural circulation has come to a steady state.

Possible explanations to the deviation are the estimated time constant for the building and the space heating system (water volume, pipe lengths etc.) and variation in primary supply temperature.

The model now makes it possible for example to investigate long time effects of a power failure for the building. In Figure 11 a power failure lasting for three days is simulated for the same building.

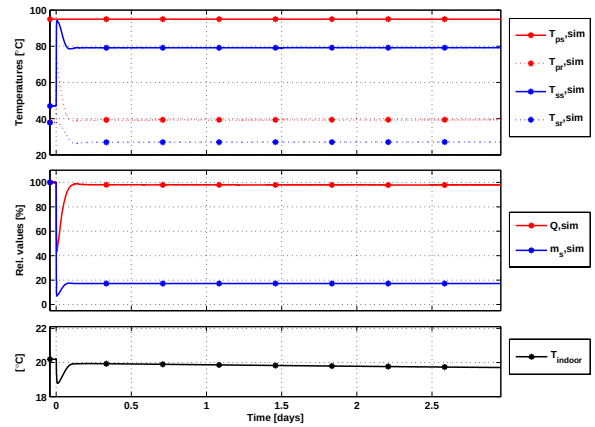

Figure 11 Simulation of a power failure lasting for three days

The figure shows the obvious fact that the natural circulation, once started, will go on at the same level after reaching a steady state. The mean indoor temperature is also represented in the figure.

5.2 Parameter variation

In the modelling world it is possible to investigate the influence of different parameters and outer conditions.

In case of a power failure the DH company can adjust the primary supply temperature level. As described in section 4.2.1 the magnitude of the natural circulation flow is dependent on the temperature drop in the radiator system. In Figure 12 steady state results (the state after infinite time) from simulations with different primary supply temperatures are shown at different outdoor temperatures. In the upper diagram the relative heat load (natural circulation compared with normal operation) is shown. The lower diagram shows the mean indoor temperature.

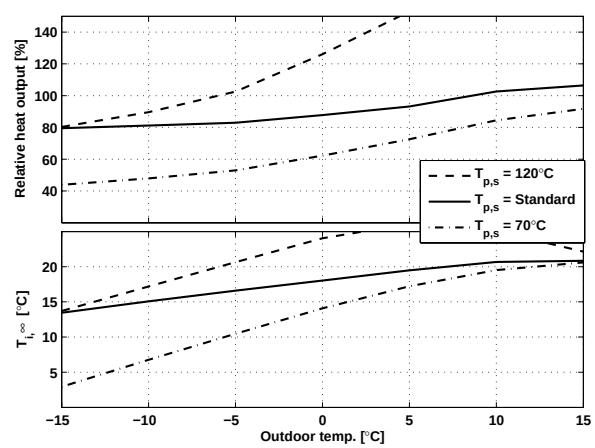

Figure 12 Variation of primary supply temperature at various outdoor temperatures at steady state conditions

The figure shows that by increasing the primary supply temperature the relative heat load increases, especially for outdoor temperatures above -10°C. At low outdoor

temperatures the effects of increasing the primary supply temperature to 120°C is small due to the already high supply temperature.

The next diagram, Figure 13, shows the dynamic indoor temperature at three different outdoor conditions. Also in this figure the influence of changed primary supply temperature is shown.

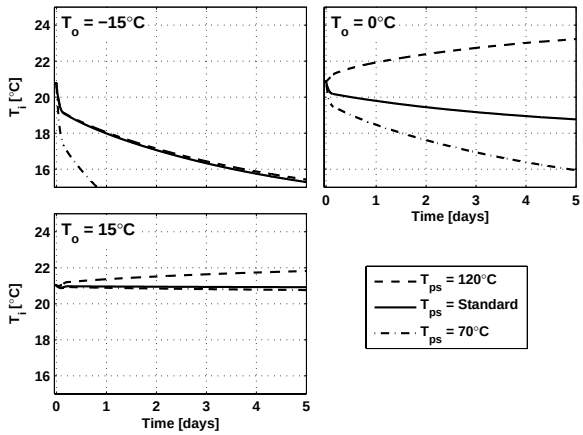

Figure 13 Dynamic indoor temperature at different outdoor temperatures and variation of primary supply temperature

5.3 Conclusions

A complete model of a building with a hydronic space heating system connected to DH has been built up. Parts of the model have been improved and developed to make it possible to simulate not only normal operation using a pump but also natural circulation during a power failure. By comparing the results from simulations with field studies good resemblance can be shown. A model makes it possible to study a building when changing input parameters, e.g. outdoor temperature. It is also possible to simulate the expected indoor temperature dynamic as well as at steady state in case of a power failure.

The model will constitute an important tool in our further studies regarding secure heat distribution during power failure.

6. Acknowledgements

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District heating in case of power failure

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ABSTRACT

Power failures in combination with harsh weather conditions during recent years have led to an increased focus on a safe energy supply to our society. Many vital functions are dependent on electricity; e.g., lighting, telephony, medical equipment, lifts, alarm systems, payment, pumps for town's water and, perhaps the most critical of all, heating systems. In Sweden, district heating (DH) is the most common type of heating for buildings in town centres. Therefore, it is of great interest to investigate what happens in DH systems during a power failure. The present study shows that, by maintaining the DH production as well as the operation of the DH network, possibilities to supply connected buildings with space heat are surprisingly good. This is due to the fact that natural circulation will most often take place in radiator systems. In Sweden, and in many other countries, so-called indirect connection (heat supply across heat exchangers) of DH substations is applied. If a DH network operation can be maintained during a power failure, DH water will continue to pass the radiator system's heat exchanger (HEX), provided that the control valve does not close. The radiator circulation pump will stop, causing the radiator water to attain a relatively high temperature in the HEX, which promotes a natural circulation in the hydronic heating system, due to an increased water density differential at different temperatures. Several field tests and computer simulations have been performed and have displayed that almost all buildings can achieve a space heat supply corresponding to 40–80% of the amount prior to the interruption. A sufficient heat load in the DH network can be vital in certain cases: e.g., for 'island-operation' of an electric power plant to be performed during a power failure. Furthermore, for many combined heat and power stations, a requirement involves that the DH network continues to provide a heat sink when no other cooling is available. Based on the findings presented herein, a set of recommendations have been set up to provide advice to, among others, DH utilities and owners of customer buildings.

1. Introduction

A large-scale power failure affects numerous vital functions of society, among them space heating. Obviously, during a power failure, heating systems that depend on electricity being converted into heat energy (either directly or via a heat pump) cannot operate at all, and boilers (oil, natural gas, pellets, etc.) are also prevented from functioning, since electricity is required for the operation of burners and control equipment.

District heating (DH) is common in many countries and in, e.g., Denmark, Finland and Sweden it constitutes the dominating method for heating larger buildings. Thus, 76% of the total building floor area of multi-dwelling houses in Sweden is heated with DH [17], and it is for this reason of vital interest to study what can be expected to happen in such buildings in case of a power failure, assuming that the production of DH can be maintained.

Hydronic radiator systems in modern buildings are fitted with circulation pumps whose motors stop when the electricity supply fails. Consequently, the heating of buildings connected to a DH network is generally assumed to stop functioning in such an event. Nevertheless, as will be demonstrated in this paper, a substantial natural circulation effect caused by an increased water density differential takes place in many cases.

The major power failures that have occurred in recent years have led to an increased focus on the possibility of a local production and distribution of electric power during a breakdown of the national power grid, so-called 'island-operation'. In order for the production of electricity in combined heat and power (CHP) stations to be maintained, it is necessary that excess heat continues to be removed from the power generating plant. If it is possible to use the DH network as a heat sink, i.e., if buildings connected to the DH network carry on consuming heat energy, provisions for additional cooling of the plant may not be necessary.

For our society, it is essential that building heat supplies be maintained, especially in the event of a power failure of long duration during harsh weather conditions. If evacuations of people from buildings can be avoided, or at least delayed, this will help protect especially elderly people and persons in need of health care from risky exposure to low air temperatures.

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Nomenclature

Variables

g	standard gravity (m/s^2)
h	height (m)
l	length (m)
T	temperature (°C)
ρ	density (kg/m^3)

Subscripts

∞	infinity, steady state
i	indoor
nat	natural circulation
p	primary
r	return
s	secondary, supply

Abbreviations

CHP	Combined Heat and Power
DH	District Heating
DOT	Design Outdoor Temperature
HEX	Heat EXchanger

1.1. Objective

The objective of the present study was to investigate what can be expected to happen in space heating systems connected to DH when a power failure occurs. It was explored whether it is possible, provided that the DH network water circulation and heat transportation continues to function, for natural circulation to arise in space heating systems, and if so, to what extent. Natural circulation has been used in the past, whereas modern systems are designed for pump circulation to overcome a significantly higher flow resistance in pipes of smaller diameters.

Several field studies have been carried out to investigate the potential for natural circulation in a variety of building types. Additionally, computer simulation has been employed to study variation effects of system parameters. Based on results from these investigations, we have set up recommendations for how the concerned parties can be appropriately prepared to minimise hazard risks. The results form a basis for municipal risk planning.

1.2. Securing heat supply

The presented project was carried out in co-operation with, among others, the municipality of Malmö (Sweden's third largest town with 290,000 inhabitants). An analysis of the society's vulnerability to a power failure of long duration during harsh weather conditions pointed out an interrupted heat supply as the most serious threat for the inhabitants as well as for municipal activities such as health and geriatric care [5]. Certain buildings and activities have back-up power, e.g., hospitals, the town house, as well as the water supply and sewage plants.

Shortcomings in the preparedness for elder care has been demonstrated [23] and it was found that more than 40% of the facilities in Sweden lacked access to back-up power or heat. Another study [2] presented similar results. Limits for acceptable duration of outages have been set to six hours for care facilities and to 24 h for dwellings [4].

One of the most well known power failures occurred when an ice storm hit Canada in 1998. 4.7 million people were left without power and heat for a prolonged period in the middle of winter, which demonstrated the need for planning and the benefit of emergency preparedness [9], [11]. There were no DH systems in the area; instead heating was primarily obtained from electricity and oil. The blackout caused extensive evacuations; many people could move in with relatives and friends while others had to stay in public shelters. Studies have shown that people felt that the most significant strain consisted in trying to keep warm [22].

The storm Gudrun that struck southern Sweden in January 2005 caused widespread, long-lasting blackouts and became a reminder of how vulnerable our society is when energy supplies fail. However, the weather was relatively mild and larger towns with DH systems did not have as long-lasting power failures as smaller towns without DH [20]. Nevertheless, the present investigation demonstrates that a small village that could restore the DH network regained approximately 80% of the heat load, despite the fact that most customers had no electricity. This indicates that natural circulation occurred for several customers.

DH is generally considered to be a reliable technique, cf. for instance a Finnish study [12], and in the USA, CHP is often presented as a very reliable form of energy supply [24]. The starting point for this work, however, is to study the implications for DH if a power failure occurs.

1.3. Island-operation

In August 2003, 50 million people in the north-eastern USA were affected by a power failure, and certain areas were without power for four days. Although DH is relatively small in the US, small-scale co-generation is rather common. For example, a CHP unit can provide a university campus, hospital, industrial or residential building with both power and heat, and sometimes also cooling. One study [1] describes how different CHP systems acted in connection with the power failure. The vast majority, including units not designed for stand-alone operation, functioned satisfactorily during the blackout. The heat load was rather low during the time in question, but the ability to maintain, or restore, the power supply in these systems was of considerable importance to minimise the impact of the blackout.

An island grid should imply that all DH customers have access to electricity and thus to a functioning heat supply. In practice, however, island-operation signifies limited supplies of electricity that can be directed to particularly sensitive activities. The availability depends on which production facilities that can be employed and where they are located; many production units are situated in areas with low population densities.

If it is possible to employ the DH network as a heat sink, the use of additional cooling in the CHP unit may not be necessary. The DH supplier in Malmö, i.e., E.ON, was interested in determining whether, during a power failure, some of their CHP units could work in island-operation mode with the DH network as the only available heat sink. Even though there existed back-up power to operate the DH network during a power failure, there was no knowledge concerning what happens in customer installations, i.e., what level of heat supply, if any, can be expected.

It can be argued that the advance of the most centralised form of building heating represented by district heating, sometimes serving a majority of all buildings within a town, has made building heating more vulnerable in certain respects. On the other hand, as this paper attempts to demonstrate, DH possesses a great potential for upholding heat supply.

2. What happens with the heat supply when the power fails?

2.1. Production

This work focuses on what happens in the buildings connected to DH. If any heat is to be transferred, the DH network needs to function, at least to a certain extent. A Swedish study has demonstrated that conditions differ between places [19]. Some DH utilities have back-up power to keep the system in operation, while others do not, or are only able to protect their network from freezing. The issue is however important, not only because of the matter of natural circulation, but regarding whether customers with back-up power, such as hospitals, can expect to receive DH.

2.2. Substations

In many countries, a so-called indirect connection is adopted in DH substations, signifying that a heat exchanger (HEX) hydraulically separates the primary system (DH network) and the secondary system (house-internal), [16] and [3]. This paper is devoted solely to indirect DH connections. However, in some countries, direct connection is common. The use of HEXs is an obstacle for obtaining heat transfer between the DH network and the secondary systems when no electricity is available. When using direct connections, it should in general be easier to uphold the circulation in the secondary systems. A special type of direct connection has often been employed in Eastern European DH systems where ejector pumps are used in substations, rendering it possible to deliver heat, even if the customers are without electricity, [13] and [14].

The actuator manoeuvring the control valve in a substation is generally motor-driven and consequently depends on electric power in order to function. Generally, the control valve stops in its current position. Contacts with manufacturers show that closing valves are not used in heating systems since they are more expensive, whereas they are often used in domestic hot water systems to avoid the risk of scalding.

The circulation pump, powered by electricity, is another key component in the DH substation. The seemingly widespread perception that DH-connected systems stop functioning at a power failure is based on that the circulation ceases when the pump stops.

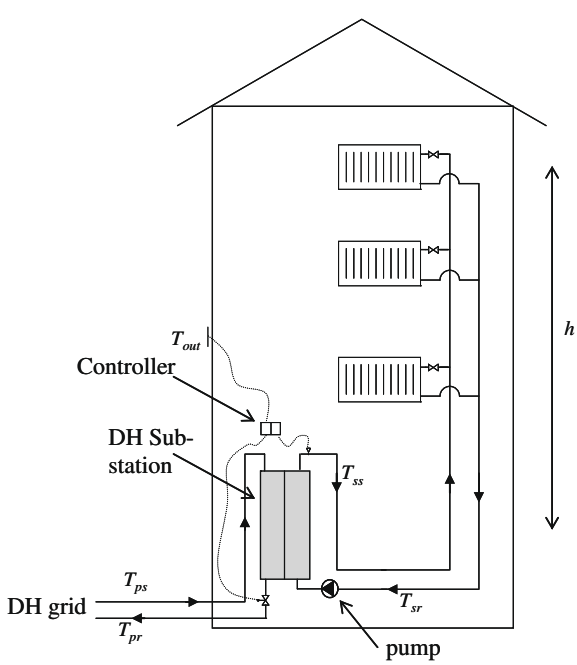

Fig. 1. A building with a hydronic radiator system connected to DH.

3. Natural circulation

3.1. Mechanism and history

Space heating systems built during the early days of central heating were constructed for natural circulation. When water is heated, its volume increases as a result of its density decreasing, and the water thus rises in the system. In reverse, when the water is cooled off in the radiators, the density increases and the water descends back to the heat source (see Fig. 1).

The pressure differential originating from natural circulation is proportional to the density difference and the height of the system according to the following equation:

$$\Delta p_{nat} = (\rho_r - \rho_s) \cdot g \cdot h \tag{1}$$

where Δp_{nat} is the differential pressure; ρ_r the water density in return pipe; ρ_s the water density in supply pipe; g standard gravity and h is the height difference between supply and return pipe.

More information regarding natural circulation can be found in reference [6]. By using large pipe dimensions in the distribution system and a boiler with a low flow resistance, pressure losses can be kept to a minimum and it becomes possible to achieve a sufficient circulation flow [15].

3.2. Natural circulation in pump-driven systems

Systems built for pump operation have significantly larger flow resistances due to smaller pipe diameters and the use of balancing valves. Modern plate HEXs also give rise to an increased flow resistance. Field studies have been carried out on 14 objects, including smaller and larger multi-dwelling buildings, single-dwelling and non-residential buildings. Although the results differed significantly, certain fundamental similarities existed between the tests.

When testing a building, the power supply to the radiator circulation pump was cut off and the control valve was set to manual mode, resulting in it becoming frozen in its current position. Temperature sensors were mounted on incoming and outgoing pipes to the HEX, both on the primary and secondary sides. The primary flow was obtained from the heat meter whereas the secondary flow was determined from a simple energy balance.

Fig. 2 describes the results of a field experiment with natural circulation. The upper diagram shows the temperature levels in the HEX, where the black lines indicate the primary side and grey lines represent the secondary side. The solid lines designate the supply and the dashed lines the return temperatures. The diagram in the middle shows the outdoor temperature and the lower diagram displays the secondary flow rate (black) and the heat supply (grey) relative to the level prior to the start of the test. Just before 9:45 AM, the power to the circulation pump was cut off and the control valve was frozen at the current position by placing it in manual mode, which caused the secondary flow to decrease drastically. Since the primary flow remained constant, the HEX became very hot and the supply temperature to the radiator system was increased – almost to the same level as the DH supply temperature. The heat supply was partly lost, and gradually recovered up to 90% of its initial level. Despite the flow being substantially decreased (to less than 15%), the increased temperature drop (from about 9 °C to 50 °C) recovered the heat supply in the radiator system. During the test, the radiator supply temperature displayed a gradual reduction when the natural circulation flow increased due to the primary supply temperature slightly decreasing.

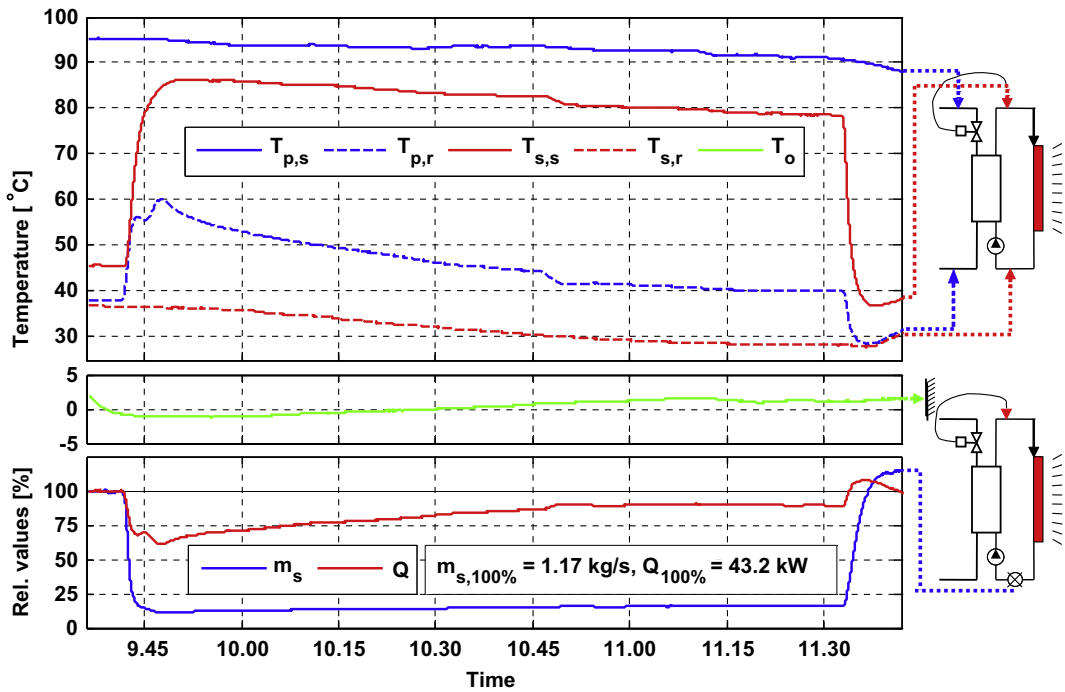

Fig. 2. A field experiment with natural circulation. The upper diagram shows the temperature levels in the HEX, where the black lines indicate the primary side and the grey lines indicate the secondary side. The solid lines represent the supply and the dashed lines correspond to the return temperatures. The diagram in the middle shows the outdoor temperature and the lower diagram displays the relative heat output (black) as well as the secondary circulation flow (grey).

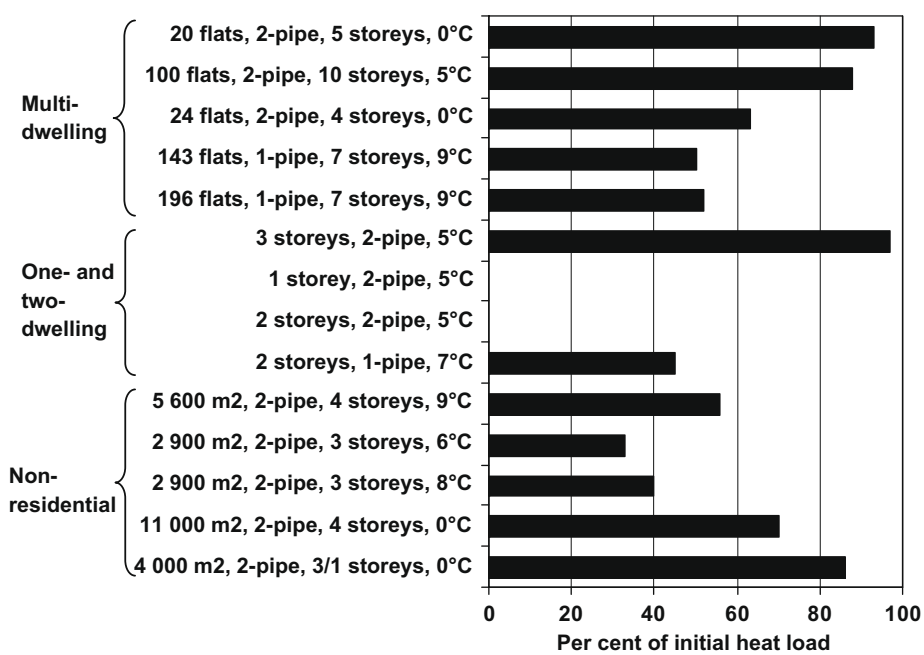

Fig. 3. Field tests of natural circulation in various buildings. The column shows the heat supply (in relation to its initial value) that was achieved during the test.

In order to verify whether the relatively low natural circulation flow was evenly distributed in the system, supply and return temperatures on four (evenly distributed) risers in the basement were measured. The temperature front reached all four risers, indicating that the heat distribution in the system was indeed uniform, although the actual flow was not measured.

To summarise, it can be stated that natural circulation functioned very well in this object, giving rise to almost the same level of heat supply as during normal operation. The remaining objects showed more or less the same pattern (i.e., a small circulation flow rate but a significant temperature drop), although most of them did not reach the same level of heat output. Various obstacles for natural circulation could be identified, which is described in the next section. A more detailed analysis is presented in reference [8].

In total, five multi-dwelling, four single-dwelling and five non-residential (three schools and two retirement homes) buildings

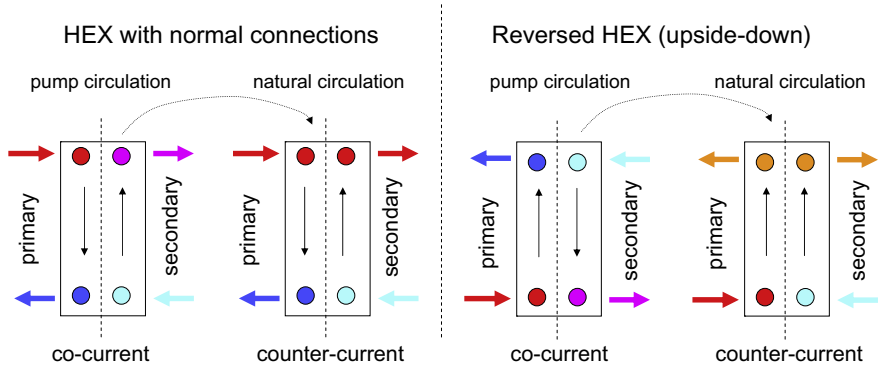

Fig. 4. The regular outline of a radiator HEX (left) as well as the reversed outline (right).

were investigated. Fig. 3 below shows the levels of heat supply which were reached in the tested buildings (in relation to the initial heat load). The tests were performed at outdoor temperatures between 0 °C and 9 °C.

3.3. Obstacles to natural circulation

The following section describes the most important obstacles to natural circulation in pump-driven radiator-heating systems that have been identified.

3.3.1. The DH substation

Most types of control valves stop in their current position and are thus not a direct obstacle to natural circulation. Valves with spring-return are generally used for dampers in ventilation systems (to avoid freezing) and for domestic hot water systems (to avoid scalding), however not in radiator systems. Still, electromagnetic actuators are naturally closed if they lose power. Nevertheless, contacts with manufacturers demonstrate that these actuators are generally much less common than their electromechanical and electro hydraulic counterparts. The Swedish District Heating Association has now, as a result of this work, included in their recommendations that control valves for heating stop in their current position [18].

One type of DH substation was found which had primary and secondary supply pipes connected at the bottom of the HEX instead of at the top. Although the latter is the more common alternative, the connection of the supply pipes has no practical significance for the normal operation of the substation. However, it does affect the natural circulation. As demonstrated earlier, the secondary flow was reduced and became very hot. Without the pump in operation, the hot water in the HEX would rise and leave at the top, which would result in a reversed circulation flow. Under such circumstances, the HEX would work as a co-current flow HEX instead of one with a counter-current flow, as demonstrated in Fig. 4. In a co-current HEX, the outgoing secondary supply temperature is limited by the outgoing primary return temperature. The heat exchange is thus less effective due to the temperature loss. Contact with the manufacturer shows that this type of substation is rather uncommon.

3.3.2. Heat distribution in the radiator system

The heat supply to buildings with reversed HEXs was found to be substantially reduced, but was still quite uniformly distributed. However, some of the other buildings received a more uneven distribution. Fig. 5 presents an overview of the heat distribution in two large objects, both built in the 1970's, with six stories and with 143 and 196 flats. In both objects, two buildings were supplied by a joint substation via long distribution pipes running through a garage or underground. Furthermore, both objects had 1-pipe systems with relatively narrow pipe dimensions and, consequently, high pressure drops. The heat supply was estimated to approximately 50% of the original values for both objects. As seen in the figure, approximately half of the systems could receive heat, implying that this half worked satisfactorily whereas the other half did not work at all.

The question is which of the factors that represents the biggest obstacle – the 1-pipe systems, the narrow pipe dimensions or the horizontally extended systems.

Since old radiator systems had very large pipe dimensions, one could fear that the higher flow resistance in newer buildings could impair natural circulation. However, other objects show that this does not necessarily have to be the case. This is not surprising if one considers the physical link between pressure and flow. In hydraulic systems, the pressure drop can be assumed to be proportional to the square of the flow, provided that the flow is turbulent [25]. (One should note that, even during natural circulation, the flow can generally be proved to be turbulent.) In most of the tested

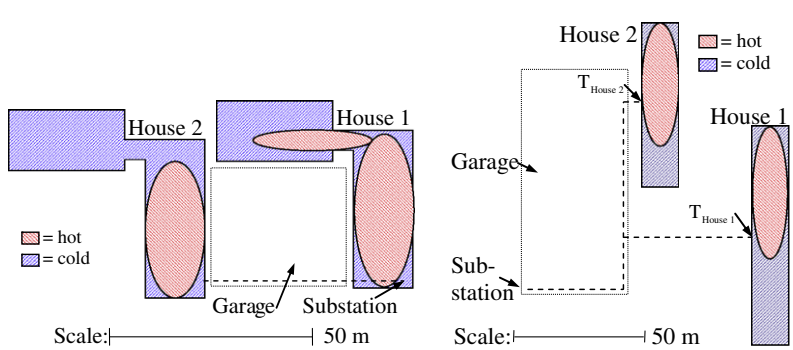

Fig. 5. An overview of two tested 1-pipe objects. The areas were natural circulation was established are indicated as 'hot'.

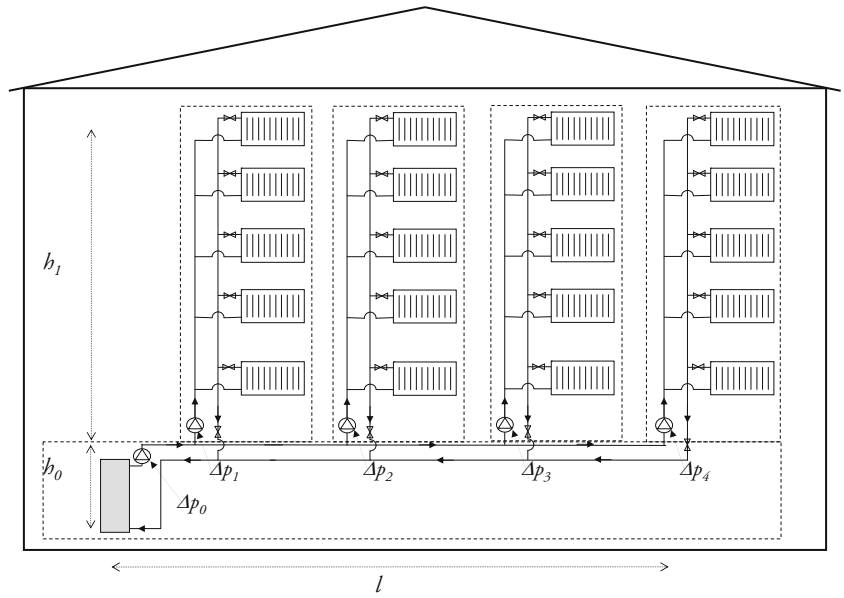

Fig. 6. A schematic drawing of pressure differentials in the system during natural circulation. The image indicates the height of the heat exchanger, h_0, the risers, h_1, and the horizontal extension of the system, l, as well as the virtual pumps due to density differentials.

objects, the natural circulation flow amounted to 10–15% of the normal flow rate. A tenth of the flow signifies that the pressure drop with natural circulation is only one hundredth of the normal pressure drop. Admittedly, the driving pressure for natural circulation was only 2–3% of the normal pressure,[1] but this shows that narrow radiator systems need not be as restrictive to natural circulation as they appear at a first glance.

1-pipe radiator systems were common during the 1960–1970's, due to an impatience to simplify and save pipe material. These systems are usually combined with narrow pipe dimensions and are found in large buildings. However, a 1-pipe system consisting of two circuits, one on the first floor and the other on the second floor, was also found in a single-dwelling house built in 1998. The substation was placed on the first floor, on a higher level than the bottom circuit and consequently no circulation occurred within it. On the second floor, the heat output was estimated to almost half of the total level before the test was started. However, almost all of the heat was emitted in the first two radiators (out of five). Risers that receive proper circulation in the larger 1-pipe systems seemed to function quite well, and since pressure drops become relatively small at low driving pressure, it thus seemed likely that the vertical and horizontal extension of the radiator systems was a more critical factor than age, pipe diameters, and the choice between 1- and 2-pipe systems.

A limitation for the experiments was that the tests could not continue more than a few hours in order not to jeopardise the comfort for residents or for activities in the buildings. In general, however, what can be described as stable conditions was achieved in the tested systems. Nevertheless, it is legitimate to consider what happens in systems where natural circulation is only established to a limited extent. Would it be possible for cold risers to receive natural circulation?

Important parameters include the ratio between the available driving pressure and the pressure drop in the horizontal distribution as well as the height of the risers and the pressure drop within them. Expressed in other words: the flow distribution is determined by the relationship between the height of the HEX, h_0, and height and the length of the system, h_1 and l, respectively. These variables are indicated in Fig. 6. At first sight, it can be assumed that the natural circulation benefits from high risers, since the driving force is proportional to the height. However, this need not be the case.

As a result of the density difference of water, one can say that the system has a small virtual circulation pump at all points where there is a height, i.e., a riser (provided that there is a temperature difference between the supply and return pipe) and the DH substation. This is demonstrated in Fig. 6. Such an effect is always present in the system, even during operation of the pump, but is negligibly small as compared to the pump head.

Let us consider an extreme system that has a very large driving height at the HEX, situated in a low building with a small bottom area. This is a system with conditions similar to normal pump operation, with the main differential pressure arising from the substation. Conversely, we can imagine a system with a low driving height in the basement circuit while the risers are very high. The heat may then display difficulties reaching the entire system. The influence of the "pumps" in the functioning risers are much more significant than that of the "pump" in the HEX, which means that it is actually possible to obtain a negative flow in one or more risers beyond the last functioning riser. The higher the altitude difference in the substation, the higher is the driving pressure in the horizontal distribution (basement circuit) and the farther out the flow will reach. Following the same reasoning, a horizontally extended building has a negative impact on the flow distribution. Risers with a high pressure differential tend to "suck" flow from subsequent risers.

The negative flow is in itself no real problem, but it is important to be aware of the fact that, in certain buildings, natural circulation functions perfectly well in some parts of the building, while other parts receive no heat at all. This should be kept in mind for buildings that are very high and/or horizontally extended.

3.3.3. Ventilation

A natural ventilation system provides no direct means of controlling the ventilation. One associated drawback includes that it may often present a low circulation rate in summer whereas it is (too) high during winter. In case of a power failure during cold

[1] The density difference is approximately 10 kg/m^3 at 65–45 °C, and assuming that the system is 10–20 m high, Eq. (1) gives a pressure difference of 1–2 kPa, which should be compared to a normal pump head of typically 30–100 kPa.

weather, these buildings run the risk of cooling down faster. To a certain extent, one can say that this is offset by the fact that mainly older buildings have natural ventilation, and that such buildings have generally shown good results in terms of natural circulation in the radiator system.

Forced ventilation systems will allow very limited air flows without power, resulting in a reduction in the buildings' heat losses. This is of course good, but also brings about an inferior air quality that, under the circumstances, is of less importance. Moreover, forced ventilation systems are generally equipped with dampers with spring-return to avoid freezing of air handling units.

Air handling units can be connected to the substation by, for instance, a separate ventilation circuit, supplying all air handling units, connected in parallel to the radiator circuit. This alternative does not give rise to any natural circulation since there is hardly any heat load or cooling of the water in this circuit. It is, however, quite common to connect both radiator and ventilation circuits to a joint circuit. The individual circuits are then connected to the main circuit using valve groups so that the desired temperatures and flows can be obtained. The most common type of valve group used with DH is shown in Fig. 7.

The main circuit is generally limited to the basement while the radiator and ventilation circuits reach the top of the building.

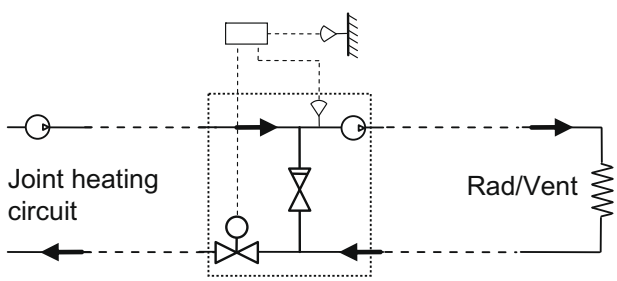

Fig. 7. The most common type of valve group used for DH-connected heating systems.

Therefore, the differential pressure in the latter circuit may be higher than the differential pressure in the former, but this naturally also depends on the temperature levels. It is then possible that the check valve in the valve group opens and allows part of the return flow to the supply pipe. In that case, the temperature to the secondary circuit becomes lower than the level in the main circuit. The results from a field test with a radiator circuit connected with a valve group are given in Fig. 8. It was likely that there occurred admixing of the return flow, which resulted in the reduced temperature. Nevertheless, the supply temperature sufficed for a certain amount of natural circulation to arise in this circuit.

3.5. General applicability

The field tests were conducted mainly in Malmö, in relatively mild weather, with temperatures ranging from 0 °C to 9 °C. The outdoor temperature is in itself not very important when it comes to examining how well the natural circulation works in a building. During all tests, the DH supply temperature remained well above 80 °C, which signified that the course of events (the high rise in the secondary supply temperature, etc.) was equivalent to what it would be at a lower outdoor temperature. Nevertheless, it was important to investigate which relative heat loads one can expect during cold weather. Computer simulations were performed using dynamic models built with the software Matlab and the associated toolbox Simulink. By verifying the model with field tests, it could be tuned to give rise to simulations resembling the experimental results. A more detailed description of the model can be found in a previous paper [7].

Figs. 9 and 10 show the results from simulations for two models representing two extremes regarding natural circulation, at various outdoor temperatures. Model 1 resembles a four-storied multi-dwelling building from the 1950's with a 2-pipe system, 20 flats, and model 2 provides simulations for two six-storied multi-dwelling buildings with a joint substation, 1-pipe system and 100 flats per house. The upper diagrams show the heat supply that can be achieved with natural circulation, relative to the original supply,

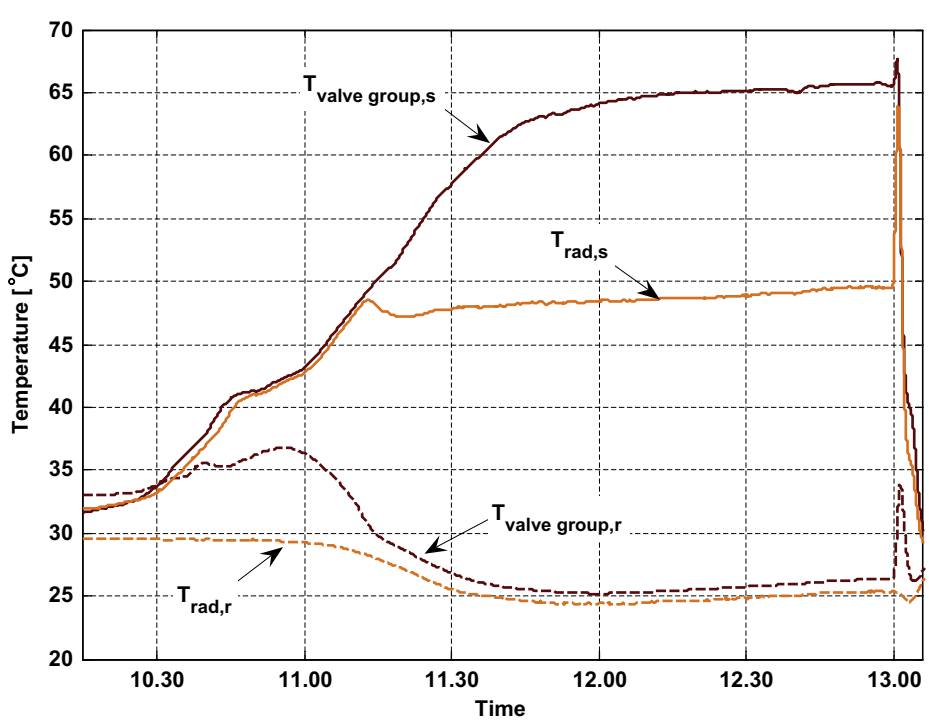

Fig. 8. Supply (solid lines) and return (dashed lines) temperatures measured on a valve group during natural circulation.

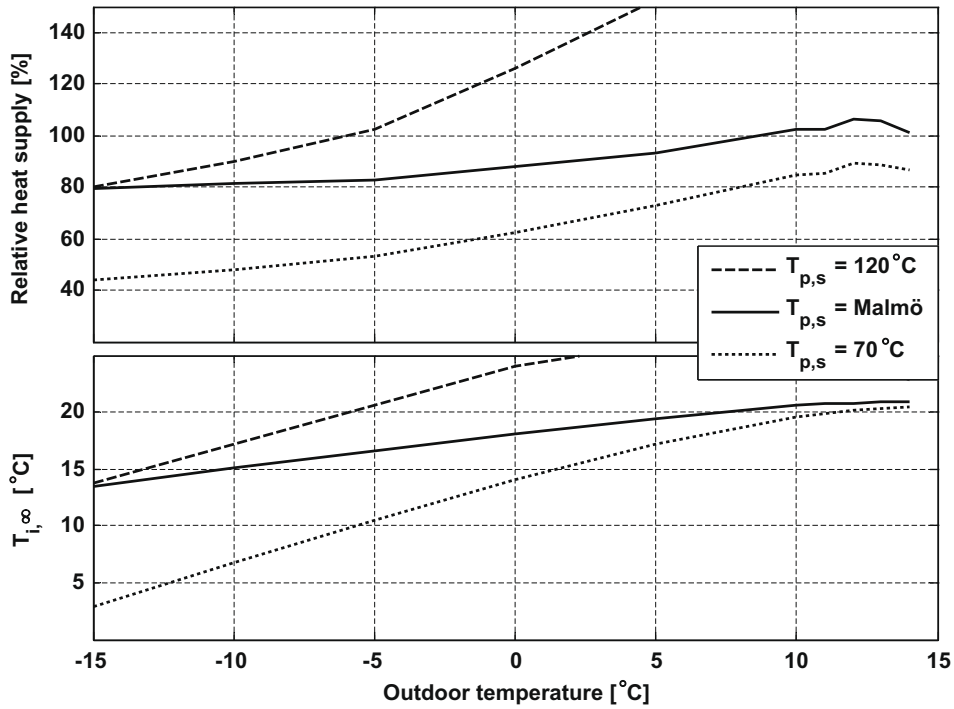

Fig. 9. The influence of variations in the primary supply temperature on the relative heat output and steady state indoor temperature, $T_{i,\infty}$, for various outdoor temperatures according to model 1.

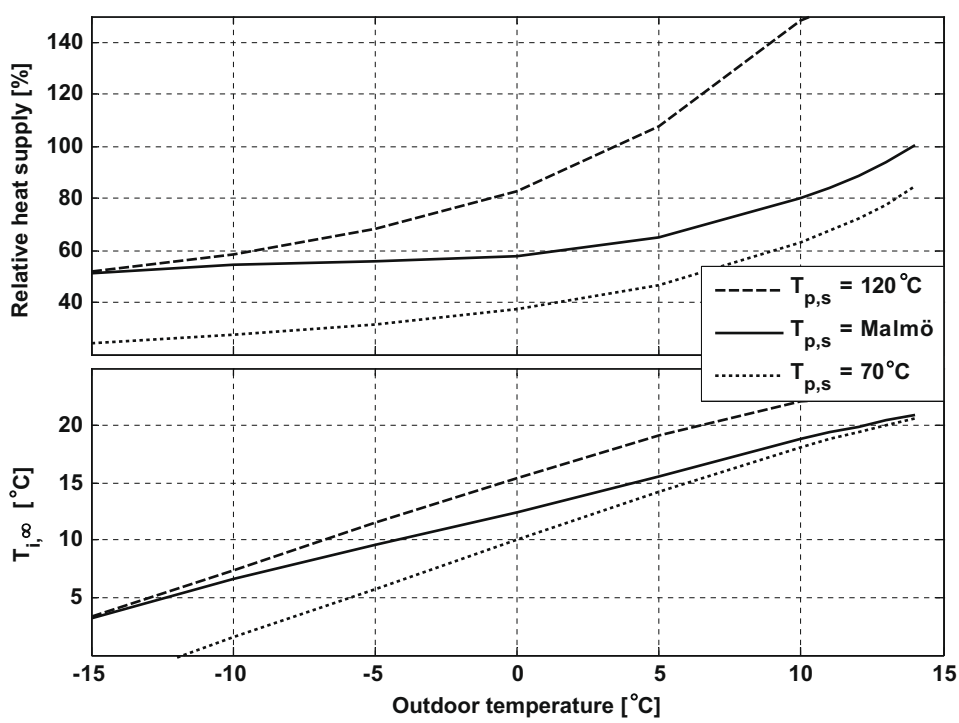

Fig. 10. The influence of variations in the primary supply temperature on relative heat output and steady state indoor temperature, $T_{i,\infty}$, for various outdoor temperatures according to model 2.

and the lower diagrams display the indoor balance temperature ($T_{i,\infty}$), i.e., the indoor temperature achieved after an infinitely long blackout. The ability to enhance the natural circulation by an increased DH supply temperature (T_{ps}) was also examined for each simulated outdoor temperature. The temperature was increased to 120 °C (which generally represents the maximum temperature used in Swedish DH systems) and was subsequently reduced to 70 °C for comparison. (One can also imagine that a lower supply temperature could be realistic in a case where only limited production resources are available during a power failure.) For the reference case, the supply temperature that is normally applied in Malmö's DH network, shown in Fig. 11, was used.

As opposed to model 2, model 1 provided much more favourable results for natural circulation. According to model 1, the

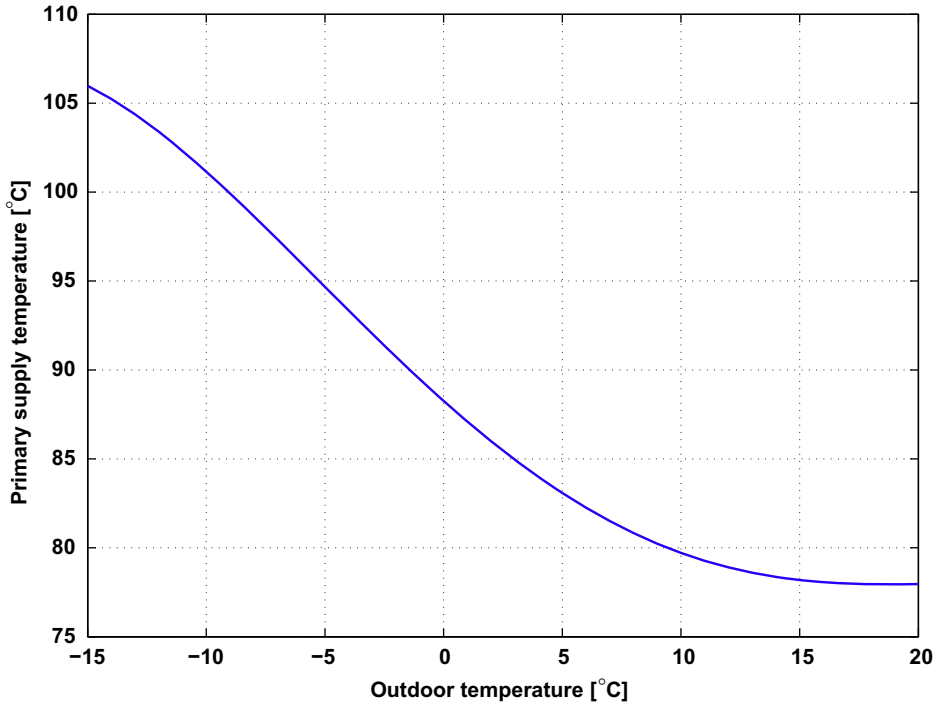

Fig. 11. The supply temperature in the DH network in Malmö.

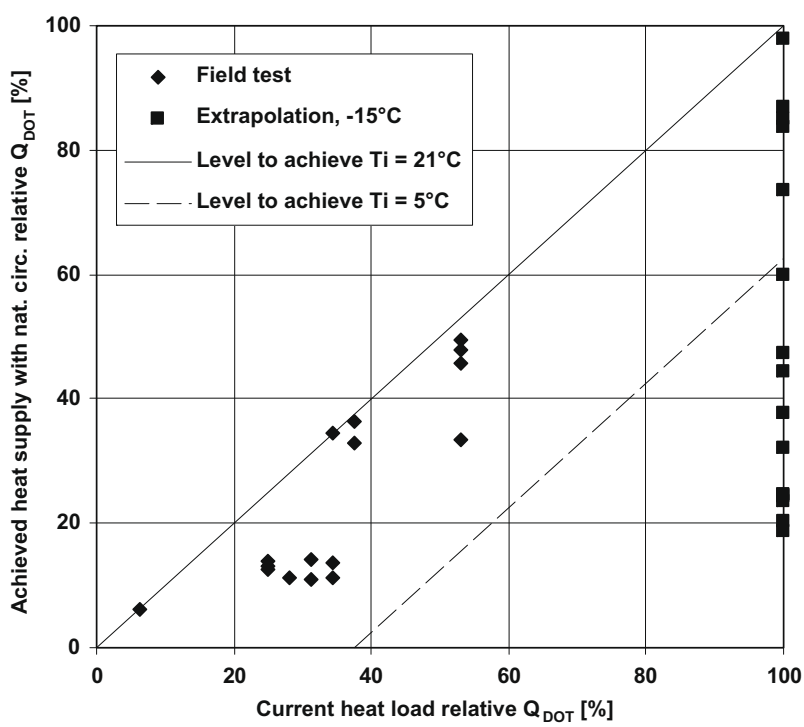

Fig. 12. Results from field tests of natural circulation including extrapolation for DOT. The solid line indicates the necessary relative heat supply in order to achieve an indoor temperature of 21 °C. The dashed line indicates the heat supply needed to achieve an indoor temperature of 5 °C.

building received 80% of the heat supply, provided that the supply temperature was maintained (i.e., normal for the present outdoor temperature). This means that the indoor temperature should never fall below 13 °C. The same temperature was attained with model 2 at an outdoor temperature of approximately 2 °C. Regarding the impact of the DH supply temperature on the indoor temperature, an increase of the supply temperature to 120 °C at an outdoor temperature of −5 °C signifies that the indoor temperature increases by 4 °C to over 20 °C according to model 1.

Fig. 12 presents the results in a slightly different way. The *x*-axis presents the current heat load relative the design heat load at DOT

(design outdoor temperature, −15 °C in Malmö). The *y*-axis shows the achieved level of heat supply with natural circulation relative the design heat load at DOT. The solid, diagonal line indicates the necessary relative heat supply in order to achieve an indoor temperature of 21 °C. The dashed line indicates the heat supply needed to achieve an indoor temperature of 5 °C. Authorities in Sweden [21] have identified this level as the lowest acceptable indoor temperature, with the exception of health and elder care activities, for which temperatures below 18–20 °C cannot be accepted. The performed tests are displayed by (♦) and as can be seen, all the tested objects achieved a heat supply resulting in an indoor temperature well above 5 °C.

To estimate the level of heat supply by natural circulation at DOT (100% heat load), extrapolated values (■) were calculated.[2] Approximately half of the objects now managed to reach an indoor temperature of 5 °C or more. However, several facts should be kept in mind: Due to the thermal inertia of the buildings, the indoor temperature does not immediately fall to the same level as the outdoor temperature. With a limited heat supply, the temperature in many buildings can be maintained several days before reaching a critical level. Also, the aim when selecting test objects was to capture various types of extremes, not to test a number of similar buildings that belong to the most common ones in the building stock. For example, two of the objects were equipped with the HEX-type with outgoing pipes at the bottom. Without this otherwise rare type of substation, the heat supply during the tests would likely have been significantly higher.

In the other objects that received a low heat supply, the heat was generally unevenly distributed in the sense that it was more or less normal in one part of a building whereas the other part was left without any heat supply at all. During an emergency, a reallocation of users in these buildings would likely be sufficient to avoid an evacuation.

4. General guidelines to enhance natural circulation

The present work has resulted in a number of recommendations addressed to all concerned parties, i.e., authorities, DH utilities and manufacturers, property owners, operators and residents [10]. The Swedish District Heating Association has modified their recommendations to include that control valves for heating stop in their current position [18].

For the DH utility, it is essential to have back-up power for production and distribution of heat. It is beneficial for the natural circulation to have a supply temperature as high as possible. How high it can be must, however, be decided by each DH utility depending on their system's prerequisites.

The recommendations also include guidelines for house owners how to test or estimate the possibilities for natural circulation in a building as well as measures of how to enhance natural circulation if a power failure occurs (e.g., by opening closed control valves).

5. Conclusions

During a power failure, there exist good opportunities for heat supply by natural circulation in DH-connected space heating systems. The vast majority of control valves remain in their positions and DH water can thus continue to pass through the space heating HEXs. Nonetheless, this is under the condition that the DH network can be operated during a power failure, i.e., that the DH utility has back-up power for production and distribution of heat. In addition, certain customers who have their own back-up power supplies, such as hospitals, depend on the DH network for their heat supply.

The obtained results demonstrate that natural circulation can arise in the vast majority of buildings, thus supplying heat equivalent to 40–80% of the current heat supply at low outdoor temperatures. This signifies that it should be possible to cope for several days before a possible evacuation becomes necessary.

Recommendations have been compiled, designed to increase possibilities for natural circulation in different systems, and have been addressed to all concerned parties. The two most important recommendations are: (1) for the DH utility to ensure that they have back-up power for the production and distribution of heat and (2) for building owners to ensure that the control valve for the radiator HEX is not closed in case of a power failure. The first point is essential in order to be able to distribute DH at all, and the second point is essential in order to achieve natural circulation in the radiator systems.

Acknowledgements

The Swedish Energy Agency, the municipality of the town of Malmö, Stadsfastigheter (owner of municipal properties in Malmö), MKB Fastighet AB (owner of a large share of multi-dwelling buildings in Malmö), the Swedish District Heating Association and E.ON Värme Sverige AB (owner of the DH network in Malmö) are gratefully acknowledged for financing this work.

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IMPROVED COOLING OF DISTRICT HEATING WATER IN SUBSTATIONS BY USING ALTERNATIVE CONNECTION SCHEMES

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Abstract. *In order to gain thermal efficiency in a district heating (DH) system it is important that the return temperature from the connected buildings be kept as low as possible. When using indirect DH connection, the choice of connection scheme in the substation affects the DH return temperature. For example, in multi-dwelling buildings the so-called 2-stage connection scheme is most commonly used. Traditionally, the 2-stage connection scheme is claimed to increase the cooling of DH water. However, the gain when used with modern space heating systems is moderate or non-existent, due to a shift from higher to lower temperature levels in the heating system, e.g. from 80/60°C to 55/40°C or even lower. To improve cooling of DH water, an alternative type of connection scheme, termed series connection, is suggested for buildings with domestic hot water (DHW) circulation. The gain from this connection is even larger for non-residential buildings where the DHW consumption is smaller.*

In the series connection scheme, the heat exchanger (HEX) for DHW provision is connected before the HEX for the space heating system. The DH return temperature from the DHW HEX is, for most of the year or, in some cases, always, higher than the temperature levels in the space heating system, when no tapping occurs and only re-circulation of DHW prevails.

In order to verify the gain in cooling of DH water, simulations have been performed using well-tested computer models of substations, including both space heating load and DHW consumption patterns. The magnitude of the gain from the series connection scheme is dependent on, among other things, the size of the building. In a building with a conventional radiator system, the gain from the series connection scheme in some cases estimates to several degrees C on the yearly average return temperature. The lower the temperature levels in the space heating system, the larger the gain from the series connection. This is a fact that agrees with the increased popularity of low temperature heating systems, such as floor heating.

Keywords*: district heating, low return temperature, substation connections*

1. INTRODUCTION

For many years, return temperatures in district heating (DH) networks have been an important issue for DH research. Many types of production units, as well as DH networks, benefit from low system temperatures. In Sweden, DH is the most common heating source for multi-dwelling and non-residential buildings, with a market share of more than 80%. The buildings in Sweden are, as well as in many other countries, connected via so-called indirect connection to the DH network, i.e. the DH network (primary side) and the house-internal systems (secondary side) are hydraulically separated by heat exchangers (HEX). This study will focus on indirect connections.

Space heating and provision of domestic hot water (DHW) is characterised by a variation of temperature levels which provides a basis for lowered DH return temperature (increased cooling of primary water), by employing cascading of HEXs in substations.

With modern low-temperature heating systems, state-of-the-art substations with either parallel (also referred to as 1-stage) or 2-stage connection schemes are not always constructed in a way that is optimal for the cooling of DH water, i.e. achieving the lowest possible primary return temperature for a given primary supply temperature. In this paper, two alternative connection schemes are investigated. Traditionally, radiator systems in Sweden have been designed for a temperature program of typically 80/60°C (secondary supply/return) or 60/40°C at design outdoor temperature (DOT). However, previous works show that oversizing of heating systems is common, meaning that a compensation of either a reduced supply temperature or a reduced mass flow (so-called low-flow system) must be applied in order to achieve the desired indoor temperature.

Well-insulated windows with reduced convection have led the way for building-integrated heating systems such as underfloor heating, working at drastically lower temperature levels. Underfloor heating has gained large popularity in detached houses, but is likely to become more common also in multi-dwelling buildings, not only in new buildings but also in older buildings, e.g. when renovating bathrooms. Today, many bathrooms are instead equipped with electric underfloor heating.

1.1. Objective

The objective with this study is to find out weather the state-of-the-art connection schemes are optimal for lower secondary temperatures, with respect to lowest possible DH return temperature. They will be compared with two alternative connection schemes.

1.2. Limitations

Instantaneous water heaters are assumed, since this type of heaters are quite common, especially in bigger installations, where load aggregation inside the building reduces peak values of hot water loads (the reason why the hot water system does not have to be dimensioned for a peak flow as a high as would be the case if adding the expected peak flow in all units in the system). Indirect DH connection has already been mentioned, i.e. HEXs provide hydraulic separation between DH network and building-internal systems. When simulating DH substations with cascading, the area relation between pre- and after-heater (PH and AH, respectively) is set to 60 and 40%.

2. CONSTRUCTION OF DH SUBSTATIONS

Figure 1 below shows two common types of connection schemes for DH substations: parallel (or 1-stage) and 2-stage connection. In the figure, simulated median temperatures for a three-day period are represented for an outdoor temperature of 8°C (the average annual temperature in Malmö, Sweden) and a radiator system designed for a 60/40°C temperature program. The space heating and DHW systems are designed for 60 flats. The HEX for space heating is designed for a temperature difference of 3°C in the cold end of the HEX and the DWH HEX for 12°C, respectively, according to general guidelines [1], [13].

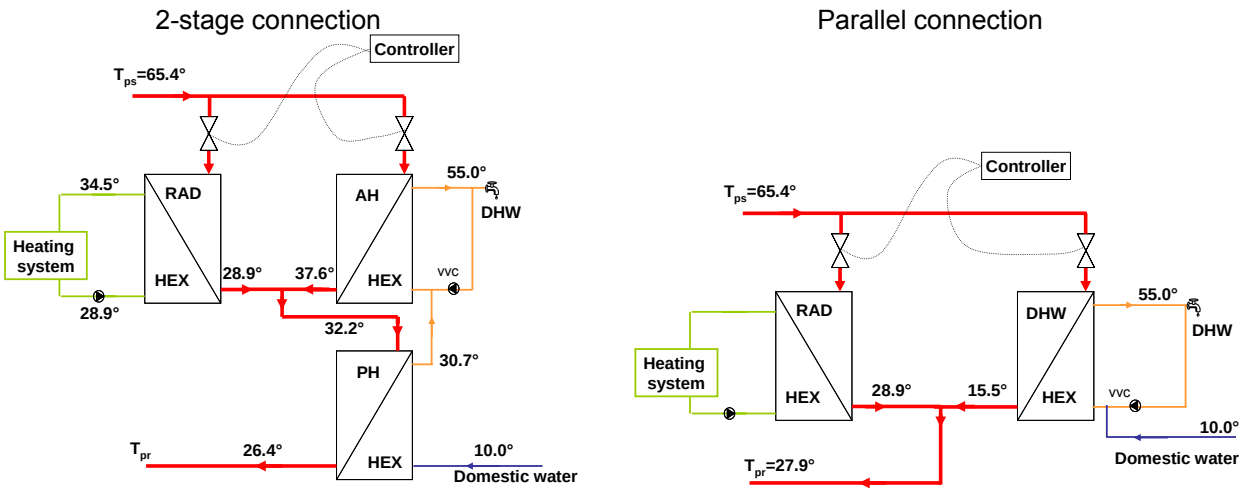

Figure 1. 2-stage (left) and parallel (right) connected DH substations with median temperatures level at T_{out} = 8°C and 60/40°C design temperature (60 flats). Temperatures are given in °C.

In some parts of Sweden, a 3-stage connection scheme is still rather common, but is generally not installed any longer. At low outdoor temperatures, the DHW becomes overheated in the AH. A subsequent shunt valve reduces the temperature, but if the town's water is too hard, there is a great risk of scaling of the DHW HEX.

A variant of the 3-stage connection scheme, termed Russian 3-stage connection was proposed by our research group in [10] and [7]. The primary temperature after the AH is typically around 50°C when no DHW is tapped and around 35°C when DHW is tapped. Given that the supply temperature to a heating system, at least for low-temperature systems, mostly is below this temperature level, it would be wise to connect the radiator HEX in series, *after* the AH, se Figure 2. The re-circulated DHW temperature should not be less than 50°C, due to the risk of legionella growth, which means that the DH water leaving the AH or DHW HEX is always hotter than 50°C when no DHW is tapped.

The matter of various connection schemes in DH substations has been dealt with by different authors over the years. Frederiksen & Wollerstrand [2] showed that, when instantaneous water heating was assumed, the gain in return temperature with 2-stage connection is rather small compared with parallel connection on a yearly average. Gummérus [4] by simulations supported this finding, which in turn was supported by derivation of analytical formulae for return temperatures and by laboratory experiments by Frederiksen et. al [3]. Results derived by Volla et. al. [14] were somewhat more encouraging for cascading in configurations when fan coil heating was added to a hydronic radiator system. Later findings by Snoek et. al. ([10] & [11]) for similar configurations yielded smaller gains, the difference in

the results from the two investigations largely to be explained by differing practices in induced hot air temperature level.

In [10], Frederiksen on theoretical grounds advocated that the simple parallel connection scheme is rather inferior to a parallel connection where re-circulated DHW is entered after a PH. In [6], simulations provided numerical support for this view. Intuitively, and from an exergetic point of view, it is in fact rather obvious that mixing incoming, cold town's water with much warmer re-circulated hot water represents a substantial thermodynamic loss. Nevertheless, the simple parallel connection scheme is generally shown by the Swedish District Heating Association [13]. Therefore, in practice, it is probably quite commonly used.

2.1. Alternative connection schemes

A further proposition was made by Frederiksen in the work [10], supported by analytical and graphical derivations: When a substantial amount of re-circulated hot water is supposed, in combination with a low-temperature space heating system, there is a good thermodynamic case for adopting 3-stage connection instead of 2-stage. A modified variant of the 3-stage connection arrangement was recommended as a general solution, rather than the traditional Swedish type, since the alternative arrangement does not suffer from the previously mentioned drawback of sensitivity to hard town's water, i.e. the scaling problem. This connection resembles a scheme found in Russian district heating literature, e.g. in the textbook [12] by Sokolov.

In the paper, this 3-stage connection scheme will from now on be denoted R3-stage, where the 'R' in the designation stands for 'Russian'. It will be compared with another alternative connection scheme, along with the conventional parallel and 2-stage connection schemes, with respect to which of them gives the lowest primary return temperature. The R3-stage connection scheme is shown in Figure 2 below. The other alternative connection scheme has been termed "series connection" and is shown in Figure 3. In both connection schemes, the primary DH water used for DHW circulation is directed into the radiator HEX during parts of the time.

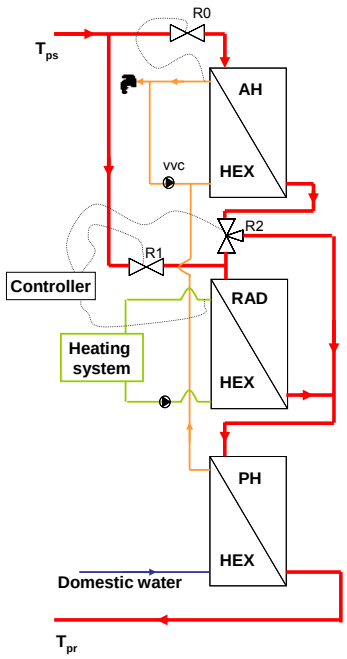

Figure 2. R3-stage connected DH substations.

The idea with the R3-stage connection scheme is to cool the DH water as efficiently as possible by taking different paths through the substation. We explain by looking at control strategies at different operating cases:

Let us follow the incoming primary mass ($\dot{m}_p$) flow through valve R0 to the AH HEX. R0 operates independently of the space heat load and keeps the DHW temperature at the desired level by controlling $\dot{m}_p$ through the AH. After leaving the AH, the $\dot{m}_p$ continues through the RAD HEX, during heating season. If the desired heating system supply temperature level is not reached, valve R1 opens and allows more $\dot{m}_p$ to mix with the $\dot{m}_p$ leaving the AH. In order to avoid overheating of the space heating system, R2 opens and allows a bypass flow from the AH directly to the PH when valve R1 is closed. Note that all $\dot{m}_p$ passes through the PH. Below, these control strategies are explained in a different way:

- $T_{ss} < T_{ss,setpoint}$ & R2 = 0 (bypass closed) → R1 opens (increased $\dot{m}_p$ to RAD HEX through R1)
- $T_{ss} < T_{ss,setpoint}$ & R2 ≠ 0 (bypass open) → R2 closes (decreased bypass flow from AH through R2)
- $T_{ss} > T_{ss,setpoint}$ & R1 ≠ 0 ($\dot{m}_p$ to RAD HEX through R1) → R1 closes (decreased $\dot{m}_p$ to RAD HEX through R1)
- $T_{ss} > T_{ss,setpoint}$ & R1 = 0 (no flow through R1) → R2 opens (increased bypass flow from AH through R2)

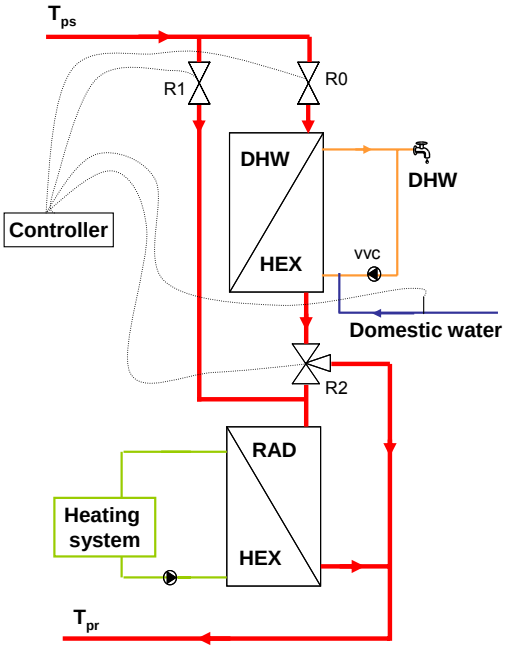

Figure 3. Series connected DH substation.

The idea with the series connection is the same as with the R3-stage connection, i.e. to utilize the DH water's rather high temperature after the DHW HEX when there is no tapping. Both the R3-stage and the series connection schemes require more sophisticated control equipment than the conventional connection schemes. However, the latter connection scheme is a simplified version of the previous one. In contrast to the R3-stage connection, the series connection only varies between two operating modes, depending on whether DHW is consumed or not.

The primary DH water supplying the DHW HEX is always led into the radiator HEX when no DHW tapping occurs. When necessary, e.g. during high heat load when the primary flow from the DHW HEX is insufficient, additional primary flow is mixed by a bypass connection supplying the HEX with more DH water. When DHW is tapped, the valve R2 will open, and the substation will work as a regular parallel-connected substation and the radiator HEX is supplied with DH water through the bypass connection.

As already mentioned, the parallel connection scheme can be designed in two different ways: generally there is only one DHW HEX, as shown in Figure 1, but the DHW HEX can be divided into two parts, referred to as PH and AH. The influence of this difference will also be studied, both for the parallel and for the series connection schemes. One can say that the reason for showing two heat exchanger design cases for the parallel connection scheme is that it is a matter of taste to say which design case provides the fairest basis for comparison between the various connection schemes.

3. MODELLEING THE DH SUBSTATION AND HEATING SYSTEM

The mathematical description of substation and heating system model is based on well-documented models of HEX, control equipment, actuators and valves. The theory and function of these components have been described in detail by a number of authors, for example [4], [5], [8] and [9]. The components are combined into a model of a DH substation in the software Simulink.

3.1. Prerequisites

Patterns for DHW tapping (tap frequency and tap length) are simulated based on statistical measured data for different number of residential flats. The tapping patterns are simulated based on a program described in [15]. The heat load is assumed to be 3 kW per flat at design outdoor temperature, T_{DOT}, (assumed to −15°C) and zero at T_{out} = 17°C. The design DHW load is based on recommendations from the Swedish DH Association [13]. The temperature levels in the DH network are based on the design temperatures given in [13]. Heat losses from the DHW circulation circuit to the

building is based on a temperature drop of 5°C and an energy loss of 0.1 kW per flat. These assumptions are shown in Figure 4. The diagram on the left also includes the duration of the outdoor temperature in Malmö.

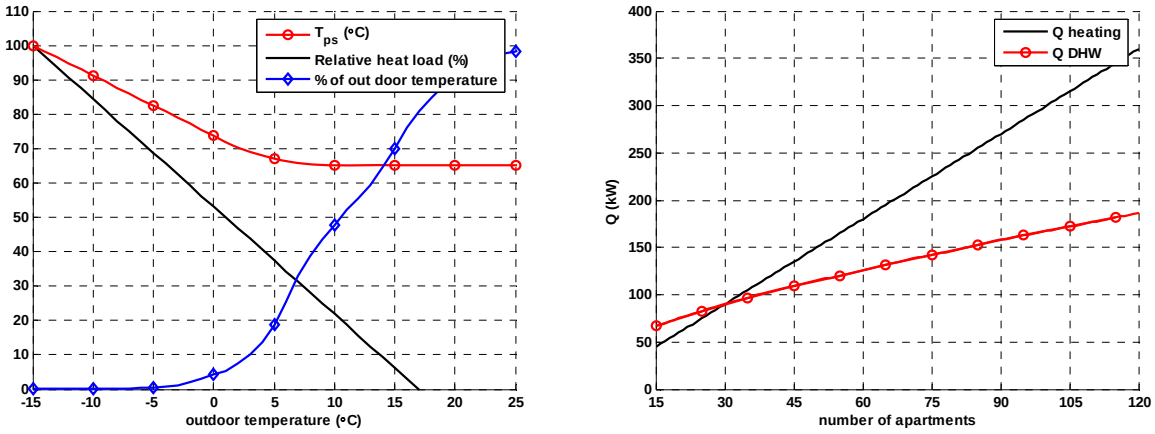

Figure 4. DH supply temperature and relative space heat load as a function of the outdoor temperature along with the outdoor temperature duration on the left and design heat load for space heating and DHW as a function of the number of apartments on the right.

The primary return temperature is simulated for four different radiator system temperatures: 60/40, 75/35, 55/45 and 40/30°C, respectively. The 40/30°C program is a low-temperature radiator program that can be assumed typical for modern buildings. For residential buildings, five different sizes are simulated: 15, 30, 60, 90 and 120 flats, respectively. The different temperature programs for the heating systems are shown in Figure 5.

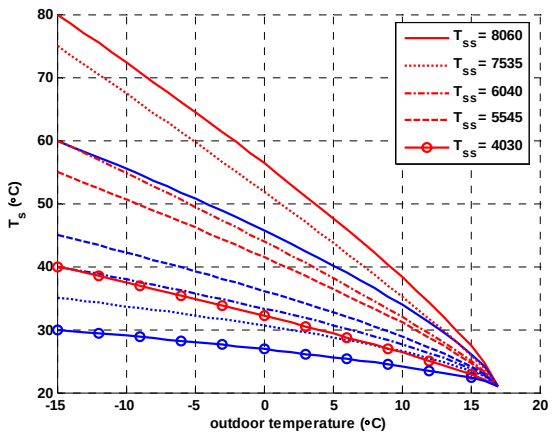

Figure 5. Different temperature programs for the heating system. Blue lines show corresponding return temperatures.

Primary flow-weighted average return temperatures for one year are simulated for the different combinations of DH substation schemes and radiator temperature programs, see equation (1). The temperature is weighted with the DH flow rate.

$$\overline{T}_{pr} = \frac{\sum^{T_{out}} T_{pr} \cdot \dot{m}_p}{\sum^{T_{out}} \dot{m}_p} \tag{1}$$

4. RESULTS

4.1. Residential buildings

Figure 6 shows the primary annual average return temperature for residential buildings. Each diagram shows results for different connection schemes for a specific radiator temperature program as a function of the number of flats. As seen in the figure the simple parallel connection is inferior to the other connection schemes for all temperature programs. However, a parallel connection scheme with DHW preparation divided into PH and AH can, for heating

systems designed for 60/40, 40/30 and 75/35°C, compete with the 2-stage connection. The R3-stage connection scheme gives the lowest return temperature under all conditions. In newer buildings with low-temperature space heating, the R3-stage connection could increase the cooling of DH water with 1.5-3°C (compared with a 2-stage connection) depending on the size of the building. 2-stage connection is often chosen instead of the cheaper parallel connection in order to reduce T_{pr}. Our study shows that this only makes sense if the DHW preparation is not divided into PH and AH. Regarding the series connection, the following can observed: it gives approximately the same return temperature as the 2-stage and parallel with PH/AH connections, except for the 55/45 program (and to some extent 60/40) were the 2-stage connection is slightly better. However, the series connection with PH/AH is the second best choice after the R3-stage connection under almost all conditions.

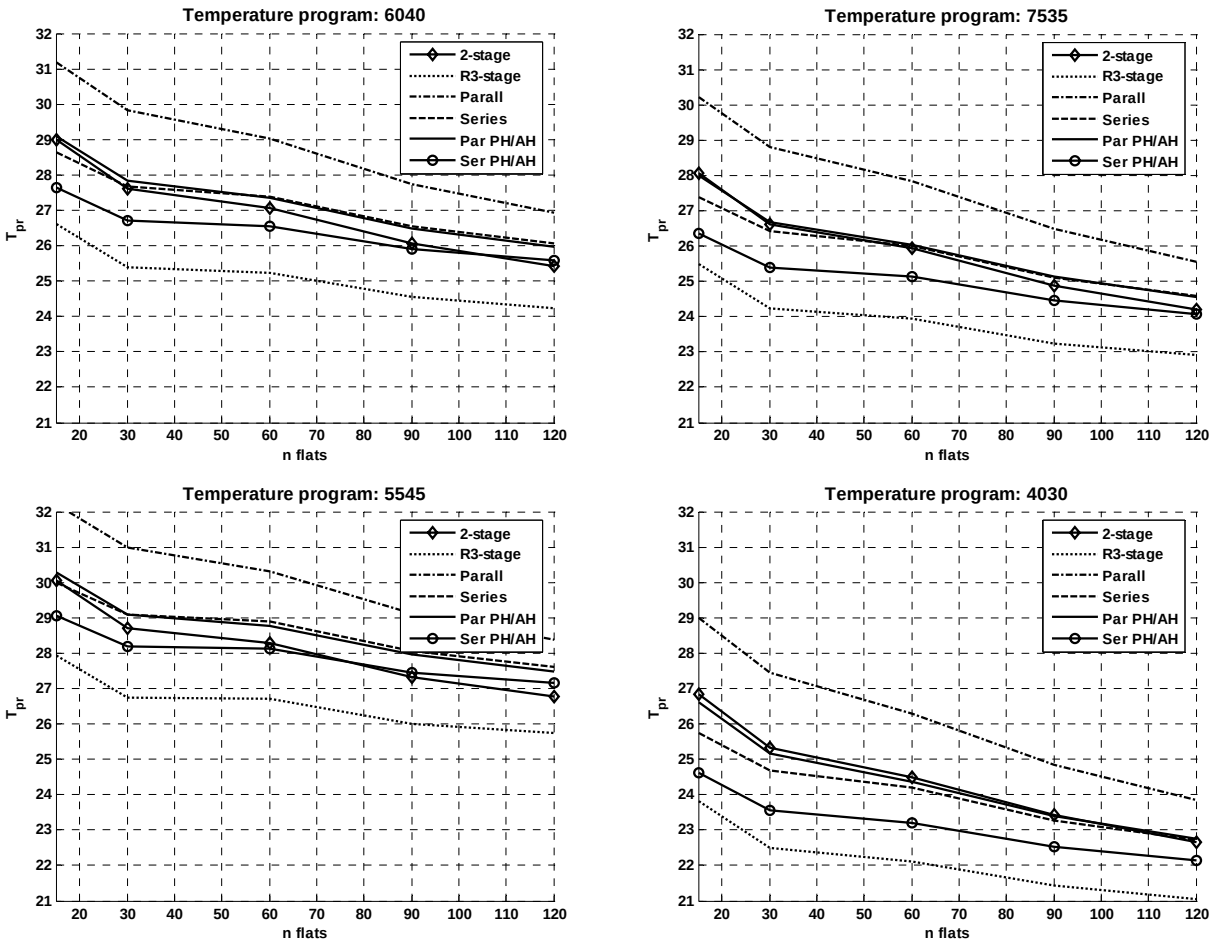

Figure 6. T_{pr} for residential buildings. Each diagram shows results for different connection schemes for a specific radiator temperature program.

Table 1 shows the results from Figure 6 in a different way. The 2-stage connection scheme is used as reference and in the column for the other connection schemes are indicated whether they are either better, worse, or equal.

Table 1. Results from simulations. A reduction of T_{pr} >1°C is indicated by '−', a reduction between 0.3-1°C '(−)'. An increase in T_{pr} of > 1°C is indicated by '+', an increase between 0.3-1°C '(+)'. Changes less than 0.3°C are indicated by '~'.

	temperature program	2−stage	R3−stage	Parallel	Series	Parallel with PH/AH	Series with PH/AH
15	8060	34.3	−	+	(+)	(+)	~
	5545	30.1	−	+	~	~	(−)
	6040	29.0	−	+	(−)	~	−
	7535	28.0	−	+	(−)	~	−
	4030	26.8	−	+	−	~	−
60	8060	33.2	(−)	+	+	(+)	(+)
	5545	28.3	−	+	(+)	(+)	~
	6040	27.1	−	+	(+)	(+)	(−)
	7535	25.9	−	+	~	~	(−)
	4030	24.5	−	+	(−)	~	−
120	8060	32.0	(−)	+	+	+	+
	5545	26.8	−	+	(+)	(+)	(+)
	6040	25.4	−	+	(+)	(+)	~
	7535	24.2	−	+	(+)	(+)	~
	4030	22.7	−	+	~	~	(−)

4.2. Non-residential buildings

There are obviously many buildings connected to DH that are used for non-residential purposes. The use of DHW generally differs quite a lot compared to residential buildings, depending on the type of activities. For example, in an office building there is little DHW consumption during evenings, nights and weekends and the consumption during working-hours is probably much smaller than in a residential building of the same size (less showers, cooking etc.) Therefore, simulations with a reduction of the number of DHW tappings with 25 and 50%, respectively, were performed. The size of the tap flow is not changed, which means that the DHW HEX is designed for the same flow rate as for a residential building with the same space heat load, as are the heat losses for DHW circulation. Simulations were made for buildings dimensioned for 180, 270 and 360 kW heat load, respectively (corresponding to 60, 90 and 120 flats, respectively). Simulated radiator temperature programs are 60/40, 55/45 and 40/30°C, see Figure 7.

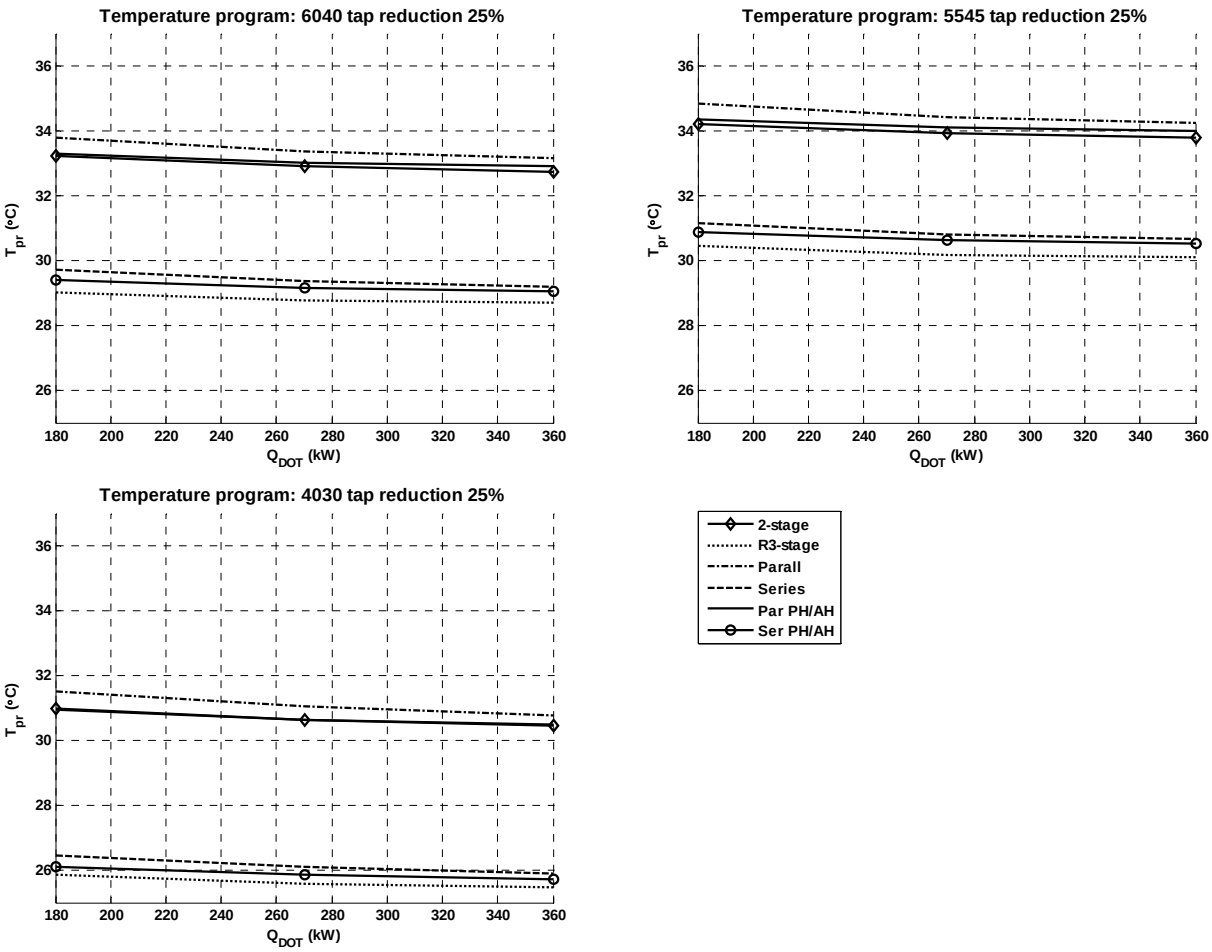

Figure 7. Three different temperature programs with a reduction of the number of tapping with 25%.

As seen in the figure, the 2-stage connection scheme and the parallel scheme, both with and without DHW preparation divided into PH and AH, give about the same primary return temperatures. However, the series connection (both with and without division into PH/AH) and the R3-stage connection gives drastically reduced return temperatures; the difference is in the order 3-4°C depending on the radiator temperature system.

With a reduction of the amount of tapping with 50%, the gain with the series connection and the R3-stage connection is even better; 4-6°C, see Figure 8.

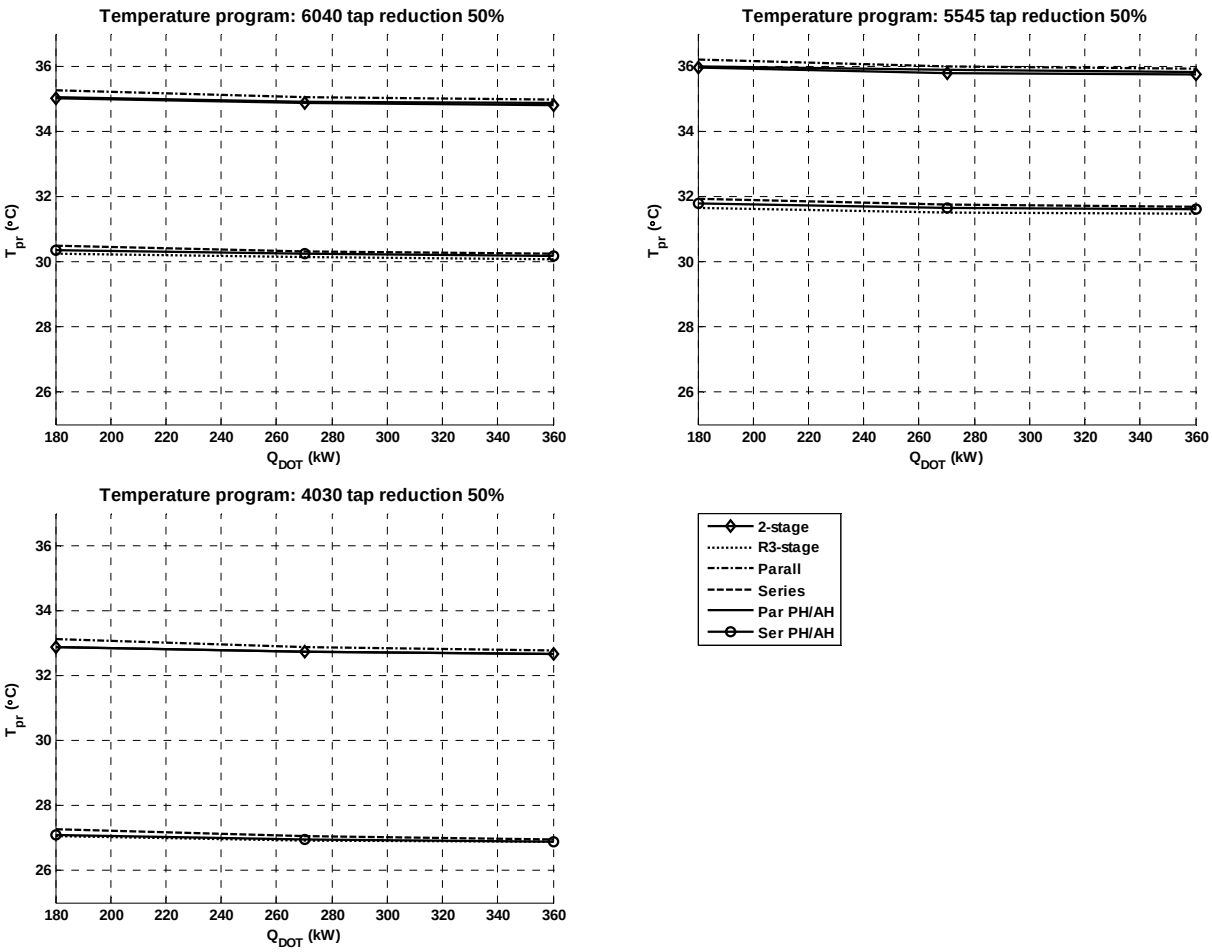

Figure 8. Three different temperature programs with reduction of the number of tapping with 50%.

7. DISCUSSION AND FURTHER STUDIES

DH substations that increases the cooling of DH water is favourable to the DH utility and in many cases also for the DH customer when DH prices are based on both energy consumption and cooling of DH water.

At all load conditions, the simple parallel connection scheme is the poorest. The R3-stage connection scheme always gives the lowest primary return temperature. However, this connection scheme demands a more sophisticated control. The commonly used 2-stage connection scheme is not always the best choice with respect to lowest possible return temperature. In some cases, a parallel connection scheme using the same HEX layout as the 2-stage connection, i.e. with PH and AH and the DHW circulation connected in between, gives approximately the same return temperature. The series connection gives a lower the return temperature compared to parallel connection and lower than the 2-stage connection in smaller residential buildings. With division of DHW into PH and AH, the series connection gives a lower return temperature than the 2-stage connection.

For non-residential buildings with less DHW consumption, the series connection and the R3-stage connection gives DH return temperatures in the same order, several degrees lower than the traditionally used connection schemes. In these cases the influence of dividing the DHW reparation into PH and AH is negligible.

ACKNOWLEDGEMENTS

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INFLUENCE OF DISTRICT HEATING TEMPERATURE LEVEL ON A CHP STATION

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Abstract. *The primary return temperature from district heating (DH) connected buildings is in focus for many reasons. A reduced primary return temperature reduces the velocity in the DH pipes which leads to decreased electricity usage for pumps and a higher capacity in the DH-network. The reduction of primary return temperature also influences the heat production units. In order to increase primary energy efficiency it is important to analyse how different DH temperature levels affects the production units. In Sweden DH is the dominating heating source in multi-family housings. With low temperature heating system (such as e.g. floor heating) connected to DH through a heat exchanger (HEX) the primary DH return temperature can be reduced dramatically. These low temperature systems can be introduced as the main heating source in new buildings, or as a complement in buildings when renovating. The benefits for the production units are calculated for a bio-fueled combined heat and power production (CHP) unit. In a CHP station both heat and electric power are produced simultaneously at high efficiency.*

The amount of heat produced in CHP stations is today over 30% of total heat input to DH in Sweden, and is believed to increase due to electricity pricing and CO_2 emissions reduction strategies in the EU.

Calculations of the heat and power station are based on an in-house advanced off-design model within the software package IPSEpro. The model is verified with data from a Swedish heat and power station in Enköping. The calculations show how a changed DH temperature is affecting the efficiency of the heat and power station and the electric power-to-heat ratio (α-value).

Keywords*: district heating, primary return temperature, primary supply temperature, electric power-to-heat ratio, CHP*

1. INTRODUCTION

In Sweden district heating (DH) is the dominating heating source. The heat market share for DH is over 75 % for multi-residential buildings, almost 60% for non-residential buildings, but only 10% for detached houses were electric heating is the most common heat source [10]. Since a DH system is an isolated energy system the fuel mix in different DH network differs from each-other mainly due to local conditions, e.g. access to waste heat from industries. In Sweden, as well as in other countries, DH from CHP stations is an important topic. By producing DH in a CHP station both heat and electricity is produced with high total efficiency. If the CHP station, as in this study, is fueled by bio-mass the electricity produced will contribute to reduce the amount of unwanted greenhouse gas emissions from e.g. coal-based power production units.

The green electricity certificate system, that was introduced in Sweden in May 2003, favours bio fuel-fired CHP generation. Due to this regulatory instrument, bio fuel-based CHP generation is in general the most profitable alternative for DH companies when constructing new heat generation [9]. The aim of the green electricity certificate system is to increase the share of electricity produced by renewable sources. Since the taxes for electricity produced in a CHP station are very favourable, the electric power-to-heat ratio (α-value) is of great interest.

Figure 1 shows the trend for electricity produced in CHP stations and the share of heat produced by CHP in DH in Sweden.

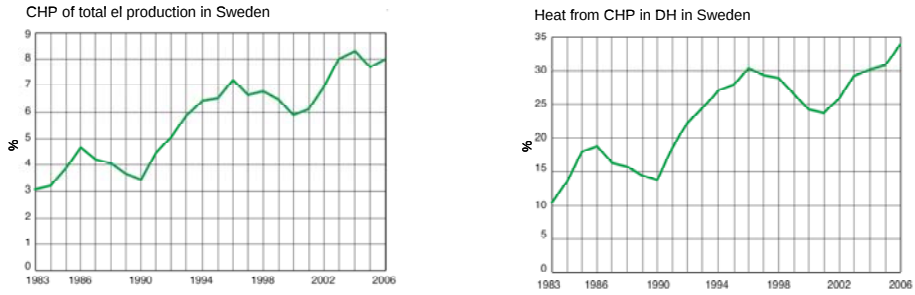

Figure 1. CHP in Sweden [9]

This paper will address how DH temperature levels are affecting the power production in a CHP-based DH network using data from Enköping DH network and CHP station as input. The calculations of the CHP station are made with a fully validated off-design model in the heat and mass balance software IPSEpro. The DH network is, in this study, just considered as a heat load with constant heat losses.

2. DISCRIPTION OF CASE STUDY OBJECT

When a CHP station is used as the main heat input in a DH network, the heat demand decides the possible production level of electricity. In order to produce as much electricity as possible it is of interest to study the electric power-to-heat ratio, that describes the relationship between produced electric power and heat in a CHP station, during different heat load situations. Figure 2 illustrates the load situation and operating production units in Enköping DH network.

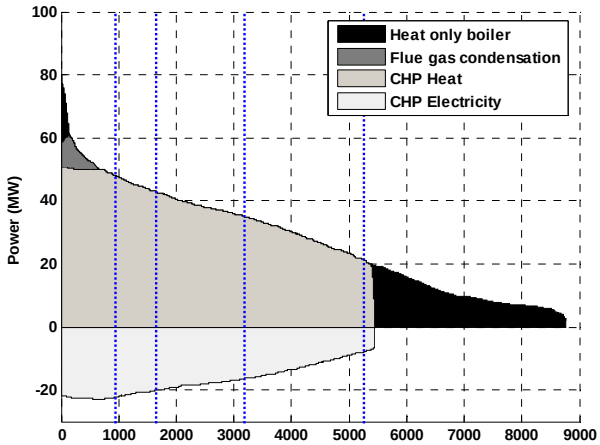

Figure 2. Load duration curve with different production units, Enköping. Dotted vertical lines are the analysed heat load situations: 21.5, 35, 43, 48 MW. CHP Electricity is produced electricity in the CHP station.

As can be seen in the figure the CHP station is the dominating heat source in this DH network. The DH production units are often said to gain from a reduced DH return temperature, however, this influence is often not shown. This study aims to show the influence of DH temperature on a CHP station by simulating different scenarios in an off-design model of the CHP station in Enköping, Sweden.

2.1. Temperature levels in Enköping

The Swedish District Heating Association has design criteria of performance of DH substations [8]. The design criteria are primary return temperature (T_{pr}) at design outdoor temperature (DOT) and T_{pr} during summertime with no space heating load at a certain primary supply temperature (T_{ps}) (100 and 65°C, respectively). The supply temperatures are considered as minimum temperature levels at the customers in order to receive a properly working DH substation. In Sweden, the heating system is hydraulically separated from the DH network. Figure 3 shows the design supply temperature level represented as a thick red line. The red and blue dots are T_{ps} and T_{pr} measured during a year in Enköping. The narrow lines represent T_{pr} simulated for different heat loads with a substation designed according to the Swedish District Heating Association. The lines represent a heating system working with a 60/40°C temperature program (60/40°C program is a commonly used temperature program). Two different sizes of multi-residential buildings are represented containing 90 and 15 apartments, respectively. Two different types of DH substations are also represented for each building, two-stage and parallel connection. The model used for these simulations is done based on theory described by e.g. [3], [4] and [6].The pink and light blue area in Figure 3 represents the temperature parameter study of the DH system.

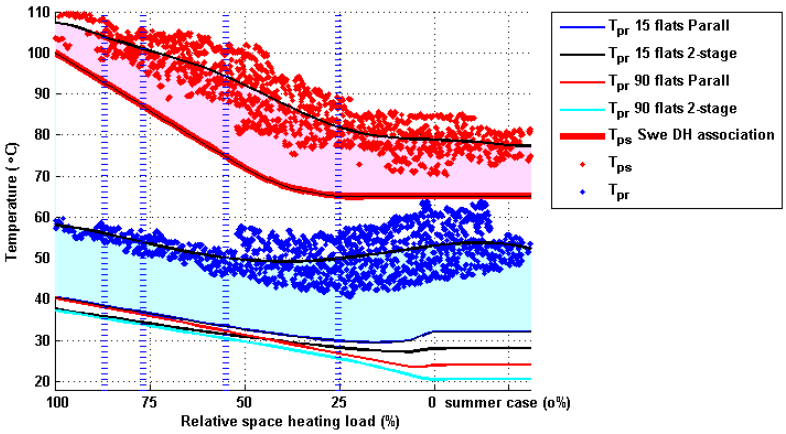

Figure 3. Temperature levels in Enköping DH network 1998/99 [7], calculated possible T_{pr} and design T_{ps}. Temperatures are given in °C. Black lines are total trend lines. Pink and light blue represent the temperature parameter study.

As previous mentioned, the design T_{ps} in the figure is the temperature level needed at the consumers. Of course this temperature level needs to be higher at the production site, however, in a well-dimensioned and well-functioning DH network this temperature can be reduced from the trend lines shown in Figure 3. The T_{pr} in the figure differs strongly from possible return temperatures from well working DH substations. This can be explained, at least partly, by bypass connections in the DH network in order to achieve a proper temperature at far-distance customers. However, this temperature level is much higher than for well-functioning substations.

The simulated examples of T_{pr} that are shown in Figure 3 are far from extreme and are done with a common, normal temperature, heating system. The calculated T_{pr} are comparable with results from other studies, see e.g. [5].

In modern buildings with low heat demand, heating systems working at a low temperature level has a large potential of reducing the T_{pr} even further. In order to gain from the decreased return temperature level at the production units, it is important that the DH network is well functioning.

A reduction of T_{pr} results not only in power production benefits, also the DH flow will decrease, resulting in more heat transfer possibilities. A lower DH flow rate results in possibilities to use narrower pipe dimensions, which will decrease piping costs, or a higher heat transfer capacity in the existing piping allowing more connected buildings.

2.2. Limitations

The aim of this paper is to analyse the response of electric power-to-heat ratio in the CHP station from different DH temperature levels. The calculation for the CHP station is made with a fully validated off-design model of a CHP-station located in Enköping. The heat source in the CHP station (bio fuel-fired boiler) is considered as a black box with constant efficiency. The DH network is not in focus in this part of the project, and is not studied in detail. The heat load is considered to be constant and only the temperature levels are varied, ignoring changes in heat losses from the DH network due to a changed flow rate. However, changes in pressure losses in the DH network (and also changed pump power needed for DH distribution) are taken into account by assuming the DH network as a heat sink with a constant flow resistance coefficient.

3. DESCRIPTION OF THE CHP STATION MODEL

The full CHP station is modeled in one single off-design model. The platform for this is the commercially available heat- and mass balance software IPSEpro. The off-design models, within IPSEpro, were developed by Genrup and Truedsson at Lund University and are described in detail in [11].

The steam turbine model is based on a standard part turbine methodology. In this approach, a number of turbine stages are lumped to a group or part of the turbine. Each of these turbines is principally described by two parameters: efficiency and swallowing capacity. The efficiency is correlated to the Parson number i.e. a gauge of the stage loading.

The swallowing capacity or flow–pressure relation is used to calculate the node pressures. The pressure at each node sets the heat drop distribution through the turbine train as well as the extraction pressures for e.g. pre-heaters. The probably most widely used model is the one evolved from Stodola's ''cone law''. The model derived from Stodola is, however, known to be quite erroneous on choked highly loaded sections (e.g. control stages) [2]. Instead, in this work, the model by Beckmann is used. The used equations are left out here, and the reader is referred to the original source [1].

Preheaters and condensers are modeled as components rather than individual tubes. This approach simplifies the calculation to convective heat transfer correlations for section and section pressure drop characteristics. The DH condensers heat transfer coefficients are based on an empirical model and are functions of the average DH temperature in each condenser. The model has been validated against a series of full DIN-1943 tests on the specific plant, under various load conditions, and has proven to be accurate and robust.

A schematic view of the CHP station connected to a DH network can be seen in Figure 4.

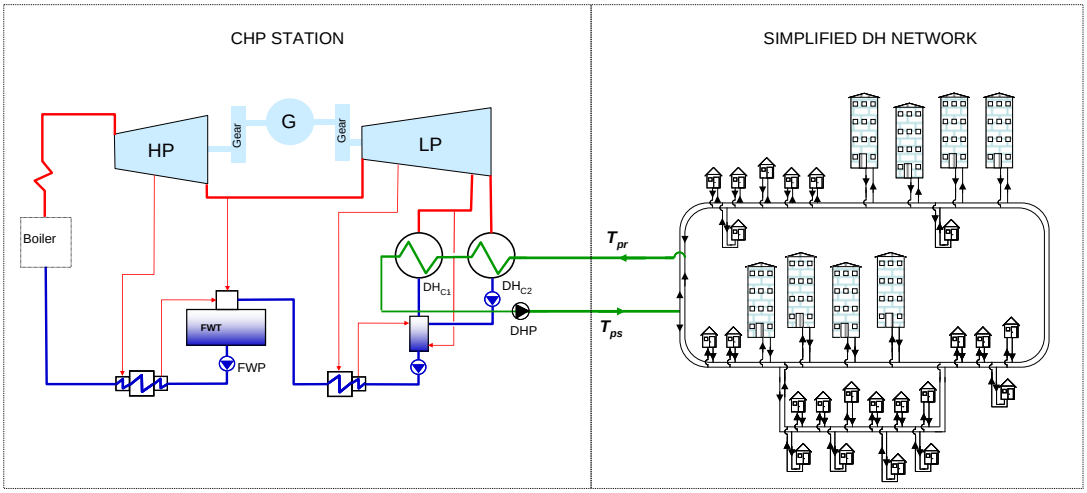

Figure 4. View of the CHP station connected to a DH network.

3.1. The hot water condenser configuration

For good utilization of heat and wide temperature flexibility the DH production is done at two pressure levels, e.g. two condensers are used. Adding an extra pressure level will allow part of the heat to be extracted at a lower pressure level. The distribution between the two condensers is determined by the load, mass flow and temperature levels in the DH network. The configuration with two condensers is the dominating layout for CHP technology used in the Nordic countries.

Figure 5 illustrates the heat distribution between the condensers at a DH load of 40 MW with two different supply and return temperatures. In order to simplify the discussion here, the terminal temperature difference (TTD) and $C_{p,DH}$ are assumed to be constant. In Figure 5a different T_{pr} are plotted, 45°C and 50°C, T_{ps} is kept constant at 85°C. In Figure 5b T_{pr} is held constant at 50°C and T_{ps} is plotted at 85°C and 90°C. Now, consider the formula for heat exchangers (equation 1):

$$Q_{DH} = \dot{m}_{DH} \cdot C_p (T_{ps} - T_{pr}) \tag{1}$$

If T_{pr} or T_{ps} is changed at constant DH load (Q) and, (1) indicates that the district heating mass flow (m_{DH}) will be affected. The slope of the curve in Figure 5 represents the mass flow times C_p (see (1)). In Figure 5 it can be seen that a steeper slope will result in a higher condensing temperature in the second DH condenser (DH_{C2}). Hence changing any of the DH temperatures will affect the DH_{C2}. Lowering the T_{pr} will not affect the temperature in the first DH condenser (DH_{C1}), see Figure 5a. Figure 5b illustrates how an increased T_{ps} will raise the condensing temperature in the DH_{C1} in the same order.

A lower condensing temperature is equal to a lower pressure, and therefore, if the condensing temperature can be reduced, the expansion ratio of the turbine will increase.

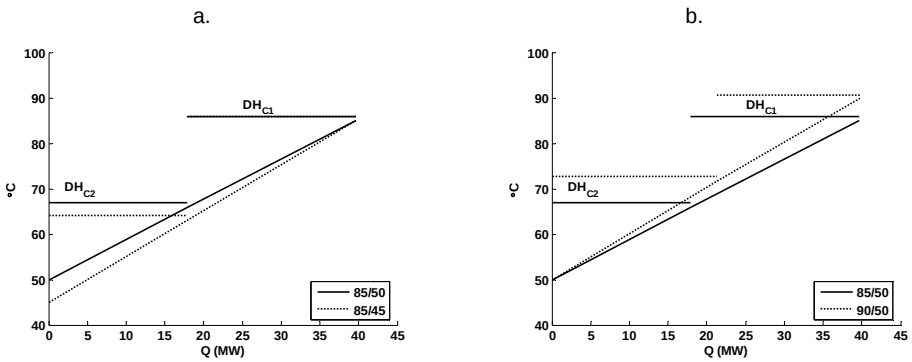

Figure 5. TQ diagram for different supply and return DH temperatures.

4. RESULTS

Calculations of the electric power to heat ratio, α, is shown in Figure 7 with reduction of T_{pr} and T_{ps} corresponding to the pink and light blue field in Figure 3 for four different heat load situations, 21.5, 35, 43 and 48 MW. As seen, a reduction of T_{ps} has a larger impact on the α-value than a reduction of T_{pr}. This corresponds with the discussion in section 3.1. The electric power-to-heat ratio is calculated using equation (2) and includes changes of power needed for pumps.

$$\alpha = \frac{Q_{el} - Q_{DH\ pump} - Q_{FW\ pump}}{Q_{DH}} \tag{2}$$

As seen in figure 6 the electric power-to-heat ratio is strongly affected when T_{ps} is reduced with 5°C starting at a high supply temperature level. By reducing T_{ps} further, the electric power-to-heat ratio will continue to increase, but more moderately.

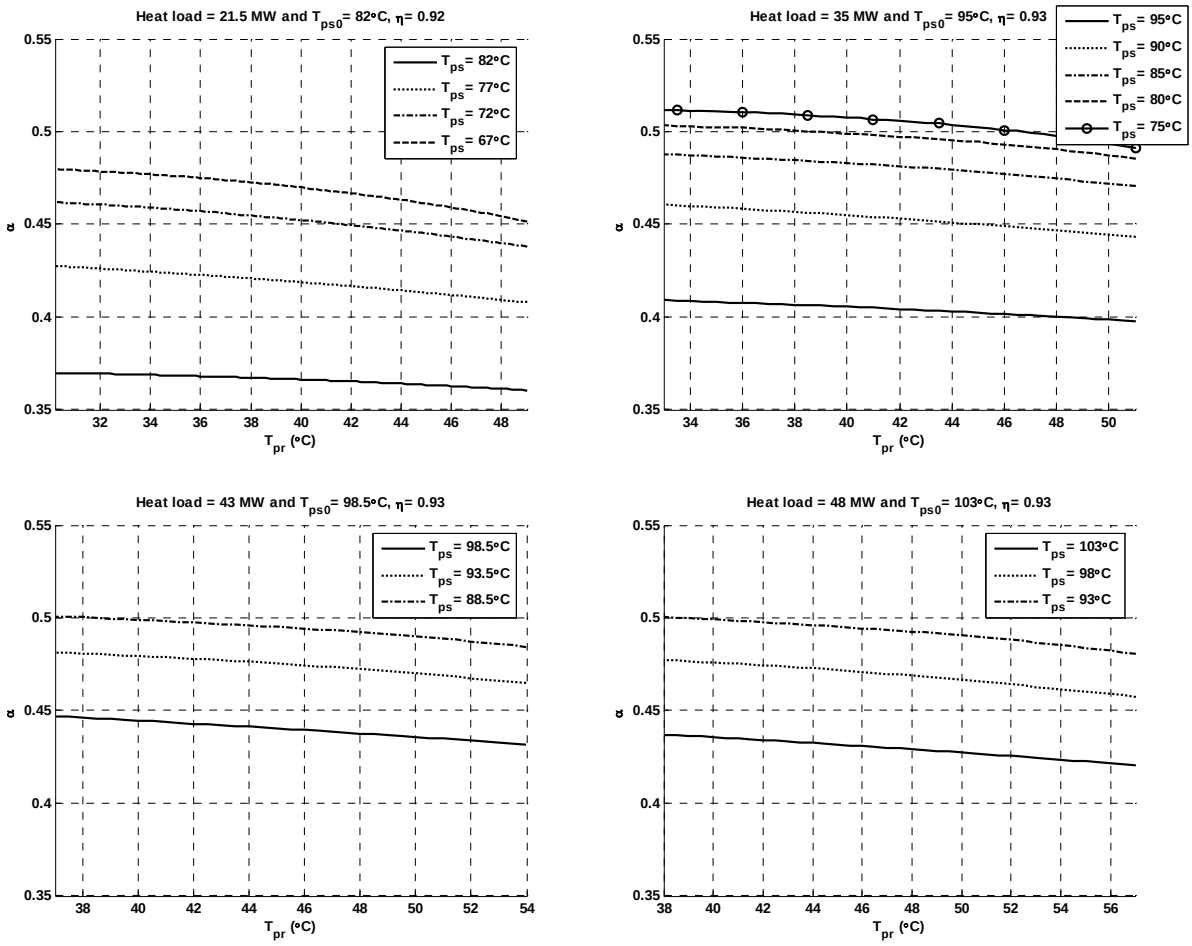

Figure 7. α-values as function of T_{ps} and T_{pr} at different load cases.

By just reducing the supply temperature, the flow rate in the DH network will increase and may cause problems in the piping system. Reducing the return temperature leads to a decreased flow rate and bottlenecks in the system can be avoided. The total efficiency (noted as η in Figure 7) is constant and does not take influence from the changes of DH supply and return temperatures.

4.1. Special case full load

During a full load situation, the heat production from the CHP station is not sufficient to cover the total heat demand in the DH network. As seen in Figure 2, additional heat sources are introduced. The additional heat sources in Enköping come from flue gas condensation and a heat only boiler. Depending on how the additional heat sources are connected, the supply and return temperature from condensers can be allowed to vary. In Figure 8 a parameter study of these temperatures are done. At full load the control stage of the steam turbine is fully open resulting in that the heat output in this case is a function of the electric power to heat ratio. Depending on the combination of electricity and heat prices and prices of alternative fuels for the boiler, the amount of produced electricity can vary.

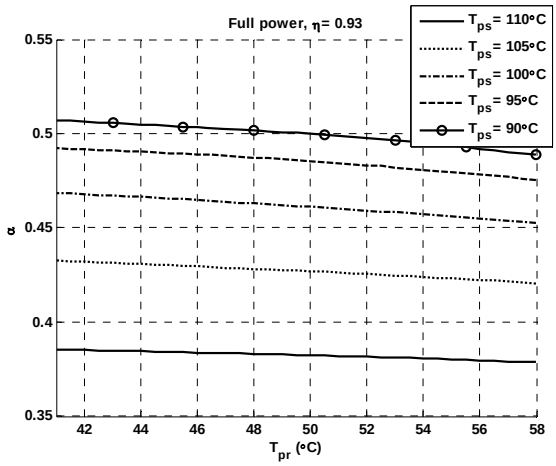

Figure 8. Special case: full power.

The heat input (to the boiler) during full load case is constant 77.2 MW and total η = 0.93. The heat load varies from 47.7 to 53 MW with corresponding power production 24.3 and 19.8 MW electricity.

4.2. Increased electric production

The CHP station will, with the same heat load, at reduced DH temperature levels increase the power production. In Figure 9 the increased power production is shown for reduced T_{pr} with 5 and 10°C, reduced T_{ps} with 5°C and a combined reduction of both T_{pr} and T_{ps} with 5°C each.

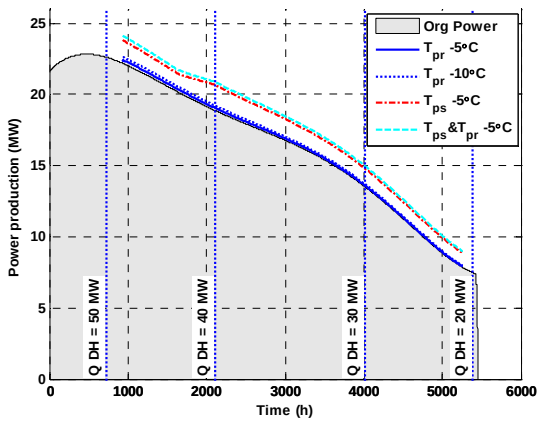

Figure 9. Power production in the CHP station.

As seen the gain in power production has larger potential when reducing T_{ps} than with a reduction of T_{pr}. However, the gain of reduction of T_{pr} is not negligible on annual basis.

In Table 1, the annual gain of power production has been calculated. Since the most common heat source in detatched houses in Sweden is electric heating, the gain of increased power production can be transferred into supplying up to 271 detatched houses with electricity based on an annual consumption of 26 000 kWh. If the full load case also is considered, the electric power will increase even further.

Table 1. Increased annual electric power production with changed DH temperatures.

Tps reduction (°C)	-	-	5	5
Tpr reduction (°C)	5	10	-	5
Increased Power production (MWh)	654(1)	1226(1)	6130(1)	7035(1)
Corresponding number of electric heated detached houses	25	47	236	271

(1): Calculated for DH heat load between 21.5 and 48 MW.

An increase of primary cooling can be used for increasing the capacity in the DH network. A greater difference between the supply and return temperature allows more heat transport per unit mass. This means that more buildings or other heat loads can be connected to the DH network without the total flow rate being affected. By using heat balance calculations equation the new heat load with an increased temperature difference can be calculated, see equation (3).

$$\left.\begin{aligned} Q_{DH,0} &= \dot{m}_{DH,0} \cdot C_p(\Delta T_{P,0}) \\ Q_{DH,\Delta T} &= \dot{m}_{DH,0} \cdot C_p(\Delta T_{P,0} + \Delta T_{p,inc}) \end{aligned}\right\} \frac{Q_{\Delta T}}{Q_0} = 1 + \frac{\Delta T_{p,inc}}{\Delta T_{P,0}} \tag{3}$$

In Figure 10, the changed heat loads with a reduction of T_{pr} (increased ΔT) with 5°C is shown. The effect on operation time for the CHP station is also included.

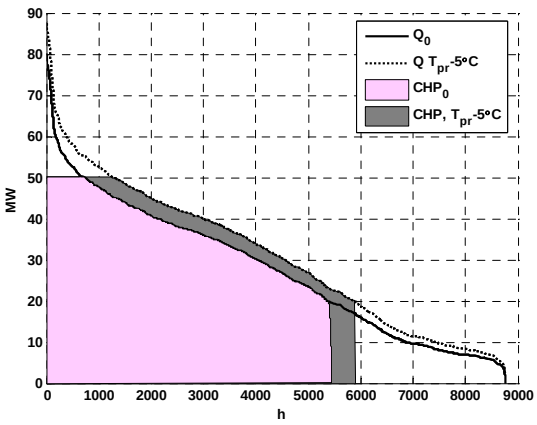

Figure 10. Increased operational time for the CHP station when the heat load is increased.

The annual working hours of the CHP station will increase with around 500 hours. The time with full heat load will increase from 660 hours to 1235 hours. With an assumed electricity production of 21 MW at full load the increased power production will be more than 10 GWh on annual basis. This corresponds to electricity consumption of 400 detached houses.

5. DISCUSSION

The analysis shows that a reduction of T_{ps} has a greater impact on the electric power-to-heat ratio than a reduction of T_{pr}. However, there is a larger potential of reducing the T_{pr} in a DH network due to the demand of relative high supply temperature and the large temperature gap between actual values and simulated T_{pr}. The return temperature is not only dependent of the performance of the DH substations, but also of the functionality and construction of the DH network. When the return temperature level differs considerably from the potential return temperature level from DH substations, the performance of the DH network should be investigated. A reduced T_{pr} results in a decreased flow rate in contradistinction to a reduction of T_{ps} that will increase the flow rate. Reducing the flow rate in the DH network can be of interest in order to avoid bottlenecks in the DH network or to allow further expansion of the DH network. If the reduced temperature level is used to increase the heat load in the DH network the annual operational hours of the CHP station will increase and more electricity can be produced. An increased power production is often beneficial to the DH company due to pricing of electricity, especially with a bio-fueled CHP station.

With modern low temperature heating systems the lowered T_{pr} from DH substations will decrease the total DH return temperature. The lowered demand of temperature level with low temperature heating systems can, at least in new DH systems, contribute to a decreased primary supply temperature level.

ACKNOWLEDGEMENTS

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VII

IMPROVED TEMPERATURE PERFORMANCE OF RADIATOR HEATING SYSTEM CONNECTED TO DISTRICT HEATING BY USING ADD-ON-FAN BLOWERS

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ABSTRACT

District heating (DH), which is the most common heat source in multifamily houses and commercial buildings in Sweden, can be produced in several different type of production units.

In order to gain thermal efficiency in a DH system it is important that DH supply and return temperatures are kept low. The temperature demand in the DH system is, during the heating season, dependent on the temperature level in the heating system of the DH connected buildings. Many production units benefit from a lowered DH return temperature, while others are more affected by a reduced supply temperature. In a CHP-station the heat to power ratio will increase when the DH supply temperature is decreasing. In order to reduce the temperature demand, low temperature heating systems are of interest, as well as systems resulting in a low DH return temperature.

To increase the heat output in an existing radiator heating system, the radiators can be complemented with small electric fans resulting in an increased share of forced convection in the heating system. Field studies have shown that the heat output, with constant supply temperature and mass flow through the radiator, can increase with more than 50%.

INTRODUCTION

For many years, return temperatures in DH networks have been an important issue for DH research. A low DH return temperature is in many cases favorable for the DH production units. However, if also the supply temperature could be kept at a low level the share of electricity produced in a CHP station could increase. This would lead the way towards an increased share of electricity produced by non fossil fuels. In Sweden more than 30 % of the DH is produced in CHP stations [4].

In many reports the gain from a reduced temperature level in the DH network has been discussed and quantified in economic terms, see e.g. [12], [13].

The DH supply temperature level in the DH network is, during heating season, dependent of the temperature demand in the DH-connected buildings heating system.

In modern buildings low temperature heating systems are common, which may allow reduced DH temperature level. In order to reduce the temperature demand in existing buildings the idea of using small add-on-fan blowers placed under the radiator to increase the heat output due to an increased share of forced convection came up.

Objective

The field study presented in this paper investigates the possibility to reduce the space heating temperature program and estimates the impact on the DH supply and return temperature. Possible reduction of the DH-flow rate is also calculated.

This paper is focusing on buildings indirectly connected to the DH network through a substation with heat exchangers (HEX).

DESCRIPTION OF ADD-ON-FAN BLOWER

The add-on-fan blower that is tested in this study consists of several regular DC motor driven fans, originally used for cooling, mounted under a radiator, see Fig. 1. In the study, two different kinds of radiators were tested, a panel and a column radiator.

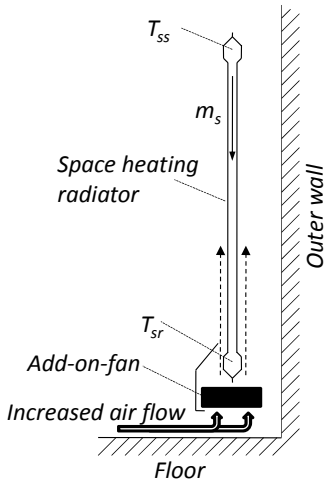

Fig. 1 The add-on-fan blower mounted on a panel radiator.

The add-on-fan blowers in this study are provided by a Swedish company: A-energi AB (the product is called “fläktelement” in Swedish). The company describes the features of the add-on-fan blower as a possibility to reduce the temperature program without replacing the radiators, with the aim to reduce the electricity demand for buildings supplied with heat from heat pumps [5].

THEORY

In this section a theoretic analysis of the impact of forced airflow on heat output from radiator with a length of (*L*) 1 m and height of (*h*) 0.59 m is described. The radiator is in this study approximated by a flat vertical plate. The indoor temperature is assumed to be constant at 21°C and equal to T_{inf}.

Heat output

The heat output from the radiator to the room arises from convection and radiation. The heat transfer process from heating water to the room through a radiator is summarized in equation 1 [7], [8].

$$Q = \dot{m}_s \cdot c_p (T_{ss} - T_{sr}) = (k \cdot A)\Delta\theta \tag{1}$$

k is the heat transfer (convection) from the water to the surrounding metal, conduction through the metal and convection from the outer surface of the radiator to the room according to equation 2.

$$\frac{1}{k} = \frac{1}{\alpha_{water-metal}} + \frac{\delta_{metal}}{\lambda_{metal}} + \frac{1}{\alpha_{conv} + \alpha_{rad}} \tag{2}$$

The dominating parameters in this equation are the convection and radiation between the radiator and the room (α_{conv} and α_{rad}), while the other terms, in this case, can be neglected. This results in a new equation for energy output, see equation 3.

$$Q = Q_{rad} + Q_{conv} = \alpha_{conv} \cdot A_{conv} \cdot \Delta\theta + \alpha_{rad} \cdot A_{rad} \cdot \Delta\theta \tag{3}$$

The temperature $\Delta\theta$ is the logarithmic mean temperature difference according to equation 4.

$$\Delta\theta = \frac{T_{ss} - T_{sr}}{\ln \frac{T_{ss} - T_i}{T_{sr} - T_i}} \tag{4}$$

Radiation

According to Trüschel [8] the heat output from radiation can be estimated according to equation 5.

$$Q_{rad} = \alpha_{rad} \cdot A_{rad} \cdot \Delta\theta \approx \approx 4 \cdot \frac{\varepsilon_{rad} \cdot \sigma}{\varepsilon_{rad} + \frac{A_{rad}}{A_{radiator}} \cdot (1 - \varepsilon_{rad})} \cdot T_m^3 \cdot A_{rad} \cdot \Delta\theta \tag{5}$$

Where the temperature, T_m, is the mean temperature of the radiator surface and the surfaces in the rooms, see equation 6. For a panel radiator the $A_{rad}/A_{radiation}$=1 [8].

$$T_m = \overline{T_{radiator} + T_{room,surface}} \approx \frac{(T_{ss} + T_{sr})/2 + T_i}{2} \tag{6}$$

Since the A_{rad} and emissivity, ε_{rad}, are constant for a specific radiator, the relation can be simplified to equation 7.

$$Q_{rad} \approx C \cdot T_m^3 \cdot \Delta\theta \tag{7}$$

Convection

The convection that arises due to the temperature difference between the radiator surface and the surrounding air is a function of the Nusselt number (*Nu*), see equation 8.

$$\alpha_{conv} = Nu \cdot \lambda / h \tag{8}$$

The heat output due to convection is divided into three sections, natural, mixed and forced convection.

For natural convection, the *Nu* number is dependent on the Rayleigh number (*Ra*), which is a product of the Prandtl number (*Pr*) and the Grashof number (*Gr*). For air, *Pr* can be considered constant, *Pr*=0.71, while

$$Gr = g \cdot \beta \cdot \Delta\theta \cdot \frac{h^3}{\nu^2} \tag{9}$$
$$\beta = 1/T_{\inf} \approx 1/T_i$$

where *g* is the gravity force, *v* is kinematic viscosity and *β* is the coefficient of expansion.

Several empirical relations describing *Nu* are available. In this study a relation described by Churchill has been used [9], see equation 10 and 11.

$$\overline{Nu} = 0.68 + \frac{0.67 \cdot Ra^{0.25}}{[1 + (0.492/\Pr)^{9/16}]^{4/9}} \qquad Ra < 10^9 \tag{10}$$

$$\overline{Nu}^{0.5} = 0.825 + \frac{0.387 \cdot Ra^{1/6}}{[1 + (0.492/\Pr)^{9/16}]^{8/27}} \qquad Ra > 10^9 \tag{11}$$

For forced convection the *Nu* number is calculated by equations described by Holman [10], see equations 12 and 13.

$$\overline{Nu} = 0.664 \cdot \mathrm{Re}^{0.5} \cdot \Pr^{1/3} \qquad \mathrm{Re} < 5 \cdot 10^5 \tag{12}$$

$$\overline{Nu} = Pr^{1/3} \cdot (0.037 \cdot \mathrm{Re}^{0.8} - 871) \qquad 5 \cdot 10^5 < \mathrm{Re} < 10^7 \tag{13}$$

and the Reynolds number, *Re*, is described as:

$$\mathrm{Re} = \frac{u \cdot L}{\nu} \tag{14}$$

The product of Gr/Re^2 describes the dominating type of convection. If Gr/Re^2>10, natural convection is dominating, if Gr/Re^2≈1, both natural and forced convection is of importance and if Gr/Re^2<<1, forced convection is dominating. When a mix of forced and natural convection occurs, the Nusselt number is calculated according to equation 15 [11].

$$\overline{Nu} = (\overline{Nu}^3_{forced} + \overline{Nu}^3_{natural})^{1/3} \tag{15}$$

Impact of air speed on space heating temperature

Results from the theoretical analysis, using the equations above, are shown in Fig. 2 to Fig. 4. The heat output for a radiator designed for the temperature program 60/45°C is illustrated as a function of the air speed in Fig. 2. The supply temperature and the mass flow through the radiator are kept constant. Two cases have been derived, one with heat output only from convection, and one with heat output from both radiation and convection. With *ε*=0.9, the share of heat output from radiation will be 65 % at DOT.

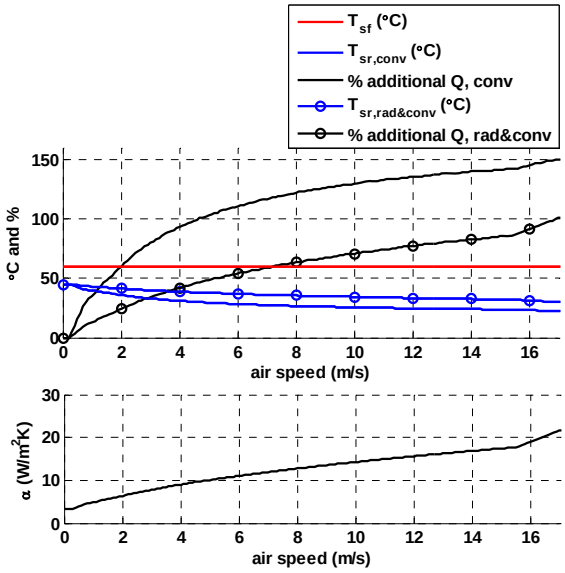

Fig. 2 Calculated heat output improvements at T_{ss}=60°C, T_{sr0}=45°C with increasing airspeed. m_s is kept constant.

As seen, the additional heat output from the radiator is increasing rapidly when the air flow is increased. With radiation taken into account, the increase is somewhat lower since the mean temperature, T_m, is decreased, see equation 7.

In Fig. 2 the heat output is increasing. In Fig. 3 and Fig. 4 the supply temperature to the radiator is reduced instead to keep the heat output constant. New T_{ss} and T_{sr} can now be calculated under the assumption that the total heat output and the mass flow (m_s) through the radiator are constant. The impact of the air flow is described for three different heat loads (Q_{rel}=100%, 50%, 25%) with standard 60/45°C temperature program as a reference. See Fig. 3.

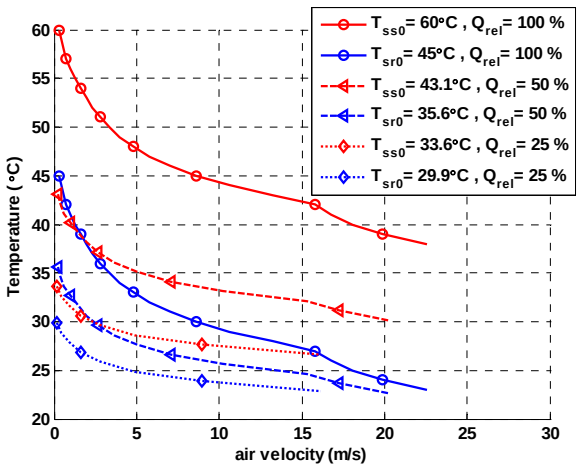

Fig. 3 Possible T_{ss} and T_{sr} to for three heat load situations at different air speeds. Q_{rad}=65 % at DOT.

New temperature programs have been derived for some moderate air speeds, see Fig. 4. As seen the impact of an increased air flow, expressed in °C, is larger at high relative heat load.

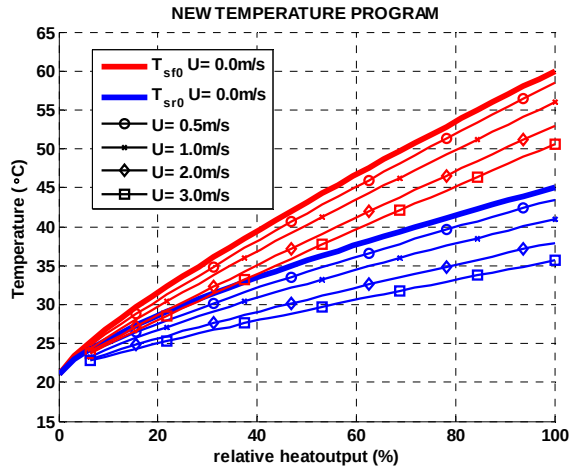

Fig. 4 New space heating temperature programs at different air speeds. Red lines: T_{ss}, Blue lines: T_{sr}. Q_{rad}=65 % at DOT.

In the calculations performed, the radiator is assumed to have the same heat output from both sides of the radiator. The air flow is assumed to be uniformly distributed through the length and height of the panel radiator. This is not the case in the real add-on-fan blower applications, however, one can expect results following the same pattern.

EXPERIMENTAL STUDY

To investigate the performance of the add-on-fan blower, two radiators of different type were supplied with such device during the heating season 2009/2010. The power supply to the fans was scheduled to switch on and off while the mass-flow (m_s) through the radiator was kept at a constant level.

Field study object

The radiators are situated in two offices at Lund University. The original temperature program for the radiators in the building is running at 60/45°C at DOT (represented by narrow black lines in Fig. 8 trough Fig. 13).

The radiator types tested were:

- Panel radiator, see Fig. 5
- Column radiator, see Fig. 6.

Fig. 5 Add-on-fan blower mounted on a panel radiator.

Fig. 6 Add-on-fan blower mounted on a column radiator.

For each radiator the fans have been run at two different rotation speeds. The net electric power consumption (P_{fan}) has been measured. See Table 1 for P_{fan} and the design heat energy output at DOT. Note that the electric power to the add-on-fan blower is constant and not dependent on the relative heat output.

Table 1 Radiator and add-on-fan blower design.

Radiator type	P_{fan} (el)	**Q @ DOT, 60/45°C** (Heat)
Panel	2.7 W 1.9 W	360 W*
Column	3.0 W 2.2 W	430 W*

* Calculated for new radiators of the same dimensions manufactured by Lenhovda radiator factory [3]

The radiators are located in traditional office environment in a building built in 1960.

Data acquisition

Measured parameters in the test were: secondary supply and return temperature (T_{ss} and T_{sr}), indoor temperature (T_i) and outdoor temperature (T_{out}).

The impact on return temperature for a given supply temperature was then calculated.

In Fig. 7, a screenshot from the logger software is shown. The return temperature is decreased by 5°C when the fan is switched on. This results in an increased heat output by more than 60 %.

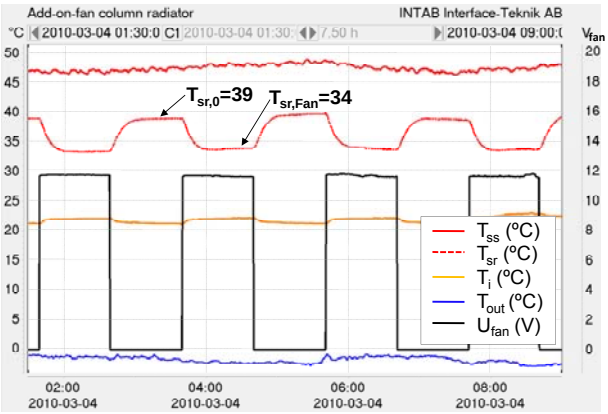

Fig. 7 Log file from field study.

New reduced temperature program will be derived in next section.

MODIFYING SPACE HEATING TEMPERATURE PROGRAM

Method

When the add-on-fan blower is switched on, the T_{pr} is decreasing, causing an additional heat output since m_s is kept constant. See Fig. 8.

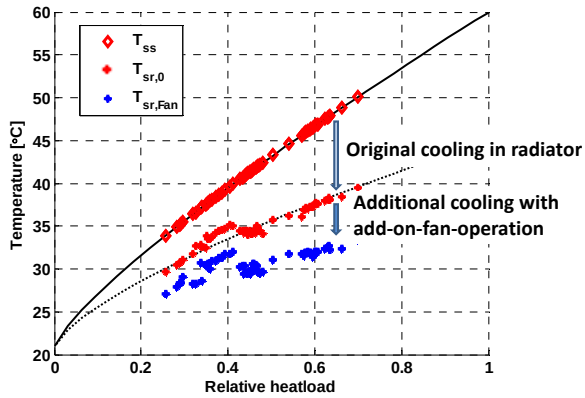

Fig. 8 Increased cooling of secondary system

The relative heat output from the radiator with and without fan operation is calculated from equation 1.

$$\left.\begin{aligned} Q_{rel,0} &= \dot{m}_s \cdot C_p (T_{ss} - T_{sr0}) / Q_{DOT} \\ Q_{rel,Fan} &= \dot{m}_s \cdot C_p (T_{ss} - T_{sr,Fan}) / Q_{DOT} \end{aligned}\right\} \quad (1)$$

Since the temperature drop in the radiator is increasing with the fan in operation, the radiator now could be considered oversized. Then, with the same type of reasoning as in e.g. [2], the T_{ss} program or m_s needs to be adjusted in order to avoid overheating of the building. In this paper, the m_s has been considered constant, allowing us to compute the new relative space heating load for a given T_{ss} according to

equation 2. $Q_{rel,0}$ is computed using the original space heating temperature program.

$$Q_{rel,Fan} = \frac{(T_{ss} - T_{sr,Fan})}{(T_{ss} - T_{sr0})} \Big/ Q_{rel,0} \qquad (2)$$

Knowing $Q_{rel,Fan}$, a new temperature curve, which will result in correct heat output from the radiator with the fan in operation, can be calculated. The curve appears to the right in the diagram, see Fig. 9.

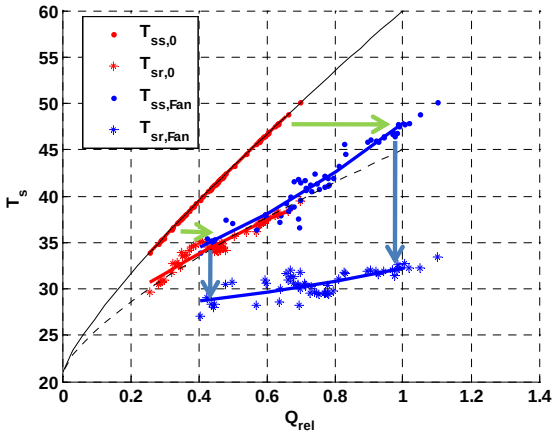

Fig. 9 Modified secondary temperature program.

New space heating temperature program - results

The procedure described above has been applied to all collected data. Results are shown in Fig. 10 and Fig. 11 for the panel radiator, and Fig. 12 and Fig. 13 for the column radiator.

As seen in the figures, the temperature program is now significantly reduced for both the panel and the column radiator. The new temperature program shows a similar pattern for both types of radiators. For the panel radiator, the measured return temperature is more concentrated, especially at the higher fan speed. This could be explained by a more favorable air flow pattern due to the physics of the radiator.

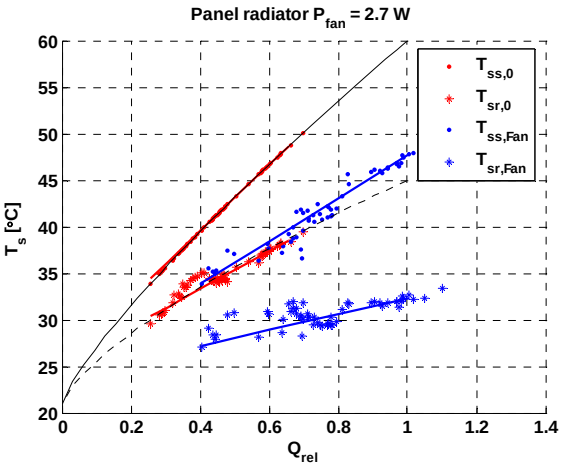

Fig. 10 Modified temperature program for panel radiator, P_{fan}=2.7 W.

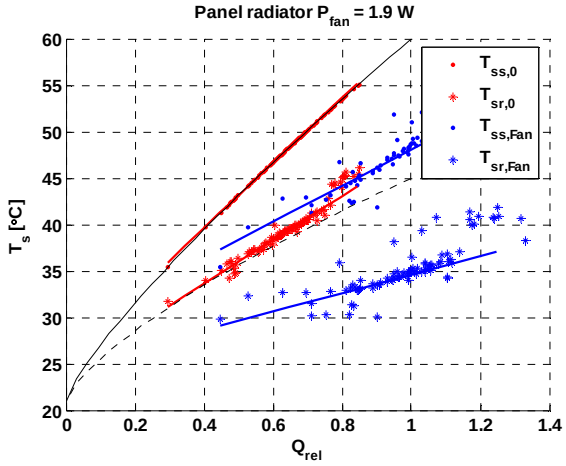

Fig. 11 Modified temperature program for panel radiator, P_{fan}=1.9 W.

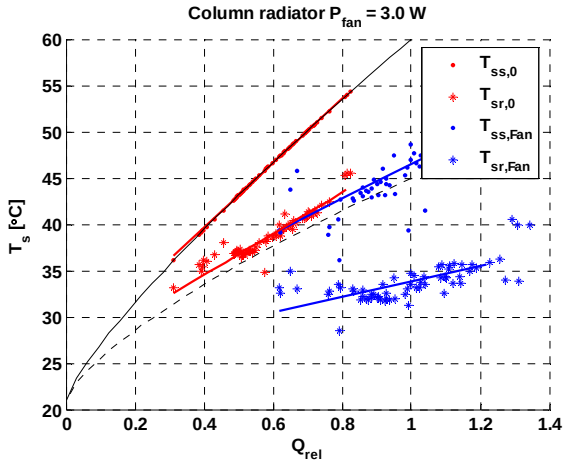

Fig. 12 Modified temperature program for column radiator, P_{fan}=3.0 W.

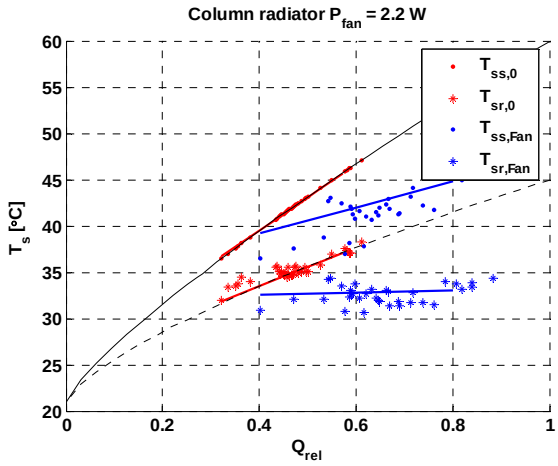

Fig. 13 Modified temperature program for column radiator, P_{fan}=2.2 W.

INFLUENCE ON DH NETWORK

Knowing the reduced temperature level on the secondary side of the HEX, the impact on the DH network can be estimated. The impact is calculated based on two different strategies:

1. Primary supply temperature (T_{ps}) is kept at the same level as before
2. The primary flow (m_p) through the HEX is kept constant

By applying the first strategy, both T_{pr} and the mass flow in the DH network is reduced. The second strategy results in a reduced T_{ps} and T_{pr} without changing the flow rate in the DH network.

Results so far will now be applied to a DH substation dimensioned as recommended by the Swedish district heating association [1]. The calculations are made with a parallel connected DH substation serving a building with 20 apartments. The substation is providing the building with heat and domestic hot water (DHW) and DHW circulation. The assumed DHW usage is 125 l/apartment&day, space heating load at DOT is 3 kW/apartment. The heat loss from DHW circulation is assumed to be 0.1 kW/apartment. For each space heating load a flow-weighted mean value for T_{ps} and T_{pr} is calculated for a time period of 24 h, including heat load from both DHW and DHW circulation. The reference DH supply temperature, dependent on the space heating load, is assumed as illustrated in Fig. 14.

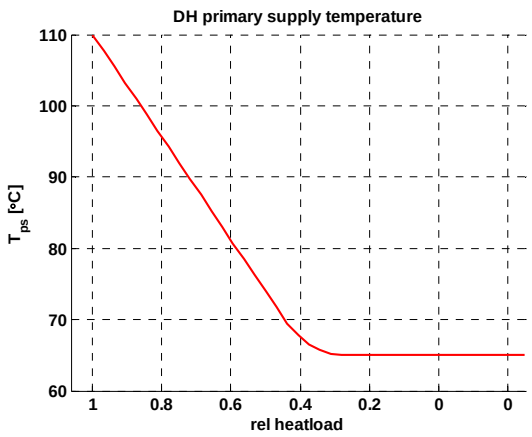

Fig. 14 DH primary supply temperature.

Results

The first control strategy is in Fig. 15 - Fig. 20 noted as *'T_{ps} unchanged'*, and the second strategy is noted as *'m_p unchanged'*.

In Fig. 15 and Fig. 16 the possible reduction of DH supply temperature is shown.

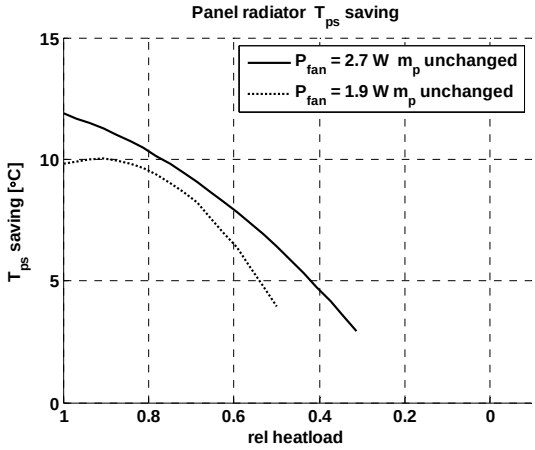

Fig. 15 Resulting T_{ps} reduction with panel radiator.

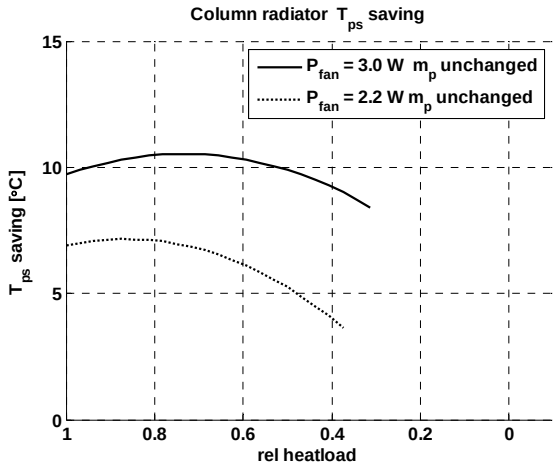

Fig. 16 Resulting T_{ps} reduction with column radiator.

In Fig. 17 and Fig. 18 the reduction of T_{pr} is shown. As seen the reduction of T_{pr} is of the same magnitude independently of which control strategy is used. However, by keeping the DH supply temperature constant (strategy 1) the flow rate in the DH network is affected, see Fig. 19 and Fig. 20.

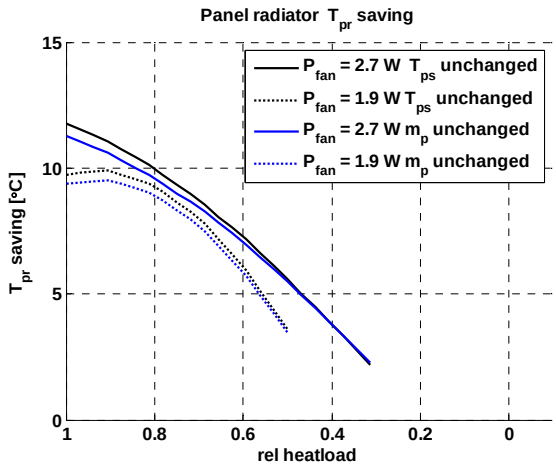

Fig. 17 Resulting T_{pr} reduction with panel radiator.

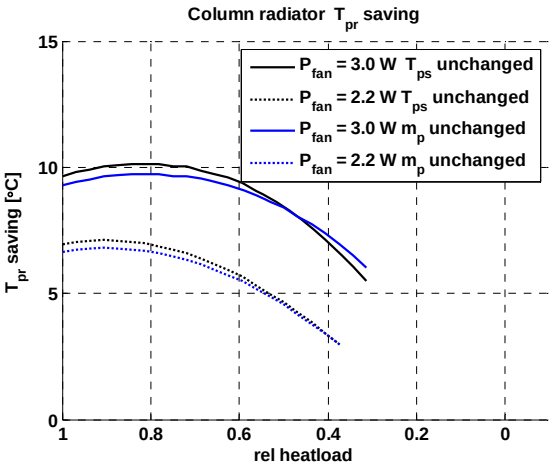

Fig. 18 Resulting T_{pr} reduction with column radiator.

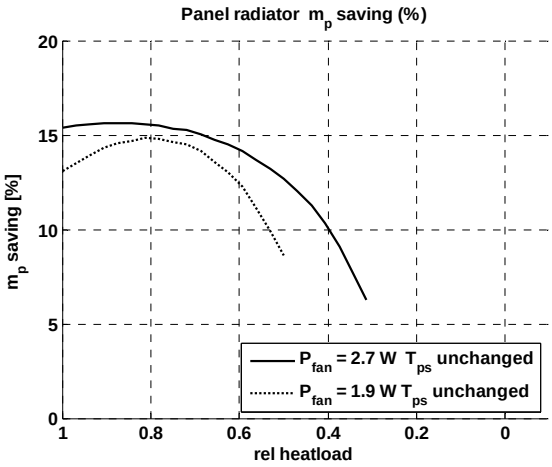

Fig. 19 Resulting m_p reduction with panel radiator.

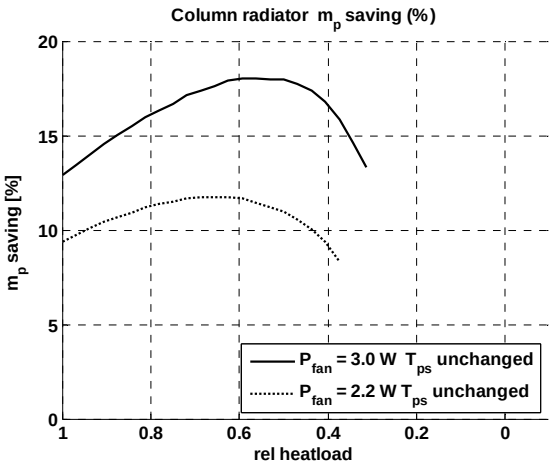

Fig. 20 Resulting m_p reduction with column radiator.

Annual gain in T_{ps} and T_{pr} during heating season

In order to evaluate the annual impact on the primary temperature level, the outdoor temperature has to be considered. In this case measured values for the outdoor temperature in Malmö have been used, see Fig. 21.

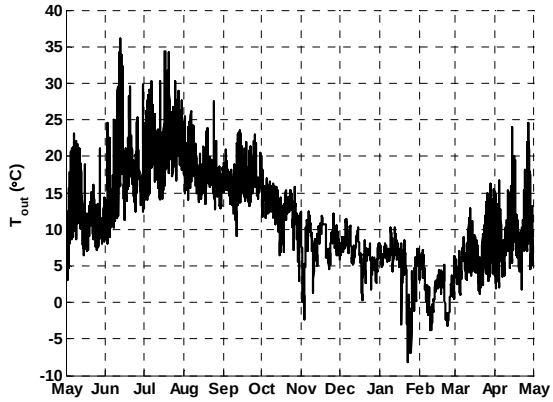

Fig. 21 Outdoor temperature in Mamö 1st May 2006 – 30th April 2007.

When calculating the annual gain for a DH-network a comparison of flow-compensated mean temperature during the heating season has been made. For the calculations we assume a maximum heat output (Q_{DOT}) at -15°C and the balance temperature, when no space heating is needed, +17°C.

Table 2 Reduction in annual primary temperature level during heating season

		T_{ps} - unchanged ΔT_{pr}	m_p - unchanged $\frac{\Delta T_{ps} + \Delta T_{pr}}{2}$
Column radiator	P_{fan}= 2.2 W	-2.2°C	-2.4°C
	P_{fan}= 3.0 W	-5.7°C	-6.6°C
Panel radiator	P_{fan}= 1.9 W	-0.8°C	-0.9°C
	P_{fan}= 2.7 W	-2.5°C	-2.7°C

Note that the annual flow-compensated mean temperature in Table 2 is based on results from the field study where measured values for low relative heat load are missing, which makes the values in the table somewhat underestimated.

CONCLUSION AND DISCUSSION

By installing the add-on-fan blower application on existing radiators the temperature level in the heating system can be substantially reduced. This will also have impacts on the DH network and DH production units.

The impact on the DH network can be applied based on two principles:

1) DH supply temperature kept at the same level as without the add-on-fan blowers. This will result in reduced primary flow rate.

2) Reduced DH supply temperature while primary flow rate is kept constant.

The first strategy could be applied immediately, since the primary supply temperature is kept as the same

level as before. This means that not all heating systems connected to the DH network need to be modified in order to apply this method. The lowered secondary temperature level results not only in reduced DH-return temperature, but also in a reduction of the DH flow rate. The reduced flow rate could be used to increase the number of buildings connected to the DH network, or to avoid bottlenecks in the DH network. The magnitude of the reduction of the DH supply temperature is between 9 and 12°C at DOT and at the same time the flow rate is decreased with more than 10 %. On annual basis the possible reduction of temperature level in the DH network is in the magnitude of several degrees Celsius.

In order to apply the second strategy the demand for a high temperature level in the DH network needs to be reduced for all the connected buildings. Otherwise the DH flow rate will increase. Calculations based on the results from the field study in this paper shows that the DH supply temperature can be reduced with about 10°C at DOT without affecting the DH flow rate. At the same time the DH return temperature will be reduced with as much as 10°C at DOT.

The performance of the tested add-on-fan blowers corresponds to the pattern of theoretical calculations. However, the results are not comparable since the air flow in the pilot project has not been measured.

The results presented here are an important part in the evaluation of effects of improvements in consumer heating systems on primary energy efficiency in DH systems including production plants, especially CHP.

ACKNOWLEDGEMENT

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NOMECLATURE

Abbreviations

CHP	Combined heat and power station
DH	District heating
DHW	Domestic hot water
DOT	Design outdoor temperature
HEX	Heat exchanger (DH substation)

Variables

α	Heat transfer coefficient (W/m^2·K)	*Gr*	Grashof number (-)
β	coefficient of expansion (K-1)	*h*	Height (m)
δ	Thickness (m)	*k*	Heat transfer coefficient (W/m^2·K)
λ	Conductivity (W/m·K)	*L*	Length (m)
ε	Emissisivity (-)	$\dot{m}$	mass flow (kg/s)
v	Kinematic viscosity (m^2/s)	*Nu*	Nusselt number (-)
σ	Stephan-Boltzman constant	*P*	Electric power (W or kW)
$\Delta\theta$	Logaritmic mean temperature difference (K)	*Pr*	Prandtl number (-)
A	Area (m^2)	*Q*	Heat output (W or kW)
c_p	Heat capacity (J/kgK)	*Ra*	Rayleigh number (-)
C	Constant	*Re*	Reynolds number (-)
g	Gravity force (N/s^2)	*T*	Temperature (°C or K)

Subscripts

0	*Design condition (without fan)*	*p*	*Primary (side)*
Fan	*Add-on-fan blower in operation*	*r*	*Return*
i	*indoor*	*rad*	*radiation*
m	*Mean*	*rel*	*Relative*
out	*outdoor*	*s*	*Secondary (side) or Supply*

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VIII

Heat Output from Space Heating Radiator with Add-on-fan Blowers

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Abstract: In a hydronic space heating system the heat output arises from natural convection and radiation from the radiators. In order to increase the heat output, without replacing the radiators with larger ones, or increasing the temperature level, the radiator can be equipped with add-on-fan blowers. In the latter case, the heat output from the radiator will increase due to increased convection. This paper describes simulations of heat output from a standard panel radiator before and after it has been equipped with add-on-fan blower.

The radiator model is made in COMSOL, using a 2D-model and the multi physic mode. The model uses General Heat Transfer and two alternate Navier-Stokes tool boxes.

With the model, heat output at different temperature levels in the radiator, and at different fan speeds, can be calculated. The aim with this work is to derive new temperature programs for the radiator at different fan speeds.

Keywords: District heating, Waterborne heating system, Space heating temperature program

1. Introduction

District heating (DH) is common in many countries and it is the dominating heating system in e.g. Denmark, Finland and Sweden. DH consists of centralized heat production units, and a piping system distributing the heat by hot water circulation. Centralized heat production allows use of waste heat from e.g. industries and combined heat and power production (CHP). In Sweden more than 30 % of the DH is produced in CHP [1].

In order to increase production and distribution efficiency the temperature level in the DH network should be kept at a low level. During heating season, the DH supply temperature is described by the temperature demand in the space heating systems of the DH connected buildings. This work describes a method to reduce the temperature demand for a hydronic space heating system with radiators. By adding an add-on-fan blower at the bottom of each radiator the heat output will increase due to an increased convection.

A reduced DH supply temperature will increase the electric production in CHP stations and lead the way towards an increased share of electricity produced by non fossil fuels.

This paper describes the modeling add-on-fan blower attached to an radiator. The goal is to show to what extent such a device can promote the reduction of the temperature program for a hydronic heating system while the heat output is left unchanged.

2. Description of the add-on-fan blower

The simulations described in this paper are based on an add-on-fan blower application mounted below a panel radiator, see Figure 1 for a schematic picture.

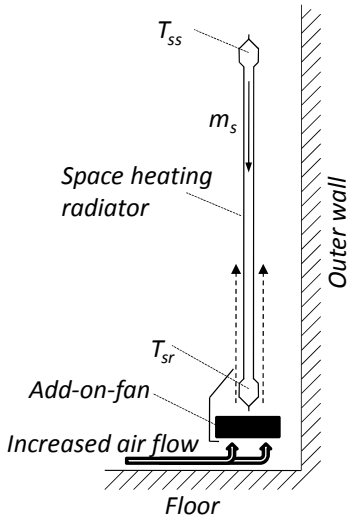

Figure 1. Principle of the add-on-fan blower mounted on a radiator.

The need for calculating and simulating the effect of an add-on-fan blower was discovered in a field study of a commercial add-on-fan blower

application carried out by the authors of this paper. The add-on-fan blower in the study was provided by a Swedish company: A-energi AB (the product is called "fläktelement" in Swedish). The company describes the features of the add-on-fan blower as a possibility to reduce the temperature program without replacing the radiators, with the aim to reduce the electricity demand for buildings supplied with heat from heat pumps. Results from the field study are presented in [7].

3. Method

In this study the radiator is represented as a 2-D model. A study of the influence of tuning parameter and mesh size has been carried out. Based on the results from the study a model of the radiator and the fan has been developed. In the studies, temperatures (radiator supply and return) are adjusted so that the total heat output is the same as with a standard temperature program when only natural convection and radiation is present.

To derive new temperature program for the radiator, two different strategies has been investigated.

1. Mass-flow through the radiator is kept constant. This results in a reduced radiator supply and return temperature while the cooling of radiator water is unchanged.
2. Supply temperature to the radiator is constant resulting in reduced mass flow though the radiator. The radiator return temperature will then decrease.

3.1 Limitations and assumptions

There is no water flow in the radiator model. A linear temperature profile in the middle of the radiator is assumed instead. No heat losses from outer wall or the floor are taken into account.

4. Use of COMSOL Multiphysics

In the paper we use COMSOL Multiphysics to model the radiator using the General Heat Transfer and Navier-Stokes toolboxes for a 2-D model. Both Weakly Compressible Navier-Stokes toolbox and the Incompressible Navier-Stokes toolbox have been used. A screenshot of the model is shown in Figure 2.

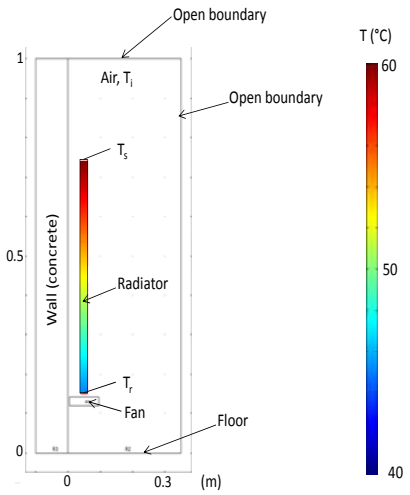

Figure 2. Picture of the 2-D COMSOL model.

In the model, the radiator is placed in a small part of a room, leaving the boundaries to the surrounding room open. The dimensions of the radiator are: height 0.59 m and width 0.02 m. The length of the radiator is 1 m. Heat output from radiation is on the left hand side surface to surface, with the concrete wall as corresponding surface, and on the right hand side, to an ambient temperature, T_i, of 21°C.

The supply temperature (the temperature at the top of the radiator), T_s, is fixed, as well as the return temperature (in the bottom of the radiator), T_r. The temperature profile along the radiator is assumed to be linear.

The fan is represented by a box where above natural buoyancy force an additional vertical force (F_{Fan}) is introduced. The dimensions of the fan in the model is: width 0.09 m, height 0.025 m and length 1 m.

5. Evaluation of model

In this section a comparison between theoretical calculations and simulation results is presented. The radiator is described as a vertical flat plate.

5.1 Heat output from a radiator

The heat output from the radiator arises from convection and radiation. The total heat transfer process from radiator water to the room through a radiator is summarized in equation 1 [2], [3].

$$Q = \dot{m}_s \cdot c_p (T_s - T_r) = k \cdot A \cdot \Delta\theta \quad (1)$$

where k is the heat transfer coefficient describing the convection from the water to the surrounding metal, conduction through the metal and convection from the outer surface of the radiator to the room according to equation 2.

$$\frac{1}{k} = \frac{1}{\alpha_{water-metal}} + \frac{\delta_{metal}}{\lambda_{metal}} + \frac{1}{\alpha_{conv} + \alpha_{rad}} \quad (2)$$

For a radiator application, the dominating parameters in this equation are the convection and radiation between the radiator and the room (α_{conv} and α_{rad}). The other terms, in this case, can be neglected. This results in a new simplified equation for energy output, see equation 3.

$$Q = Q_{rad} + Q_{conv} = \alpha_{conv} \cdot A_{conv} \cdot \Delta\theta + \alpha_{rad} \cdot A_{rad} \cdot \Delta\theta \quad (3)$$

When dealing with a radiator where the supply and return temperature are not equal, the $\Delta\theta$ is the logarithmic mean temperature difference described in equation 4. When calculating a plate with a constant temperature $\Delta\theta$ is the temperature difference between the plate (radiator) temperature and the surrounding temperature (T_i).

$$\Delta\theta = \frac{T_s - T_r}{\ln \frac{T_s - T_i}{T_r - T_i}} \quad (4)$$

5.2 Convection

Heat output from convection arises due to a temperature difference between the radiator surface and the surrounding air. The heat transfer coefficient describing this phenomena is a function of the Nusselt number (Nu), see equation 5.

$$\alpha_{conv} = Nu \cdot \lambda / h \quad (5)$$

Heat output from convection is divided into three different cases: natural, mixed and forced convection.

For natural convection, the Nu number is dependent on the Rayleigh number (Ra) which is a product of the Prandtl number (Pr) and the Grashof number (Gr). For air, Pr can be considered constant, Pr=0.71, and

$$Gr = g \cdot \beta \cdot \Delta\theta \cdot \frac{h^3}{\nu^2} \quad (6)$$
$$\beta = 1/T_{\inf} \approx 1/T_i$$

where g is the gravity force, β is the coefficient of expansion and ν is the kinematic viscosity of air.

Many empirical relations describing the Nu number are available. In this study a relation described by Churchill has been used for natural convection [4], see equation 7.

$$\overline{Nu}^{0.5} = 0.825 + \frac{0.387 \cdot Ra^{1/6}}{[1 + (0.492/\Pr)^{9/16}]^{8/27}} \quad Ra > 10^9 \quad (7)$$

In case of forced convection the Nu number is calculated by equations described by Holman [5], see equations 8 and 9.

$$\overline{Nu} = 0.664 \cdot \mathrm{Re}^{0.5} \cdot \Pr^{1/3} \quad Re < 5 \cdot 10^5 \quad (8)$$

$$\overline{Nu} = Pr^{1/3} \cdot (0.037 \cdot \mathrm{Re}^{0.8} - 871) \quad 5 \cdot 10^5 < \mathrm{Re} < 10^7 \quad (9)$$

where the Reynolds number, Re, is described as:

$$\mathrm{Re} = \frac{u \cdot L}{\nu} \quad (10)$$

The product of Gr/Re^2 describes the dominating type of convection. If Gr/Re^2>10, natural convection is dominating, if $Gr/Re^2 \approx 1$, both natural and forced convection is of importance and if $Gr/Re^2 << 1$, forced convection is dominating. When a mix of forced and natural convection occurs, the Nusselt number is calculated according to equation 11 [6].

$$\overline{Nu} = (\overline{Nu}^3_{forced} + \overline{Nu}^3_{natural})^{1/3} \quad (11)$$

5.3 Radiation

A simplified expression for heat output from radiation is described by Trüschel [3] according to equation 12.

$$Q_{rad} = \alpha_{rad} \cdot A_{rad} \cdot \Delta\theta \approx \approx 4 \cdot \frac{\varepsilon_{rad} \cdot \sigma}{\varepsilon_{rad} + \frac{A_{rad}}{A_{radiator}} \cdot (1 - \varepsilon_{rad})} \cdot T_m^3 \cdot A_{rad} \cdot \Delta\theta \quad (12)$$

In equation 5 the temperature, T_m, is the mean temperature of the radiator surface and the surfaces in the room, see equation 13. For a panel radiator the $A_{rad}/A_{radiator}$=1 [3].

$$T_m = \overline{T_{radiator} + T_{room,surface}} \approx \frac{(T_{sf} + T_{sr})/2 + T_i}{2} \quad (13)$$

Since the Arad and emissivity, ε_{rad}, are constant for a specific radiator, the relation can be simplified to equation 14.

$$Q_{rad} \approx C \cdot T_m^3 \cdot \Delta\theta \qquad (14)$$

5.4 Validation of model, natural convection

According to a calculation program provided by a panel radiator manufacturer, the total heat output from the radiator with a supply temperature of 60°C and a return temperature of 45°C is 330 W [10], which corresponds very well to simulations performed at the same temperature level, with an emissivity of ε_{rad}=0.95 and ε_{wall}=0.8.

In a study, [8], performed by van der Wijst, the heat output from a vertical plate surrounded by air is analyzed. The plate temperature is set to 40°C. Heat output at different maximum mesh sizes applied to the plate is tested for different rate of isotropic diffusion, see Figure 3.

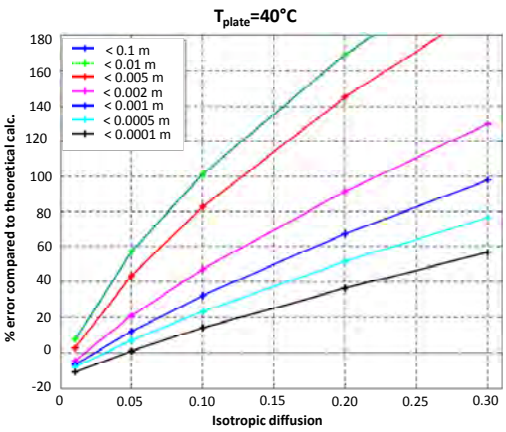

Figure 3. Result of variation of mesh size and isotropic diffusion [8].

In our model we used standard triangular mesh, with a maximum mesh size of 0.002 m on the radiator and 0.01 m in the room. This mesh size has resulted in good resemblance with theoretical calculations for both natural and forced convection by using theory from section 5.1.

5.5 Validation of model, forced convection

In order to validate the model against calculations, the influence of the tuning parameter isotropic diffusion has been in focus. The aim was to create a model giving proper results both with and without an additional fan blower. In this section, the influence of the isotropic diffusion is investigated for various forced air velocities. The simulations in this section are made with a simplified model without the concrete wall replaced with an open boundary and the floor replaced with an inlet at fixed air velocities. The radiator temperature is fixed to be 50°C (T_s=T_r=50°C) and heat output from radiation is turned off. See Figure 4 for a screenshot of the simplified model.

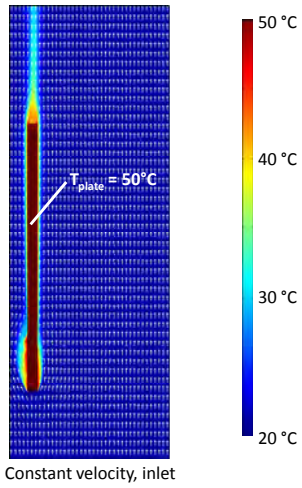

Figure 4. Screenshot of COMSOL model with constant air velocity.

In Figure 5 to Figure 7 the heat output is plotted as function of the isotropic diffusion for three different air velocities. Each figure includes results from theoretical calculations according to section 5.2 and results given by COMSOL built in equations. In case of COMSOL equations, a weight between the influence of natural and forced convection is used by using equation 11. Results from the simulations with both Weakly Compressible Navier-Stokes toolbox (chns) and Incompressible Navier-Stokes toolbox (ns) are presented.

The relative error of the simulation result compared to the results given by equations is also presented. Figure 8 shows the same type of plot, but for natural convection.

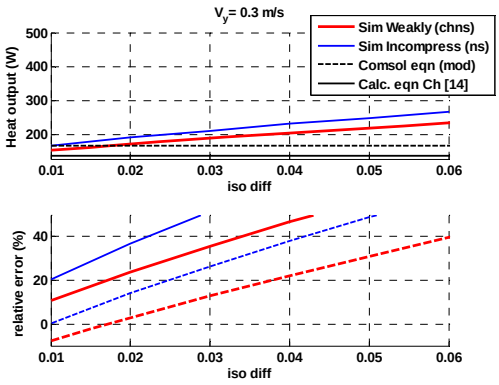

Figure 5. Heat output as a function of isotropic diffusion, forced convection, V_y=0.3m/s.

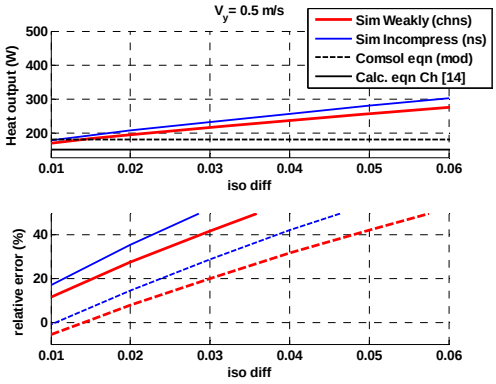

Figure 6. Heat output as a function of isotropic diffusion, forced convection, V_y=0.5m/s.

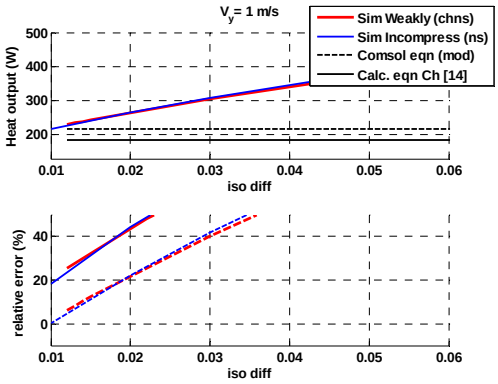

Figure 7. Heat output as a function of isotropic diffusion, forced convection, V_y=1m/s.

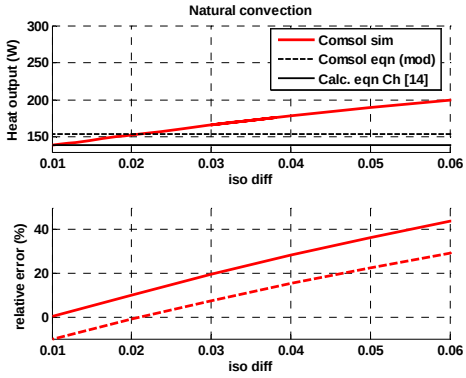

Figure 8. Heat output as a function of isotropic diffusion, natural convection.

As seen, when using a low value of isotropic diffusion, the error could be kept at an acceptable level within the air velocity range investigated. Results from the simulations in Figure 5 to Figure 7 shows that results when using the Incompressible Navier-Stokes toolbox results in slightly better correspondence to theoretical calculations than result achieved by using the Weakly Compressible Navier-Stokes toolbox, especially at air velocities higher than 0.3 m/s.

In further simulations when the fan is active, an isotropic diffusion of 0.01 and Incompressible Navier-Stokes toolbox is chosen. Corresponding simulations did not converge when using the Weakly Compressible Navier-Stokes toolbox for a vertical force in the fan higher than 15 N/m^3.For all calculations, the time dependent Direct UMFPAK solver is used with a simulation time of 259200 s (3*24 h). At that time heat output is stable for all calculations.

6. Results

New, simulated temperature programs for the radiator as a function of the relative space heating load are presented in Figure 9 and Figure 10.

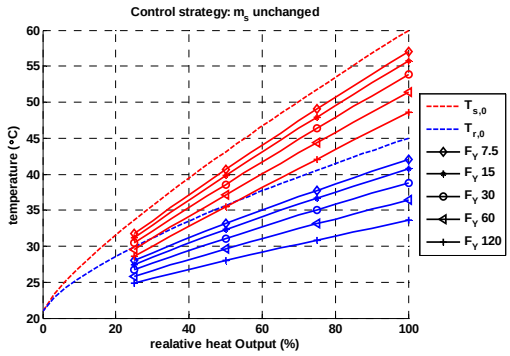

Figure 9. New temperature program for the radiator with a constant mass flow through the radiator. Red lines: supply temperature, blue lines: return temperature.

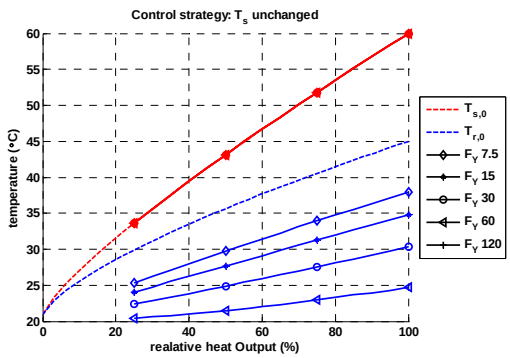

Figure 10. New return temperature for the radiator with an unchanged supply temperature program. Red lines: supply temperature, blue lines: return temperature.

The strategy resulting in constant mass flow through the radiator, see Figure 9, reduces both the supply and return temperature to the radiator with as much as 12°C at full load when high driving force is applied. When keeping the supply temperature program unchanged, and adjusting the mass flow through the radiator instead, the return temperature from the radiator is reduced even more, see Figure 10. However, one has to keep in mind that at very low mass flows through the radiator, the assumption regarding a linear temperature profile along the radiator is not valid any longer.

In Table 1 corresponding air flow through the fan for different driving forces is presented as well as the differential pressure over the fan and the corresponding work of the fan according to the relation: $P_{fan} = \Delta p \cdot V_{air}$.

Table 1: Air flow through the fan at different driving forces.

Vertical force F_y (N/m^3)	Air volume flow V_{air} (m^3/s)	Δp (N/m^2)	P_{fan} (W)
7.5	0.025	0.11	0.003
15	0.036	0.26	0.009
30	0.051	0.55	0.028
60	0.073	1.12	0.081
120	0.104	2.23	0.231

7. Conclusions/discussion

Results from this study show that by using add-on-fan blowers it is possible to reduce the space heating temperature program with several degrees C. The two simulated control strategies show that the return temperature from the space heating system can be reduced in both cases. If the mass flow through the radiator remains constant, the supply temperature is also reduced. Otherwise, with an unchanged radiator supply temperature program, the mass flow through the radiator can be reduced and the additional reduction of the return temperature from the radiator is obtained.

The simulation model shows good correspondence to theoretical calculations within the simulated range. The simulated additional heat output and possible new temperature program calculated according to the strategy with maintained flow through the radiator shows good correspondence to measurements from a field study presented in [7].

The advantage with an add-on-fan blower is that improvements can be achieved without, often costly, modifications of the heating system itself.

It is important to point out that, according to Holman, results from real applications can differ from theoretical calculations with as much as ±25% [5]. On the other hand, the calculations of the relative heat output change, which is in focus in this study, are much more reliable.

8. Nomenclature

8.1 Abbreviations

CHP Combined heat and power station
DH District heating

8.2 Variables

α	Heat transfer coefficient (W/m2.K)	Gr	Grashof number (-)
β	coefficient of expansion (K-1)	h	Height (m)
δ	Thickness (m)	k	Heat transfer coefficient (W/m^2·K)
λ	Conductivity (W/m.K)	L	Length (m)
ε	Emissisivity (-)	$\dot{m}$	mass flow (kg/s)
ν	Kinematic viscosity (m^2/s)	Nu	Nusselt number (-)
σ	Stephan-Boltzman constant	P	Electric power (W)
$\Delta\theta$	Logaritmic mean temperature difference (K)	Pr	Prandtl number (-)
A	Area (m^2)	Q	Heat output (W)
c_p	Heat capacity (J/kg·K)	Ra	Rayleigh number (-)
C	Constant	Re	Reynolds number (-)
g	Gravity force (N/s^2)	T	Temperature (ºC or K)
		V	Velocity (m/s) or volume flow (m^3/s)

7.3 Subscripts

0	Design condition (without fan)	rad	Radiation
i	Indoor (ambient)	rel	Relative
m	Mean	s	Supply
r	Return		

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IX

The impact from building heating system improvements on the primary energy efficiency of a district heating system with cogeneration

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Abstract:

The scope of this paper is to implement EU directive on Energy performance of buildings on a district heating system (DH). The impact of lowered temperatures in the connected hydronic heating systems on the primary energy factor (PEF) and the CO2 emission of a DH system was studied. The methods to lower the DH temperatures are based on previous studies. In one study, the space heating temperature demand was lowered by mounting a fan below the radiator. In another study, the control of the heating system was optimised to always provide the lowest possible DH return temperature. The impact of lower DH temperatures on the performance of a combined heat and power (CHP) station has also been investigated. This way, the primary energy factor for the DH system can be calculated by applying the power bonus method according to the EU standard. The total CO_2 emission for the DH system is also calculated with the same approach. Depending on how the electricity produced in the CHP station is considered, the PEF for DH varies. Lowered space heating temperatures substantially reduce the DH PEF, up to 25%, and CO_2 emission if electricity produced in the CHP station is assumed to replace electricity produced from coal. However, if the produced electricity is assumed to replace Nordic electricity mixture, the PEF is hardly affected at all. PEF calculations according to the power bonus method are sensitive to how the substituted electricity is considered and the temperature level in the DH network.

Keywords:

District Heating, Primary Energy, CHP.

1. Introduction

In this paper, a system analysis of a district heating (DH) network with cogeneration and a heat only boiler is described. The objective is to investigate how the total primary energy use and the total CO_2 emissions are affected by improvements in the DH-connected hydronic heating systems. The study is based on four previous papers. In three of these [8], [9] and [10], various ways to lower the return temperature in DH substations have been investigated. One way was to increase the temperature drop over a radiator by mounting a fan under the radiator in order to achieve forced convective heat transfer. The other way was to optimise the control of the heating system, by control of both supply temperature and circulation flow rate, in order to always provide the lowest possible return temperature to the DH network. In the fourth study [7], the impact of lower DH network temperatures on the performance of a specific combined heat and power (CHP) station was investigated. By applying primary energy factors and the power bonus method according to EU standards [3], the total impact on primary energy use and CO_2 emissions for the DH network, including the production units, can be estimated.

2. Primary energy

Primary energy is a measure to compare various energy sources and energy carriers based on the entire energy chain. In this way, it is possible to estimate a system's total energy use which includes extracting, processing, conversion in power plants, distribution etc. This approach is promoted by the EU [2]. The European standard EN 15316-4-5 [3] describes the procedure to estimate the

primary energy use for, e.g., a building's heating system. By applying the primary energy approach, the system boundary becomes such that not only the net energy use of, e.g., a building is considered but the total energy use caused by the building. This way, the whole world becomes the system boundary.

Primary energy is energy that has not been subjected to any conversion or transformation process. A primary energy factor (PEF) expresses how many units of primary energy are needed to supply one unit of power, heat or cold to an end user. Primary energy efficiency (PEE) refers to the total energy use for a system, from extraction to end use.

The benefit of the improvements in heating systems in the two previous studies was quantified in terms of a lowered primary return temperature, which is a central performance measure in DH technology. In the study regarding the CHP station's performance, the benefit of lowered DH network temperatures was quantified on an annual basis by estimating the change in the power-to-heat ratio. In this study, the objective is to estimate how the primary energy efficiency and CO_2 emissions for the whole DH system including heat production is affected. This is done by estimating how the DH production (mainly CHP) is affected by the suggested improvements of heating systems (fan and optimised control).

The standard [3] describes how to calculate PEFs for a DH system, comprising the production plants and the DH system. The so-called power bonus method, also described in the standard, takes additional power produced in the CHP plant into account. From an energy balance of the system, the power bonus method can be derived and written as:

$$E_p = \sum_i f_{F,i} \cdot E_{F,i} = f_{DH} \cdot E_{DH} + f_{el} \cdot E_{el} \qquad (1)$$

where

E_P = Primary energy input to the system

$f_{F,i}$ = PEF of energy carrier i

$E_{F,i}$ = Input heat energy carrier i

f_{DH} = PEF of the DH system

E_{DH} = Generated DH

f_{el} = PEF for replaced electricity

E_{el} = Cogenerated power

The cogenerated power can be assumed to replace electricity produced elsewhere in the power system. The PEF f_{el} can therefore be based on either the average, or an assumed marginal, power generation efficiency within a geographic area (e.g., Sweden, the Nordic countries or the EU).

The objective with this study is to investigate the impact from improvements in the DH-connected building heating systems on the PEF of the total DH system. Equation (1) yields:

$$f_{DH} = \frac{\sum_i f_{F,i} \cdot E_F - f_{el} \cdot E_{el}}{E_{DH}} \qquad (2)$$

The standard [3] suggests primary energy factors for some commonly used energy carriers such as oil, log and electricity produced from coal, nuclear and for a European average.

The same procedure used for calculating primary energy can be used to calculate total CO_2 emissions for a system. In other words, it is possible to estimate the total amount of fossil fuel derived CO_2 emitted per unit energy supplied for end use. Instead of using PEF for each energy carrier in the system, corresponding CO_2 emission coefficients are used in the equations above.

In equation (2), there are three parameters that affect the PEF for the DH system: E_F, f_{el} and E_{el}. We affect them by introducing the add-on-fan blower and the radiator optimisation, respectively. The amount of delivered heat, E_{DH}, and the fuel used in the DH production, f_F, is held constant. As mentioned above, f_{el} depends on the view on the replaced electricity production while E_{el} depends on how the CHP station is affected by changes in the DH network temperatures.

As will be explained further on, the impact on f_{DH} due to other changes has also been taken into account, such as lower heat losses in the DH network as a result of lower network temperatures, reduced pump energy as a result of reduced flows, both in the network as well as in the secondary systems.

3. Previous studies

In this section follows a short review of the previous studies that the present study is based on.

3.1. Impact of lowered DH network temperatures on CHP

The use of cogeneration is promoted by the EU in Directive 2004/8/EC [4]. In a study by Johansson et al [7], the impact of lowered DH network temperatures on the performance of the heat and power production in a CHP-based DH network was investigated. The calculations of the bio-fuelled CHP station were performed with an off-design model in the heat and mass balance software IPSEpro. The model was verified with data from the CHP station in the Swedish town Enköping.

It was found that a lowered DH network supply temperature has a greater impact on the power-to-heat ratio than a lowered return temperature. The potential to lower the return temperature is, however, generally much larger. A return temperature reduction could, to some extent, be transformed into a reduction of the supply temperature. On the other hand, if only the return temperature is reduced, the flow rate in the network will decrease which can be of interest in order to eliminate bottlenecks in the network, save pump energy or to allow further expansion of the existing network.

3.2. Add-on-fan blower

Johansson and Wollerstrand [8], [9] have investigated the possibility to enhance heat transfer from a radiator in a hydronic heating system by mounting a fan under the radiator. In this way, forced convection takes place and the temperature drop in the radiator will increase substantially. This also gives the possibility to lower the supply temperature in the heating system. In Figure 1, an illustration of the add-on-fan blower mounted on a panel radiator is displayed.

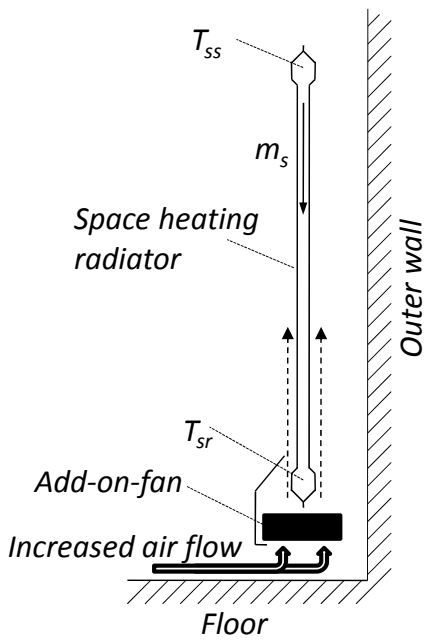

Fig. 1. Principle of add-on-fan blower.

The extent to which the space heating system temperatures can be lowered when the add-on-fan blowers are used is dependent on the speed of the fan. In Figure 2 new temperature programmes are displayed for two different fan speeds together with the reference temperature programme of

60/45°C at DOT. The calculations of the new space heating temperatures are presented by Johansson in [9]. The electricity usage of the add-on-fan blowers in Figure 2 is 1% and 0.1% of the heat output at DOT, respectively. The heat output from a panel radiator increases with approximately 60% with the higher fan speed.

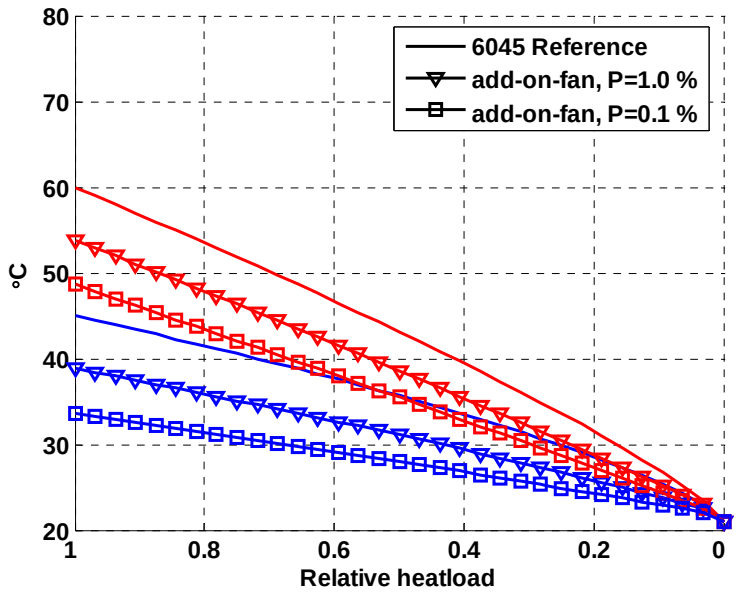

Fig. 2. Space heating system temperatures with and without add-on-fan blowers.

Lowered space heating temperatures means a lowered DH return temperature. The lowered space heating temperatures also make it possible to lower the DH supply temperature without increasing the DH flow rate. This strategy is presented by Johansson and Wollerstrand in [8].

3.3. Optimum space heating control

Paper [10] describes how the control of a radiator system can be optimised to provide the lowest possible DH return temperature. The radiator circuit's supply temperature and circulation flow rate are varied in a systematic way at different operating points in the DH substation. Such an algorithm can gradually create a modified control curve for the radiator circuit that always provides the lowest return temperature.

In Figure 3, the optimum secondary supply temperature and flow are shown, assuming the primary supply temperature of the network in Enköping, and the resulting primary return temperature.

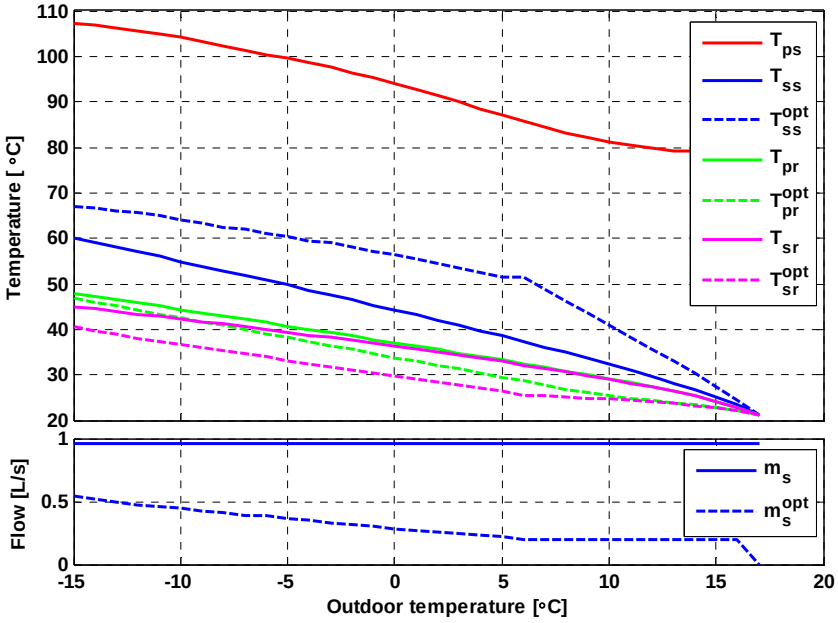

Fig. 3. Space heating system temperatures. Original 60/45°C temperature programme with associated constant flow rate in solid lines. Optimised temperature programme, with associated variable flow rate, in dashed lines.

The resulting primary return temperature from the optimisation is used in the power bonus method calculations. The flow-weighted yearly average primary return temperature reduction estimates to 3.1°C. In addition to the lowered DH return temperature, the primary flow rate is decreased, which results in a reduction of DH pump energy consumption.

4. Assumptions for present study

There are several factors that affect the total PEF for the DH system. As already mentioned, the CHP station benefit from lowered system temperatures, but the DH network is also affected. Lowered temperatures in the network means lowered heat losses. Reduced flow rates, in the network as well as in the secondary systems, means reduced pump energy. In this section, all assumptions, which will influence the total PEF, are listed.

The DH network is assumed to consist of 1 150 buildings with 20 flats each, which corresponds to the annual heat load in the DH system in Enköping. Each building is equipped with a parallel-coupled DH substation. The space heating systems are assumed to be designed for 80/60°C, but oversized and adjusted to operate at 60/45°C at design load. The models for the heat exchange in the substation are thoroughly described in, e.g., [6] and [13].

Concerning pump energy savings from reduced flows in the DH network and in the space heating systems, consider the fact that the required pressure increase in the pump, ΔP, is proportional to the square of the volume flow in the system, $\dot{V}$. The power required by the pump is in turn proportional to the pressure increase and the volume flow, which means that the required pump energy is a function of the cube of the flow rate.

The pump energy after a reduction of the volume flow (denoted by index 1) in relation to the flow before (index 0) can then be expressed as:

$$\dot{W}_{p,1} = \dot{W}_{p,0}\frac{\Delta P_1 \cdot \dot{V}_1}{\Delta P_0 \cdot \dot{V}_0} = \dot{W}_{p,0}\frac{k \cdot \dot{V}_1{}^2 \cdot \dot{V}_1}{k \cdot \dot{V}_0{}^2 \cdot \dot{V}_0} = \dot{W}_{p,0}\frac{\dot{V}_1{}^3}{\dot{V}_0{}^3} \qquad (3)$$

Consequently, the power needed for pumping decreases with the cube of the flow reduction, whether the reduction takes place in the radiator system or in the DH network. The design power for the space heating circulation pump in a building is 100 W. The pump energy in the DH network is assumed to be 0.3 percent of the total DH load [5].

The heat losses in the DH network are assumed to be 10 percent of the heat load on an annual basis. They are assumed to vary according to:

$$Q = U \cdot A \cdot (T_{ps} - T_g) + U \cdot A \cdot (T_{pr} - T_g) \qquad (4)$$

where

U = Overall heat transfer coefficient for the pipes

A = Area for the pipes

T_{ps} = Supply temperature

T_{pr} = Return temperature

T_g = Ground temperature

This relation is a simplification, where the heat transfer coefficient and the ground temperature are assumed constant. However, the influence of the heat losses on the total PEF is rather small and the main purpose was to have a simple model for how the heat losses vary with the heat load.

The DH production is based on the DH system in Enköping, but is slightly simplified in order to make the study more transparent and easier to follow, c.f. Figure 4. The main source of heat production is the CHP station. A heat-only-boiler is used for low heat load (during summer) and for peak load. Both the CHP station and the heat-only-boiler are assumed to have a thermal efficiency of 90 per cent.

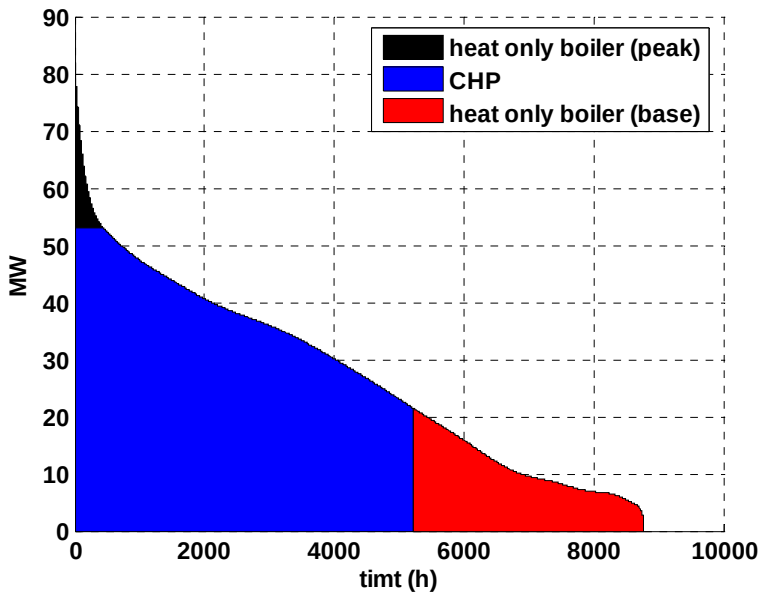

Fig. 4. Duration of the DH production units.

The supply and return temperatures in the DH network are displayed in Figure 5.

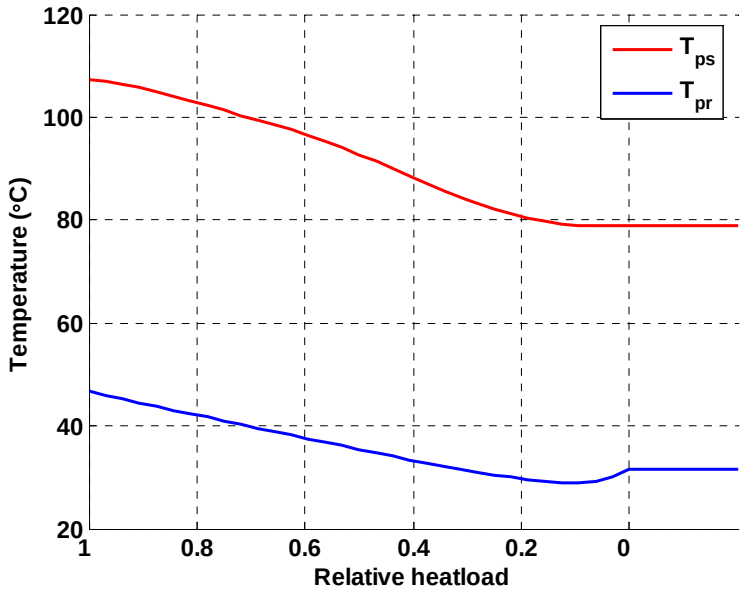

Fig. 5. Supply and return temperatures in the DH network.

Biomass (log) is used as fuel in both the CHP station and in the boiler. According to [3], the PEF is 1.09 for log and this is the value used in the calculations. The corresponding CO2 emission is 14 kg/MWh. Different PEFs for electricity have been considered, representing different views on the electricity assumed to be replaced by the CHP station. All figures are taken from [3], except the average Nordic electricity mix which is derived with data from [1], c.f. Table 1. Another possibility to determine PEF for the average Nordic electricity mix is to use PEF for each input energy carrier to the electricity production and to use an electricity substitution approach when electricity is produced in combined cycles. The PEF for electricity produced in combined cycles could then be considered as replacing electricity produced on the margin with coal. This approach will result in a higher PEF for the average Nordic electricity mix (PEF=2.35) [12]. In this paper this approach is not appropriate since different substitution methods are compared with each other.

Table 1. Primary energy factors and corresponding CO_2 emissions for the DH production fuel (biomass) and for the different compositions of electricity production replaced by the CHP station. Figures taken from [3] except Electricity (Nordic mix) which is derived with data from [1]

Energy carrier	PEF [-]	CO_2 emission [kg/MWh]
Biomass (log)	1.09	14
Electricity (Nordic mix)	2.15	13.5
Electricity (UCPTE mix)	3.31	617
Electricity (Coal)	4.05	1340

3.3. Alternatives to lower DH temperatures

The PEF calculations for the DH system have been performed for five alternatives with respect to how to implement the add-on-fan blowers and the radiator optimisation. The alternatives are listed in Table 2. Alternative 1 and 2 represents the case when the original primary supply temperature is considered and the gain achieved with the add-on-fan blower is transferred into a lowered primary return temperature and a reduced primary flow. Alternative 1 represents a higher power supply to the fan (1 percent of the design heat load), while alternative 2 represents a lower power supply (0.1 percent). The radiator optimisation is represented by alternative 3 and results in lowered primary return temperature and primary flow. The last alternatives, 4 and 5, represent the hypothetical alternative that the gain from the use of add-on-fan blower enables for a lower primary supply temperature to be used in the DH network.

Table 2. Various alternatives for the implementation of add-on-fans and radiator optimisation

	Type of implemented measure	Strategy	Reduced
1	Add-on-fan, high fan power (1% of DOT)	Keep original T_{ps}	T_{pr}, m_p
2	Add-on-fan, low fan power (0.1% of DOT)		
3	Radiator optimisation		
4	Add-on-fan, high fan power (1% of DOT)	Keep original m_p	T_{ps}, T_{pr}
5	Add-on-fan, low fan power (0.1% of DOT)		

Another consideration regarding various calculation alternatives was whether electricity use for pumps in the DH network and in the radiator systems, as well as for the add-on-fan blowers, should be considered in the PEF calculations. According to the standard [3], the PEF for the DH system should be calculated with the DH production and distribution network taken into account but not what happens inside the connected buildings. The idea of using add-on-fan blowers inevitably involves the use of additional electricity. At the same time, alternatives 1, 2 and 3 leads to a reduction of the required pump energy, both on the secondary, as well as on the primary, side. To take this into account, we chose to perform the calculations both without taking electricity use into account, referred to as system boundary 1, and with electricity use taken into account, referred to as system boundary 2. Changes of electric power production in the CHP station due to lowered DH temperatures are always considered.

The next consideration concerns the control of the add-on-fans. The first strategy is to constantly keep the fans running. However, there will be an operational point above which fan operation is no longer beneficial since the additional electricity consumption exceeds the additional power production in the CHP station. Therefore, the other strategy regarding fan control is to turn them off whenever the electric consumption exceeds the additional power production.

5. Results

Figure 6 displays the primary return temperature reduction achieved from implementing optimisation alternatives 1, 2 and 3, i.e., while keeping the original primary supply temperature.

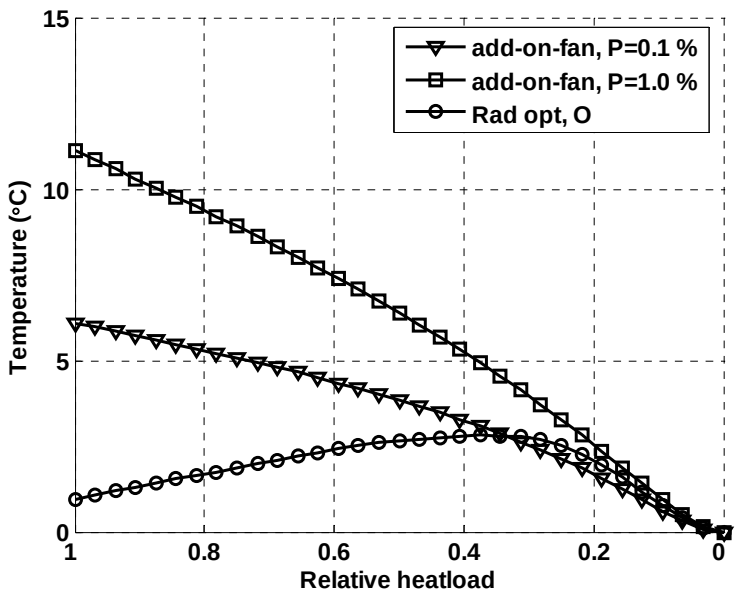

Fig. 6. Return temperature reduction with alternatives 1, 2 and 3, i.e., unchanged DH supply temperature.

Figure 7 displays the outcome of use of add-on fan blowers according to alternatives 4 and 5, i.e., the possible primary supply temperature reduction if the secondary supply temperature is lowered

and the primary flow is held constant. In this case, both primary supply and return temperatures will be equally lowered.

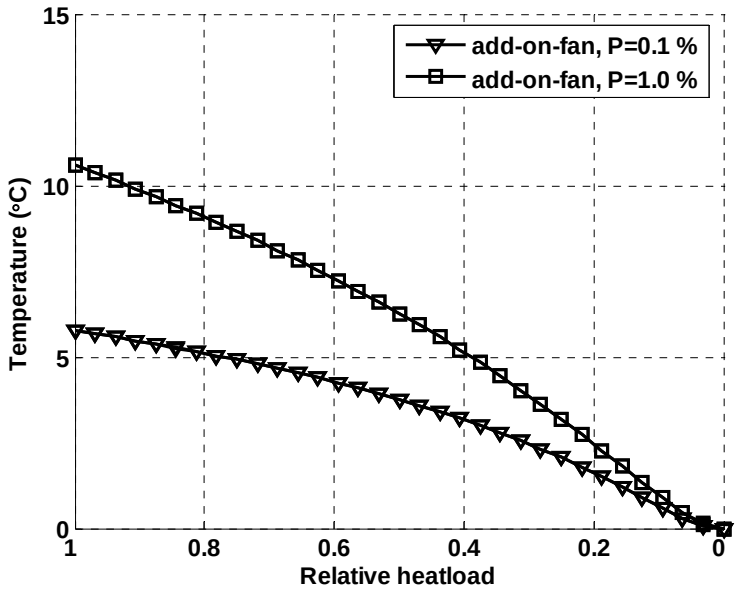

Fig. 7. Achievable primary supply temperature reduction with alternatives 4 and 5, i.e., unchanged primary flow rate.

Let us now have a look at how the PEF for the DH system changes when the add-on-fan blower is in use according to alternative 4. Figure 8 displays the PEF as a function of relative heat load in the DH network.

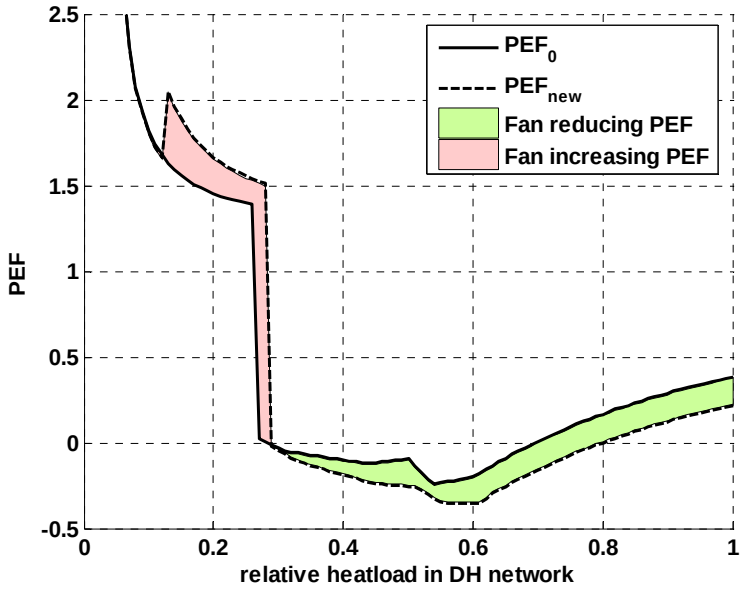

Figure 8 PEF as function of relative heat load with implementation of add-on-fan blowers with an electricity consumption of 1%. Substituted electricity considered as UCPTE mix.

The PEF is substantially higher for low heat loads, i.e., when the additional electricity consumption by the add-on-fan blowers exceeds the additional electricity production in the CHP station or when only the heat-only-boiler is in use. As can be seen from the diagram, the use of add-on-fan blowers reduce the PEF most of the time, i.e., as long as the heat load is large enough.

For each combination of calculation alternatives (fan, with return or supply temperature reduction, or radiator optimisation), system boundary (without or with electricity use) and fan control strategy (always on, denoted as 'On', or turned off when not useful, denoted as 'Off'), the PEF for the DH system was calculated. The results for optimisation alternatives 1, 2 and 3 are displayed in Table 3. Also shown in the table are the corresponding CO_2 emissions.

Table 3 Calculated primary energy factors and corresponding CO_2 emissions for alternatives 1, 2 and 3.

		Nordic				UCPTE				Coal			
		System boundary 1		System boundary 2		System boundary 1		System boundary 2		System boundary 1		System boundary 2	
		PEF	CO_2	PEF	CO_2	PEF	CO_2	PEF	CO_2	PEF	CO_2	PEF	CO_2
Ref. case		1.00	18.0	-	-	0.578	−203	-	-	0.307	−467	-	-
1	On	−1-0%	~0%	5%	0-2%	−2-0%	−1-0%	12%	6%	−5%	−1-0%	28%	6%
	Off	~0%	~0%	~0%	0-1%	−2-0%	−2-0%	−1-0%	−1-0%	−5%	−1-0%	−2-0%	−1-0%
2	On	~0%	~0%	~0%	~0%	−1-0%	−1-0%	~0%	~0%	−3%	~0%	~0%	~0%
	Off	−1-0%	~0%	~0%	~0%	−2-0%	−1-0%	−2-0%	−1-0%	−5%	~0%	−4%	−1-0%
3	-	−1-0%	~0%	−1-0%	~0%	−2-0%	−1-0%	−2-0%	−2-0%	−5%	−1-0%	−6%	−1-0%

In Table 4, the results for optimisation alternatives 4 and 5 are displayed, that is, when both DH supply and return temperature are lowered according to Figure 7.

Table 4 Calculated primary energy factors and corresponding CO_2 emissions for alternatives 4 and 5.

		Nordic				UCPTE				Coal			
		System boundary 1		System boundary 2		System boundary 1		System boundary 2		System boundary 1		System boundary 2	
		PEF	CO_2	PEF	CO2	PEF	CO_2	PEF	CO_2	PEF	CO_2	PEF	CO_2
Ref. case		1.00	18.0	-	-	0.578	−203	-	-	0.307	−467	-	-
4	On	−5%	~0%	0-1%	0-2%	−16%	−12%	−2-0%	−5%	−41%	−12%	−7%	−4%
	Off	−5%	0-1%	−1-0%	0-2%	−17%	−13%	−8%	−8%	−43%	−12%	−21%	−8%
5	On	−3%	~0%	−2-0%	0-1%	−10%	−8%	−9%	−7%	−25%	−7%	−22%	−6%
	Off	−3%	~0%	−3%	0-1%	−11%	−9%	−10%	−8%	−28%	−7%	−26%	−8%

6. Conclusions and discussion

Depending on which type of electricity we assume to be replaced, we end up with completely different PEFs for the DH system. The higher the value of the PEF and CO_2 emissions for the electricity, the lower become the resulting PEF and CO_2 emissions for the DH system. This is especially evident in the extreme cases Nordic electricity and electricity based on coal, respectively. When the Nordic electricity composition is used, the PEF is hardly affected by the primary temperatures. In fact, the CO_2 emission factor can even increase because the CO_2 factor is lower than the CO_2 factor for the fuel used in the CHP station.

By using a marginal electricity approach, which means that the replaced electricity is assumed to be produced in condensing power stations fired by coal, the PEF for the DH system can be reduced with 25-40% with utilisation of add-on fan blowers if both the supply and the return temperature in the DH network are reduced. This is, if the additional electricity consumption from the add-on-fan blowers is not considered. If the latter is taken into account, the PEF still can be reduced with 20-25%, provided that the fans are in operation only when they are beneficial. It becomes especially important to turn off the add-on-fan blowers when the higher fan speed is used.

For the case with a lowered DH return temperature and flow rate, the PEF for DH is not reduced to the same extent as when both DH supply and return temperatures are lowered. However, the PEF for the DH system can be reduced with approximately 5% if the substituted electricity is considered as electricity produced in coal-fired power stations. This is the case both with the use of add-on-fan

blowers at the higher rotational speed, as well as with implementation of an optimised control of the space heating system.

If the substituted electricity is considered according to the European mixture a reduction of the primary return temperature could decrease with 2 percentages with an optimized space heating control algorithm. Utilization of add-on-fan blowers in order to reduce both the DH supply and return temperature is will decrease the PEF for DH with up to 10% even if the additional electricity consumption is included.

The additional electric consumption that arises from utilisation of add-on-fan blowers is of great importance to the total PEF for DH. The results show that add-on-fan blowers can be used to reduce the PEF, even with increased electricity consumption. The results also show the importance of a control of the fan operation, especially when the higher fan speed is used in order to further reduce the PEF. In this paper, a simple on-off control was used, where the fans are turned off when the electricity consumption of the fans exceeds the additional electricity production in the CHP station.

It should also be mentioned that the results are valid for the assumptions previously listed, i.e., regarding the used fuel, efficiencies, heat losses, pump energy etc. However, the influence of these parameters is small compared to the composition of the electricity replaced by the CHP production.

Nomenclature

A	area, m^2
E	primary energy input/output to the system, W
$f_{F,i}$	primary energy factor of energy carrier i, -
k	constant, -
m	mass flow, L/s
ΔP	differential pressure, Pa
Q	Energy, W
T	temperature, °C
U	heat transfer coefficient, $W/(m^2K)$
$\dot{V}$	volume flow, m^3/s
$\dot{W}_p$	pump power, W

Subscripts

0	initial conditions
1	second state condition
el	electricity
F	fuel
g	ground
p	primary
r	return
s	secondary or supply

Abbreviation

CHP	combined heat and power
DH	district heating
DOT	design outdoor temperature

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Summary of report A

Cavitation in control valves

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This is a summary of the Swedish report *"Kavitation i styrventiler - laboratorieundersökning" (Cavitation in control valves – laboratory test)* which is a laboratory study of cavitation in three different control valves commonly used in DH systems. Two sizes of each valve type were tested.

Introduction to cavitation in control valves

Control valves are used in district heating (DH) substations to control the DH flow through heat exchangers in order to adjust the heat output to the secondary circuits. In Sweden, the control valves are exposed to rather high differential pressures, especially during the winter season and if situated close to DH production units. In other countries, such as Germany, differential pressure controllers are commonly used in order to avoid high pressure drops and large pressure drop variation over the substations. This is uncommon in Swedish DH systems and, even though the differential pressure over the control valve can exceed 6-8 bars, cavitation is not generally considered to be a problem. However, according to data sheets from control valve manufacturers, cavitation can occur in this pressure range.

Cavitation occurs when the flow through the control valve is throttled and the pressure locally decreases below the vapour pressure (p_v) immediately after the narrowest passage (see Figure 1). The most obvious disadvantage of this phenomenon in the DH context is the noise generated in the valve. The noise may be transmitted further in the secondary installations. Cavitation may also reduce the valve's capacity and, depending on the intensity of the erosion, significantly reduce the lifetime of the control valve. Even if the valve still works, erosion of the valve plug and the seat can cause distorted control behaviour and considerable leakage in closed position, which is highly undesirable.

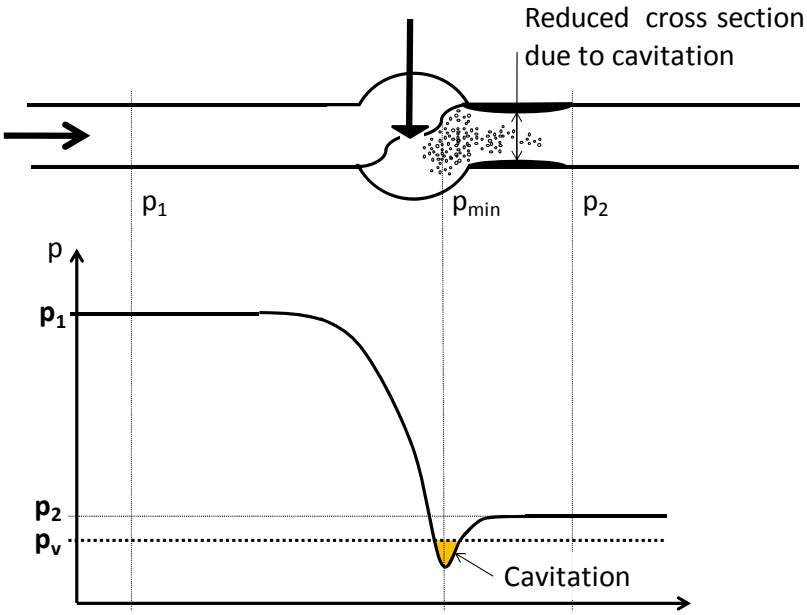

Figure 1. Principle of cavitation in control valve.

As seen in Figure 1, the pressure level before and after the control valve strongly affects the tendency for cavitation. The fluid temperature is also of importance as it affects the vaporization pressure of the fluid.

Strategies to reduce cavitation

Depending on their design, control valves may be more or less sensitive to cavitation. The design aspects of control valves are not investigated within this study, however it can be mentioned that multistage flow throttling in the valve is an effective method to prevent cavitation from occurring.

For the given geometry, flow temperature and pressure in the control valve are crucial for cavitation. From this point of view and in the DH system perspective there are three methods to decrease the risk of cavitation:

a) Lowering the temperature level in the DH system / increasing the cooling of the DH primary water
b) Increasing the pressure level in the DH return pipe
c) Installation of differential pressure controllers or pre-throttling before the control valve to reduce the differential pressure over the valve (multistage throttling)

See Figure 2 for an illustration of the three methods.

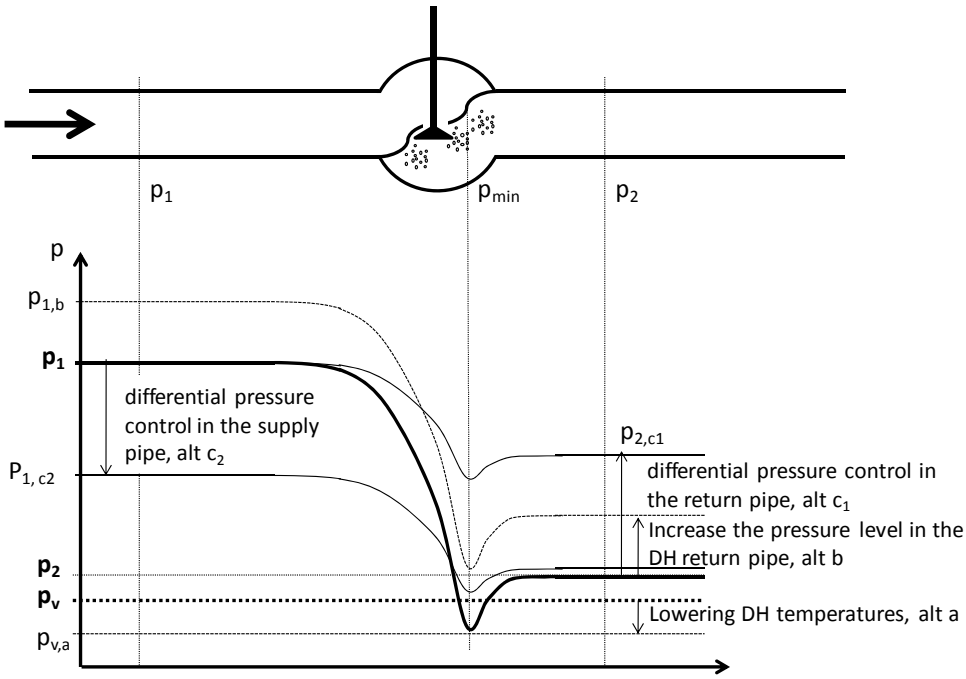

Figure 2. Illustration of the three methods to reduce the risk of cavitation in control valves.

Experimental study – cavitation in control valves

As already mentioned, three different types of control valve were experimentally tested for cavitation. See Figure 3 for a picture of the tested valves. Details of the valve types and sizes (k_{vs}-values) are given below the figure.

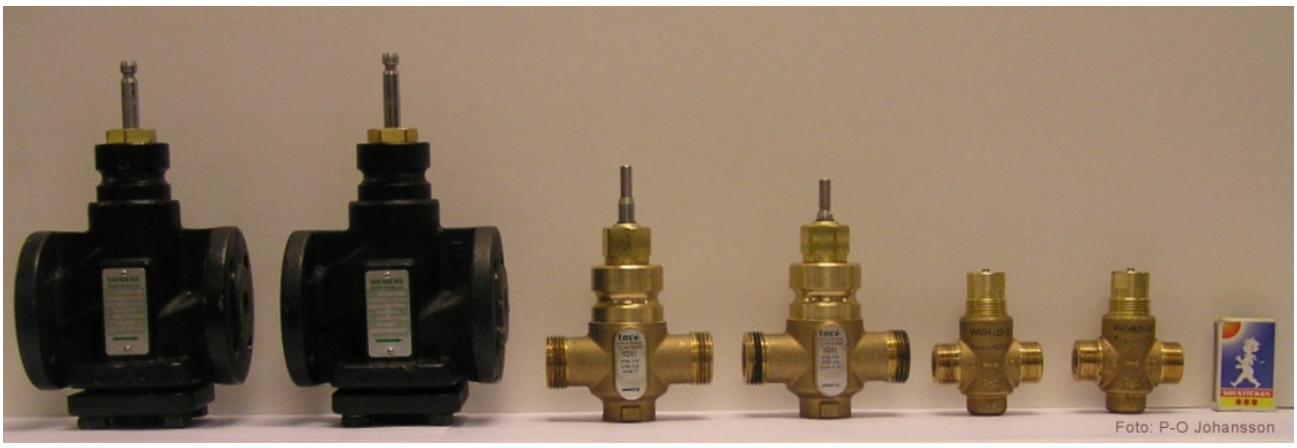

Figure 3. The tested control valves. From left: Siemens VVF 52.15, PN 25, DN 15, k_{vs} 1.0 and k_{vs} 1.6, TAC V241, PN 16, DN 15, k_{vs} 1,0 and k_{vs} 1,6, Siemens VVF 52.15, PN 25, DN 15, k_{vs} 1.0 and k_{vs} 1.6.

The valves were fitted in a test rig and tested for a fixed number of degrees of opening and different differential pressures as well. The differential pressure was varied within the range of 1 to 10 bars in steps of 0.5 bar. The opening range was 10% (z=0.1) and fully open (z=1) in steps of 0.1. All the tests were performed at two different fluid temperatures (50 and 100°C).

Detection of cavitation

Cavitation is usually detected by noise registration. Typical cavitation starts from an 8 kHz noise signal; however, massive cavitation can also be detected at lower frequency [Koivula].

The noise from the valve was registered by a microphone attached to the valve housing. Also, fluid vibrations immediately after the control valve were registered using an accelerometer sensor (see Figure 4).

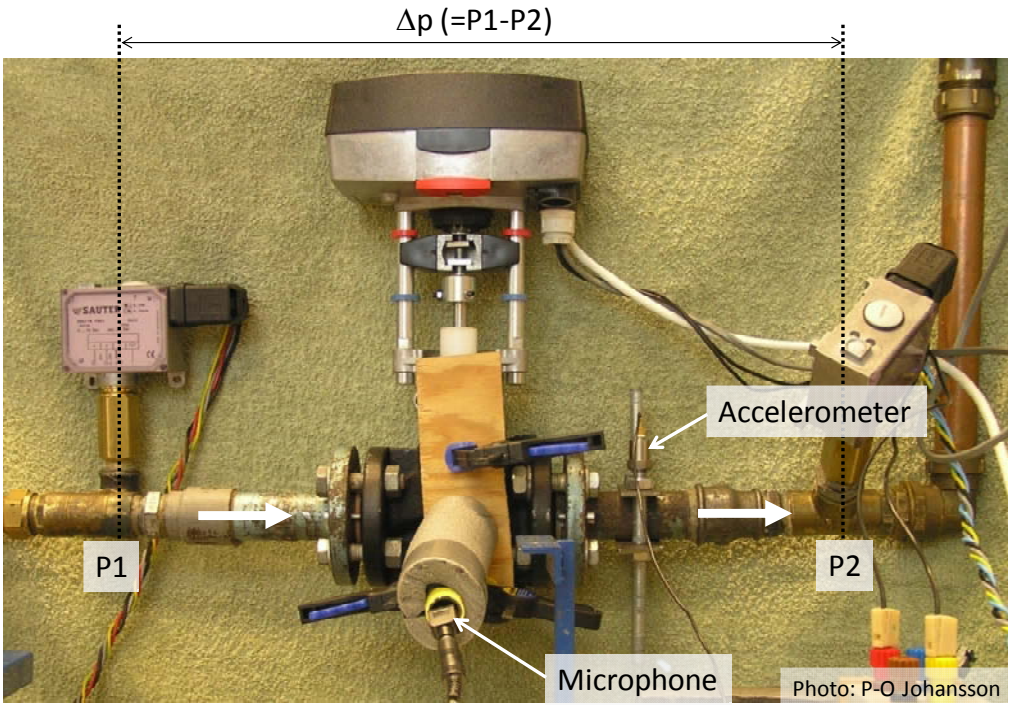

Figure 4. Control valve fitted in the test rig.

By analysing the signal from the microphone and the accelerometer, cavitation can be detected. Figure 5 shows an example of signal registration. It can be seen that the slope of the logarithmed accelerometer signal increases greatly when cavitation starts (upper picture), and decreases when the cavitation is developed. The lower part of the figure shows the amplitude of the noise and how it increases with the development of the cavitation.

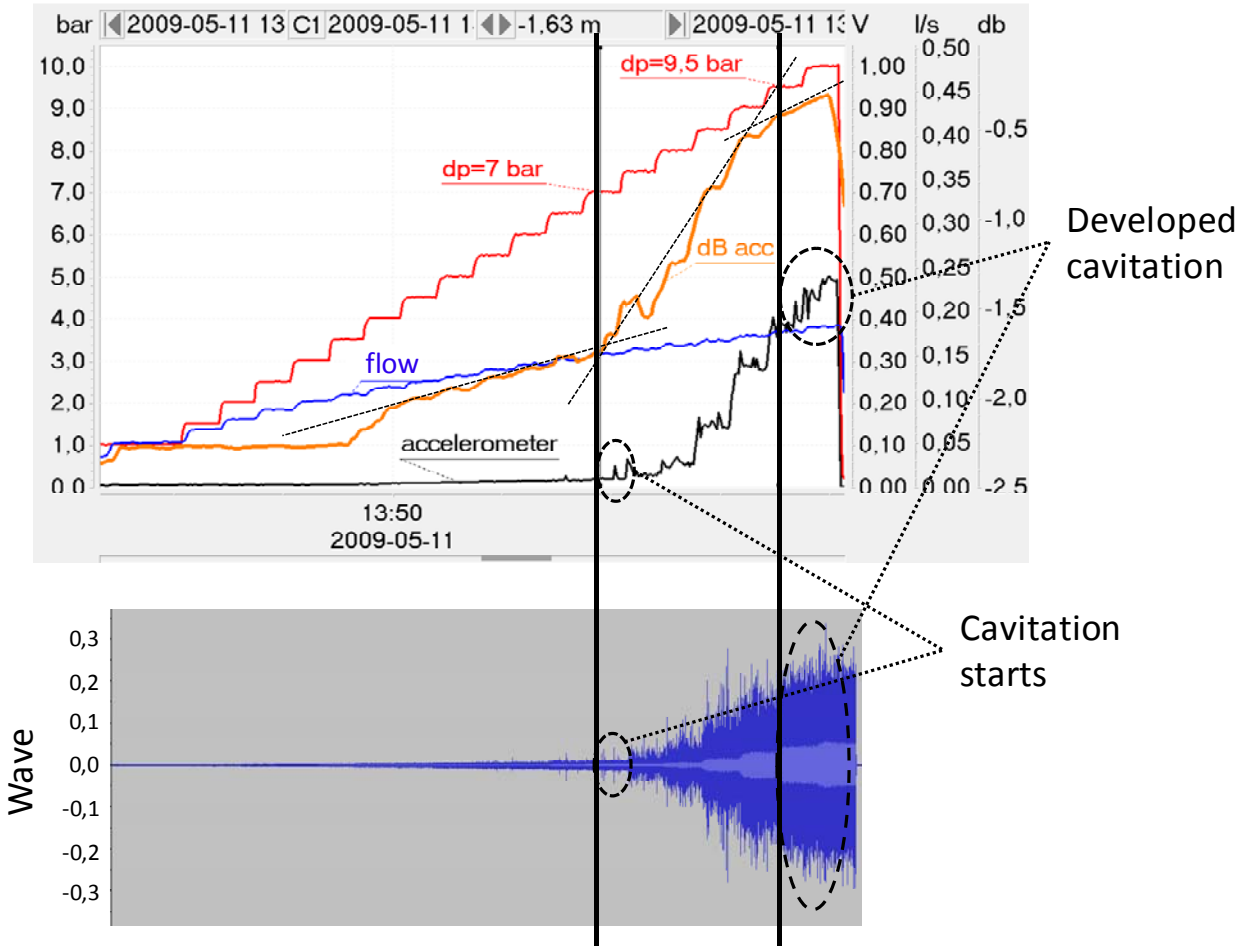

Figure 5. Example of experimental measurements.

Results

Cavitation occurred below 10 bar pressure drop in most of the tests. The theoretical consideration that the risk of cavitation increases with increasing flow temperature was confirmed. The degree of valve opening and the differential pressure over the valve strongly affects the point at which cavitation starts. In general, the sensitivity to cavitation increases with the opening and with differential pressure for all the valves tested. However, the increase depended on valve type and design (see Figure 6).

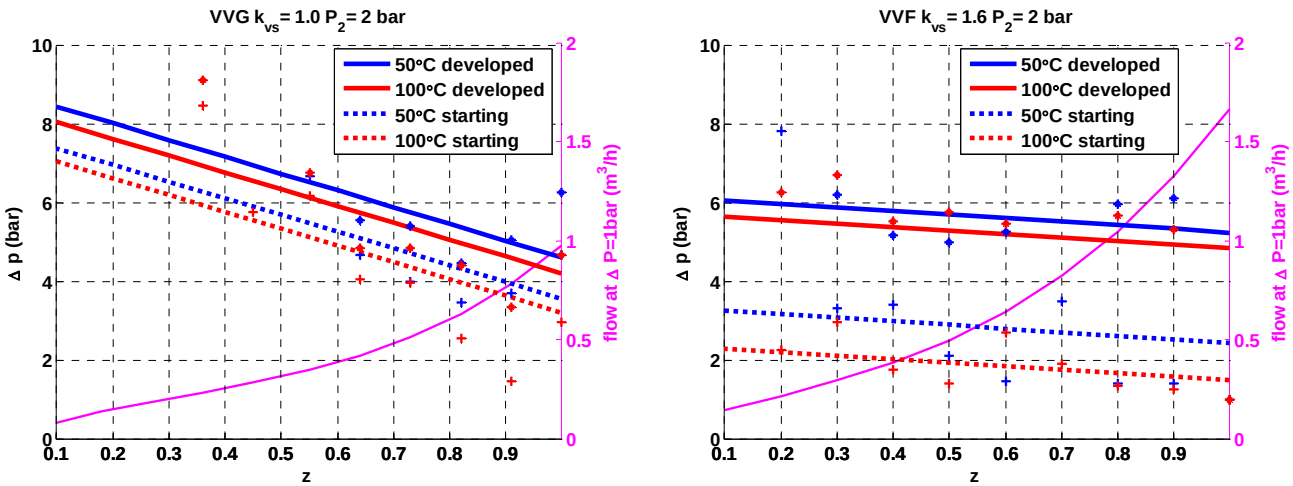

Figure 6. Example of cavitation for two different control valves and flow temperatures.

The two valves presented in the figure differ in construction; the valve to the right has a plug of multi-hole cage-type, which results in less dependency on the degree of opening. Flow temperature dependency is also detected. None of the tested valves choked during cavitation (there was no fluid evaporation after the valve).

A practical conclusion was that the valves used in Swedish DH systems are probably subject to limited cavitation during unfavourable operating conditions. Oversizing of control valves in DH substations may be a reason for the very low frequency of valve failures reported as caused by cavitation. Attention should be paid to the fact that the current trend to reduce oversizing of the valves in DH substations may lead to increased risk of failures.

A more detailed description of the cavitation test and the results can be found in the full Swedish report "*Kavitation i styrventiler - laboratorieundersökning*" by Per-Olof Johansson and Janusz Wollerstrand, Report 2009:45, Swedish District Heating Association.

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