International Energy Agency

IEA District Heating and Cooling

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

EFFICIENT SUBSTATIONS AND INSTALLATIONS

Published by

Novem

Acting as operating agent for the IEA District Heating and Cooling project

1996: N5

IEA - District Heating and Cooling Project

EFFICIENT SUBSTATIONS AND INSTALLATIONS

(ESI)

by

•Rune Volla •Rolf Ulseth •Jacob Stang

••Svend Frederiksen

•••Allan Johnson ••••Robert Besant

SINTEF
•The Foundation for Scientific and Industrial Research
at The Norwegian Institute of Technology

LTH
••Lund Institute of Technology, Sweden

•••EnEff Technologies Inc., Canada
••••The University of Saskatchewan

PREFACE AND ACKNOWLEDGEMENTS

The International Energy Agency (IEA) was established in 1974 within the framework of the OECD to implement an International Energy Program. A basic aim of the IEA is to strengthen the cooperation between the member countries in the energy field. One element of this cooperative activities is to undertake energy research, development and demonstration (RD&D).

District Heating is, by the IEA, seen as a means by which countries may reduce their dependence on oil and improve their energy efficiency. It involves increased use of indigenous or abundant fuels, the utilization of waste energy and combined heat and power production.

The IEA "Program of Research, Development and Demonstration on District Heating" was established at the end of 1983. Under Annex I, ten countries participated in the program: Belgium, Canada, Denmark, Federal Republic of Germany, Finland, Italy, The Netherlands, Norway, Sweden and USA.

The National Energy Administration, Sweden was Operating Agent for the program under Annex I, in which the following technical areas were assessed:

- Development of heat meters
- Cost efficient distribution and connection systems for areas of low heating density
- Small size coal-fired hot water boiler
- Medium size combined heat and power plants
- Low temperature applications in district heating systems

The results of these topics have been presented in printed reports published by the National Energy Administration, Sweden.

In 1987 it was decided by nine of the original ten participating countries (ex. Belgium) to continue the implementation of cooperative projects under an Annex II. The Netherlands Agency for Energy and the Environment (NOVEM), was Operating Agent for Annex II, in which the following technical areas were assessed:

- Heat meters
- Consumer installations
- Piping
- Advanced fluids
- Advanced heat production technology
- Information exchange

In 1990 the cooperating countries decided to continue the implementation of new cooperative projects under a new Annex III. During this annex United Kingdom joined the project. NOVEM was Operating Agent also for Annex III, in which the following areas have been assessed:

- District Heating and the Environment
- Supervision of District Heating Networks
- Advanced Fluids
- Piping
- District Energy Promotion Manual
- Consumer Heating System Simulation

The results from Annex II and III have been presented in printed reports published by NOVEM.

In 1993 the cooperating countries (ex. Italy) decided to continue the implementation of new cooperative projects under a new Annex IV. The name of the main cooperating project was now changed to "IEA - District Heating and Cooling Project" which emphasise the increasing awareness of District Cooling as an energy efficient technology. During this annex The Republic of Korea joined the project. NOVEM has been Operating Agent also for Annex IV, in which the following technical areas have been assessed:

- Combined Heat and Power/Cooling Guidelines
- Advanced Transmission Fluids
- Piping Technology
- Network Supervision
- **E**fficient **S**ubstations and **I**nstallations (**ESI**)
- Manual on DH-piping, Design and Construction
- Development of long term Cooperation with East-European Countries

This report describe the project called "**E**fficient **S**ubstations and **I**nstallations" (**ESI**).

The work on the ESI-project has been monitored by the "IEA-Experts Group on ESI" (EG) with Associate Professor Rolf Ulseth from The Norwegian University of Science and Technology (NTNU) as project leader and "Chairman of EG-ESI".

The members of "EG-ESI" have been:

- Tom Onno (Canada)
- Benny Böhm (Denmark)
- Veli-Matti Mäkelä (Finland)
- Huub Stroeken (The Netherlands)
- Rolf Ulseth (Norway)
- Audun Årøen (Norway)
- Gunnar Nilsson (Sweden)
- Paul S Woods (UK)

The Chairman wants to thank everybody who has contributed and made it possible to carry through this work - especially every individual of the EG for making a good effort and showing a positive will to cooperate. A special thank to Rune Volla, Svend Frederiksen and Allan Johnson for their contribution to the joint work and in the writing of the report. Thanks should also be given to the "Executive Committee" who gave priority to do work on the ESI-project.

On behalf of SINTEF I will also take this opportunity to thank "The Research Council of Norway" for the financial support that made possible our participation in "The IEA-District Heating and Cooling Project". The technical development in our country, on this and adjacent fields, depend on research cooperation on such international projects. And besides - the network of professional colleagues you learn to know by the cooperation is invaluable.

SINTEF Energy, March 1996

Rolf Ulseth

Rolf Ulseth

SUMMARY

As part of Annex IV of the International Energy Agency's District Heating and Cooling Project (IEA-DH&CP), a project called **E**fficient **S**ubstations and **I**nstallations (**ESI**) has been performed.

The main objective of the project was to develop more efficient consumer heating systems in commercial buildings, where heating energy is supplied by district heating (DH). The need for more efficient systems has increased in recent years, as low temperature DH is considered to be favourable in a future perspective. The project strategy was to undertake a systematic, theoretical study of the design of consumer heating systems, based on thermodynamic analysis. Then some basic system configurations were chosen which make the best compromises between theoretical goals and practical limitations.

To document the performance of the chosen systems, a simulation tool was needed. This was done with an extended and improved version of the simulation program called **CHESS** (**C**onsumer **H**eating **S**ystem **S**imulation) which was formerly developed in Annex III of the IEA-DH&CP. The extended and improved version of CHESS is called **CHESS-ESI**.

The theoretical studies lead to the conclusion that the common system, where service water is heated in two steps, has the potential to give the maximum cooling of the DH-water. This basic concept for service water heating was therefore chosen as a part of the new systems.

Three alternative principal system configurations for space heating were evaluated in CHESS-ESI:

System 1 Ventilation heating coil and radiator system connected in parallel on the secondary side of the heat exchanger (Used as "Reference system" in the documentation of the performance of the new systems since this system is common today)

System 2 Ventilation heating coil connected in series with the radiator system on the secondary side (See Figure 1)

System 3 Ventilation heating coil connected in series with the radiator heat exchanger on the primary side

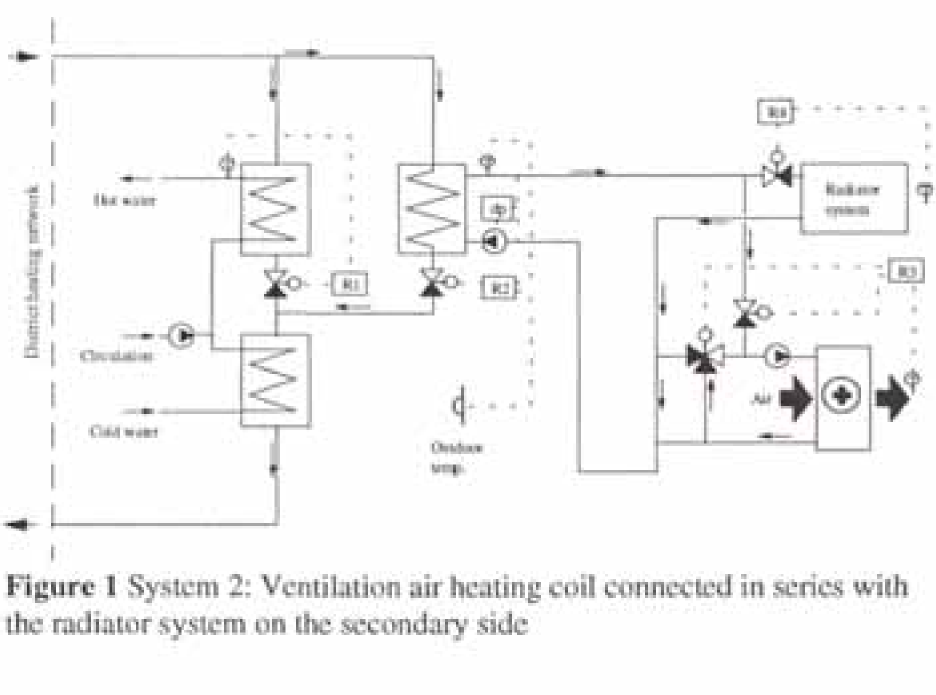


Figure 1 System 2: Ventilation air heating coil connected in series with the radiator system on the secondary side

From the theoretical studies it was deduced that there should be an optimal secondary supply temperature which would give the lowest primary outlet temperature from the space heating system's heat exchanger. From simulations, the optimal secondary supply temperatures could be found for the actual conditions, as demonstrated for System 2 in Figure 2. The overall conclusion from these simulations is that every individual space heating system in practice has its own optimal "heating curve" which normally is a nonlinear function of the outside temperature.

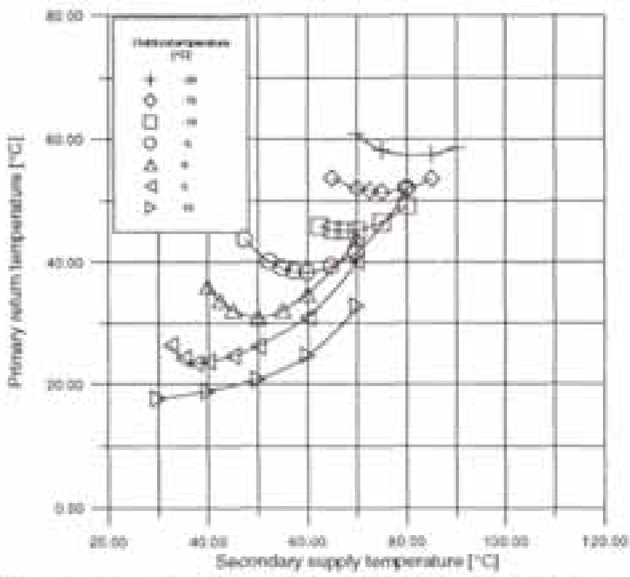


Figure 2 Primary return temperature from space heating heat exchanger as function of secondary supply temperature for System 2

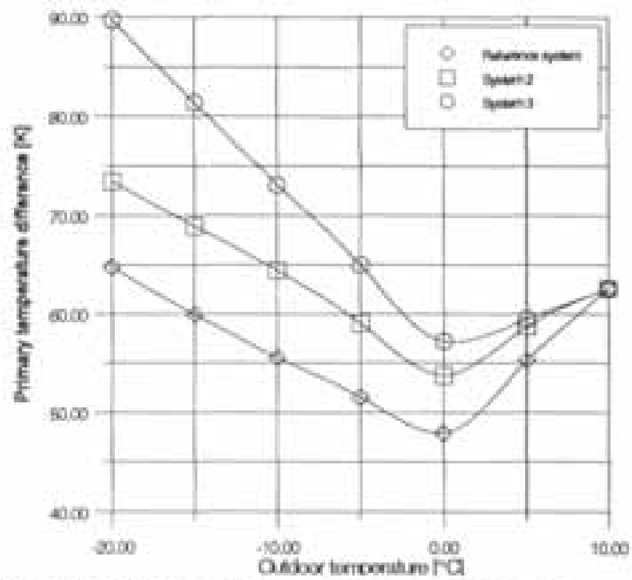


Figure 3 Primary temperature difference for space heating system with optimized heating curves, high temperature DH-system

The simulations showed that for Systems 2 and 3 a small modification of the ventilating heating coil design could significantly increase the cooling of the primary water. Figure 3 shows the cooling of the primary water across the space heating heat exchanger by optimised heating curves and a modified ventilating heating coil design for the three principal systems in CHESS-ESI.

In these simulations we have a conventional high temperature DH-system with 120°C design temperature and 80°C primary supply temperature in summer.

Modern DH-systems will normally be designed for lower primary supply temperatures compared with old systems. Figure 4 shows the results for similar conditions as in Figure 3 but now with a low temperature DH-system with a constant supply temperature of 70°C throughout the year.

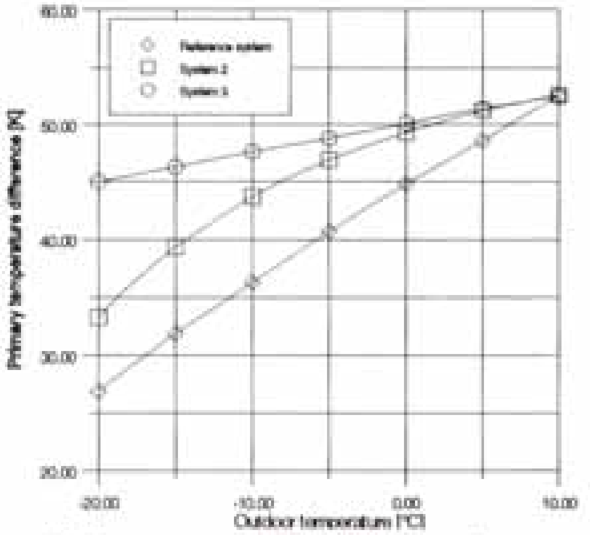


Figure 4 Primary temperature difference for space heating system with optimized heating curves, low temperature DH-system

In practice, the cooling of the DH-water in consumer installations is a very important factor for the total economy of DH. It will, for instance, increase the capacity of the expensive DH-pipeline system, and reduce the cost of pumping the hot water. The total cooling of the DH-water across the consumer's installation depends to a great extent on the amount and nature of the service hot water consumption in the actual building. The CHESS-ESI simulations were carried out for typical conditions for an office building and a hospital building for the three system configurations.

For the same conditions as in Figure 3 and a two-step system for the service water heating the results for System 2 and 3 respectively show about 11% and 18% increase in the "annual volumetric mean temperature difference" for the office building compared to the traditional "Reference system". The equivalent values for the hospital building for the two systems were about 8% and 13% respectively.

For the cases above, a decrease in the "design primary flow" for System 2 and 3 respectively were found to be about 8% and 17% for the office building and about 4% and 9% for the hospital building compared to the reference system.

Simulations including service hot water were also carried out for a low temperature DH-system. The results of the simulations for the actual systems above, and a constant primary supply temperature of 70°C, show an increase in the annual volumetric mean temperature difference of about 12% and 16% for the office building and about 8% and 10% for the hospital building, compared with the reference system.

For the low temperature case, the decrease in the design primary flow for System 1 and 2 is more significant than for the high temperature case. The results show a decrease in the design primary flow for System 1 and 2 respectively of about 9% and 21% for the office building and about 13% and 27% for the hospital building, compared with the reference system.

The simulations in this project are done with a two-step system for service water heating. The CHESS-ESI package may also simulate a one-step system for service water heating. This is achieved by setting the area of the preheat-exchanger to zero (see Figure 1). The space heating system can also be simulated with radiator system only. This is achieved by closing the heating coil control valves.

TABLE OF CONTENTS

INTRODUCTION TO THE JOINT REPORT

The main objective of the present work has been to develop more efficient consumer heating systems in buildings where the heating energy is supplied by hot water district heating (DH). The need for more efficient systems has increased in the latest years due to fact that low temperature DH is considered to be favourable in a future perspective.

The cooling of the district heating water delivered to a building is directly affecting the capacity of the DH network. and is one of the most important factors to reduce the total cost of DH. The work in this project has therefore, to a great extent, been focusing on that problem.

In the project **E**fficient **S**ubstation and **I**nstallations **(ESI)** the strategy has been to do a systematic, theoretical study of the design of the consumer heating system based on thermodynamical grounds. From there some basic system configurations were chosen which presumably make the best compromises between theoretical and practical goals.

To document the performance of the chosen systems a simulation tool was needed. For this purpose it was planned to use an extended and improved version of the simulation program called **C**onsumer **HE**ating **S**ystem **S**imulation (**CHESS**) which was developed and reported in the former Annex III of the IEA - District Heating and Cooling Project.

It was considered that there was a special need for validation of the heating coil model in the CHESS program, and it was decided to do some work on that topic.

From the start it was decided that the ESI-project should be performed as a joint project between SINTEF, LTH and a work group with close connection to the University of Saskatchewan.

On this background and for technical reasons it was found appropriate to make the joint report in the following tree parts:

Part I: **Performance Analyses of Efficient Substations and Installations.**

Part II: **Discussion of Low Temperature Substations - motives, state-of-the-art and some key issues for progress.**

Part III: **Validation of the Heating Coil Model used in the CHESS program**

The extended and improved version of the CHESS is described in part I of the joint report, and the new simulation program is called **CHESS-ESI**.

In the appendix to the joint report you will find a brief introduction to the use of CHESS-ESI.

A diskette with the executive programmes in the CHESS-ESI package may be requested from the operating agent for the IEA - District Heating and Cooling Project, Annex IV:

NOVEM Sittard, The Netherlands.

Part I

PERFORMANCE ANALYSIS OF EFFICIENT SUBSTATIONS AND INSTALLATIONS

Rune Volla Rolf Ulseth Jacob Stang

SINTEF
The Foundation for Scientific and Industrial Research
at The Norwegian Institute of Technology

March 1996

Table of Contents

NOMENCLATURE

A	Area	[m²]
A_c	Heat transfer area on hot side of heat exchanger	[m²]
A_h	Heat transfer area on hot side of heat exchanger	[m²]
c	Heat loss gradient	[°C^{-1}]
c_{pc}	Specific heat capacity of cold side fluid	[kJ/kgK]
c_{ph}	Specific heat capacity of hot side fluid	[kJ/kgK]
c_{pw}	Specific heat capacity of wall material	[kJ/kgK]
C	Scaling factor in heat transfer correlations	[-]
C_{room}	Total heat capacity of room	[kJ/K]
D_i	Inner diameter of tube/duct	[m]
DOT	Design outdoor temperature	[°C]
F	Heat loss fraction	[-]
h	Convective heat transfer coefficient	[W/m²K]
h_c	Convective heat transfer coefficient on cold side of heat exchanger	[W/m²K]
h_h	Convective heat transfer coefficient on hot side of heat exchanger	[W/m²K]
K	Constant factor	[-]
L_{tube}	Length of tube	[m]
LMTD	Logarithmic mean temperature difference	[K]
n	Section number	[-]
Q	Energy	[J]
$\dot{Q}_0$	Design heat loss	[W]
$\dot{Q}_{corr}$	Correction of heat loss with deviation in room temp. compared to design value	[W]
$\dot{Q}_{net\ loss}$	Net heat loss from building	[W]
$\dot{Q}_{rad}$	Heating power from radiator	[W]
$\dot{Q}_{vent}$	Heating power from ventilation air heating coil	[W]
t	Time	[s]
Δt	Simulation time step	[s]
T	Temperature	[°C]
T_{ai}	Inlet ventilation air temperature	[°C]
T_c	Temperature in fluid on cold side of heat exchanger	[°C]
T_h	Temperature in fluid on hot side of heat exchanger	[°C]
T_{out}	Outdoor temperature	[°C]
T_{room}	Room temperature	[°C]
ΔT	Temperature difference	[K]
ΔT_c	Temperature difference between hot side fluid and wall	[K]
ΔT_h	Temperature difference between hot side fluid and wall	[K]
ΔT_m	Volumetric mean temperature difference	[K]
U	Overall heat transfer coefficient	[W/m²K]
V	Volume	[m³]
V_c	Volume on cold side of heat exchanger	[m³]

V_h	Volume on hot side of heat exchanger	$[m^3]$
V_w	Volume of wall material	$[m^3]$
$\dot{V}$	Volume flow	$[m^3/s]$
$\dot{V}_c$	Volume flow on cold side of heat exchanger	$[m^3/s]$
$\dot{V}_h$	Volume flow on hot side of heat exchanger	$[m^3/s]$
η	Efficiency [0-1]	[-]
λ	Thermal conductivity	[W/mK]
λ_c	Thermal conductivity of fluid on cold side of heat exchanger	[W/mK]
λ_h	Thermal conductivity of fluid on hot side of heat exchanger	[W/mK]
ν	Kinematic viscosity	$[m^2/s]$
ν_w	Kinematic viscosity close to wall	$[m^2/s]$
ξ	Drag coefficient	[-]
ρ	Density	$[kg/m^3]$
ρ_c	Density of fluid on cold side of heat exchanger	$[kg/m^3]$
ρ_h	Density of fluid on hot side of heat exchanger	$[kg/m^3]$
ρ_w	Density of wall material	$[kg/m^3]$

Dimensionless numbers:

Nu_d	Nusselt number based on diameter
Pr	Prandtl number
Pr_w	Prandtl number at wall temperature
Re_d	Reynolds number based on diameter
Re_{Dh}	Reynolds number based on hydraulic diameter

1. Introduction

In the search for more efficient consumer heating systems for district heating, a tool to simulate the operation of different system configurations is useful to evaluate the performance. As mentioned previously, the CHESS simulation concept was developed in a former project under the IEA- District Heating and Cooling Project (Hjorthol and Ulseth 1992). To perform the new simulations in the IEA-Efficient Substations and Installation project, a further development of the CHESS concept was needed.

The new simulating tool that is presented and used in the present project is called CHESS-ESI. With this dynamic simulation tool we are able to simulate the complete heating system within a building connected to a district heating network on a "standard" PC of today.

2. CHESS-ESI - A computer tool for analysing district heating substations and heating installations

As previously noted, the CHESS consept originated in a former IEA-District Heating and Cooling project reported in Annex III. In this project the concept has been further developed into the simulation tool, CHESS-ESI. The purpose of this development has been the need to simulate the operation of new consumer heating system configurations to test their performance.

The former version of the CHESS program was limited to representing the space heating system with one fixed system configuration. This was partly due to settings in the source code of the CYPROS equation solver that limited the number of component models, and partly due to the speed of personal computers at the time.

The CHESS-ESI system models include both space heating and hot water preparation. Hence, the number of component models is increased. To do so the source code has been modified. Additionally, the parameter text has been adjusted as an attempt to improve the user interface.

An evaluation of the CHESS program showed that further development of the component models was required. In CHESS-ESI some models are therefore further developed and some are adjusted compared with CHESS. The details of these refinements are found in Chapter 3. The heating coil model has been specially evaluated in the work by Johnson and Besant (1995), which is found in Part 3 of this report. Their suggestions for improvements have been taken into account in the heating coil model developed for CHESS-ESI.

CHESS-ESI consists of three principal system simulation models. The first is a reference system and two have been developed as attempts to improve the performance of the consumer heating system. The systems are presented in detail in Chapter 4, and the results of case studies to test the systems are presented in Chapter 5.

3. Mathematical models - Refinements of CHESS-ESI compared with CHESS

CHESS-ESI is, as previously noted, a further development of the CHESS program. The mathematical description of the CHESS program is described in detail by Hjorthol and Ulseth (1992). This chapter gives an introduction to the model refinements done in CHESS-ESI. A more thorough treatment can be found in Volla (1996).

3.1 Models used unchanged as described in the CHESS report

The following models are used unchanged in CHESS-ESI and are described in the CHESS report:

- Radiator model
- Temperature sensor model
- Two-way valve model
- Controller model

3.2 General changes in the dynamic heat transfer models

The energy equations for a section, n, of the heat transfer models can generally be described by Figure 3.1 and Equations 3.1 to 3.3. The following assumptions are then introduced:

- All heat conduction between the sections is neglected
- The temperature gradient in the tube and plate walls is neglected
- No heat loss to the surroundings

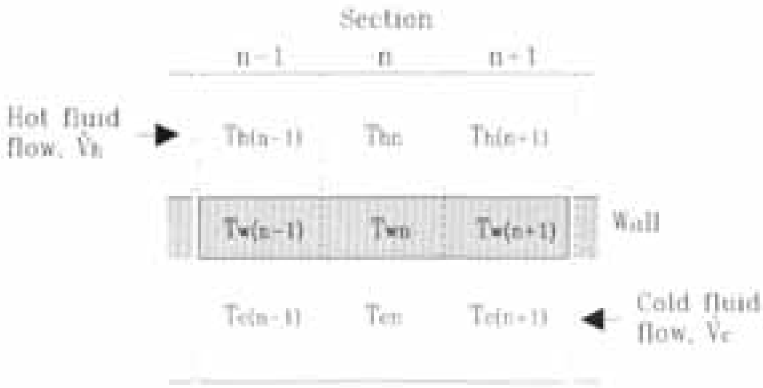


Figure 3.1 Segment of lumped heat exchanger model

Hot fluid element:

$$\frac{\delta}{\delta t}(\rho_h \cdot c_{ph} \cdot V_{h,n} \cdot T_{h,n}) = \rho_h \cdot c_{ph} \cdot \dot{V}_h \cdot (T_{h,(n-1)} - T_{h,n}) - h_h \cdot A_{h,n} \cdot \Delta T_{h,n} \tag{3.1}$$

Wall element:

$$\frac{\delta}{\delta t}(\rho_w \cdot c_{pw} \cdot V_{w,n} \cdot T_{w,n}) = h_h \cdot A_{h,n} \cdot \Delta T_{h,n} - h_c \cdot A_{c,n} \cdot \Delta T_{c,n} \tag{3.2}$$

Cold fluid element:

$$\frac{\delta}{\delta t}(\rho_c \cdot c_{pc} \cdot V_{c,n} \cdot T_{c,n}) = \rho_h \cdot c_{pc} \cdot \dot{V}_h \cdot (T_{c,(n-1)} - T_{c,n}) + h_c \cdot A_{c,n} \cdot \Delta T_{c,n} \tag{3.3}$$

The temperature difference, ΔT, between adjacent elements in a section can be described in various ways. Steiner (1989) has given three possibilities for the temperature difference in a heat exchanger model without wall mass representation:

1) Temperature difference represented by the difference between the outlet temperatures of the two elements, called "the direct temperature difference" (DTD).

2) Temperature difference represented by the arithmetic mean temperature difference (AMTD) between the inlet and outlet temperatures of the elements.

3) Temperature difference represented by the logarithmic mean temperature difference (LMTD) between the inlet and outlet temperatures of the two elements.

For the models presented in this work where the wall mass is included, the equivalent temperature difference representation will be:

1) $$\Delta T_{h,n} = T_{h,n} - T_{w,n} \qquad (3.4)$$

2) $$\Delta T_{h,n} = \frac{T_{h,n-1} + T_{h,n}}{2} - T_{w,n} \qquad (3.5)$$

3) $$\Delta T_{h,n} = \frac{(T_{h,n-1} - T_{w,n}) - (T_{h,n} - T_{w,n})}{\ln\frac{(T_{h,n-1} - T_{w,n})}{(T_{h,n} - T_{w,n})}} \qquad (3.6)$$

Steiner shows that using the DTD causes an underestimation of the temperature difference when a small number of sections are used. This steady state error decreases when using AMTD. Using LMTD is equivalent to the analytical reference model.

In CHESS the temperature difference, ΔT, is described by DTD. The static error introduced by this simplification can be eliminated by parameter estimation from measurements like for instance done by Jonsson (1990). Such measurements are, however, generally not available to the users of this simulation tool. To improve the temperature difference representation, AMTD has therefore been introduced in the heat exchanger models of CHESS-ESI. For the same reason the number of sections has been increased.

3.3 Tube and duct

To include starting effects in short tubes the following heat transfer correlations are used to describe the heat transfer on the inside of tubes:

Turbulent flow
Gnielinski (1976) found the following correlation to be valid for both liquids and gases in the range $2300 > Re_d > 10^6$ and $0.6 > Pr > 10^5$:

$$Nu_d = \frac{(\xi/8)\cdot(Re_d - 1000)\cdot Pr}{1 + 12.7\cdot\sqrt{(\xi/8)}\cdot(Pr^{2/3} - 1)} \cdot \left[1 + \left(\frac{D_i}{L_{tube}}\right)^{2/3}\right] \cdot K \qquad (3.7)$$

Where the factor K for liquids is: $K = (Pr/Pr_w)^{0.11}$

Laminar flow

Seider and Tate (1936) presented the following empirical correlation that is generally preferred (Chapman 1987). The correlation is valid in the range $0.48<Pr<16700$ and $(d_i/L)Re_d Pr>10$.

$$Nu_d = 1.86 \cdot \left[\left(\frac{D_i}{L_{tube}} \right) \cdot Re_d \cdot Pr \right]^{1/3} \cdot \left(\frac{\nu}{\nu_w} \right)^{0.14} \quad \textbf{(3.8)}$$

The range $(d_i/L)Re_d Pr>10$ gives a criterion to determine the tube length where the starting length effects should be included.

3.4 Plate heat exchanger

The plate heat exchanger model is basically similar to the one in the CHESS program. The changes are described below.

Heat transfer coefficients

The heat transfer between fluid and plate is given by:

$$Nu_{D_h} = C \cdot Re_{D_h}^{\,n} \cdot Pr^{\,m} \quad \textbf{(3.9)}$$

A review of literature concerning the estimation of the parameters by Volla (1996) shows that for plate heat exchangers with the so-called "herringbone pattern" which is used for district heating applications the following exponents are close to the ones identified in most of the studies:

m=0.4

n=0.62

Given the exponents n and m, the scaling factor can be calculated using given temperatures and flows for the specific heat exchanger. It is assumed that the two sides of the heat exchanger are similar and therefore follow the same heat transfer correlation. The scaling factor C can be calculated through the following procedure.

The ratio between the two heat transfer coefficients can be written:

$$K=\frac{h_h}{h_c}=\frac{\lambda_h}{\lambda_c}\cdot\left(\frac{Re_{D_h,h}}{Re_{D_h,c}}\right)^n\cdot\left(\frac{Pr_h}{Pr_c}\right)^m \tag{3.10}$$

Rearranging the energy balance at steady state conditions and substituting the overall heat transfer coefficient U with the ratio K gives:

$$h_h=(1+K)\cdot\frac{\dot{V}_h\cdot\rho_h\cdot c_{ph}\cdot(T_{hi}-T_{ho})}{A_h\cdot LMTD} \tag{3.11}$$

The scaling factor for both sides, C, is given by:

$$C=\frac{h_h D_h}{\lambda_h}\cdot Re_{D_h,h}^{-n}\cdot Pr_h^{-m} \tag{3.12}$$

If measurements that cover the total area of operation are available, the C that gives the best fit should be used.

Users of a system simulation tool generally do not have measurements for the specific heat exchangers. In this case the constant C can be determined from the design conditions that are normally given by the heat exchanger supplier. The litterature review by Volla (1996) shows that C typically is found to be in the range 0.3-0.6.

Optional use of the heat exchanger in the system models.
In CHESS-ESI a possibility for the user to disable the heat exchanger models is included. This is done by setting the height or width of the plates to zero. In this state the temperature on both sides remains unchanged through the heat exchanger without time delay.

This option is specially useful when examining the hot water preparation system, because decreasing the preheater in a two-step scheme to zero gives a parallel scheme.

3.5 Heating Coil Model

The heating coil model of CHESS-ESI is based on the heating coil model in CHESS. However some major changes have been done mostly as a result of the evaluation of the model by Johnson and Besant (1995). A report, which presents all details of this evaluation, is found in Part 3 of this report. The following presents the new features of the heating coil model.

Geometry
The geometry of an air heating coil is complex compared with most components. Nevertheless, having accurate calculations of heat transfer areas and heat storage volumes is important. CHESS had only simplified relations to calculate the geometry. In CHESS-ESI the detailed geometrical relations presented by Johnson and Besant (1995) are included.

Heat transfer correlations
The heat transfer coefficient for the water side of the heating coil is calculated with the same correlations as described for tubes.

To calculate the air side heat transfer coefficient, Johnson and Besant (1995) suggest the use of empirical correlations. However, Volla (1996) has found a wide range of different air side geometries are used in modern heating coils. Since the existing empirical relations only cover a small part of this variation, an empirical calculation of the air side heat transfer coefficients will include undesirable uncertainties.

By assuming constant air flow through the heating coil this problem can be avoided. Constant air flow gives constant heat transfer conditions that can be calculated from measurements or from design data using a similar method as described for the plate heat exchanger.

Since so called "Constant Air Volume Systems (CAV)" are by far the most common for ventilation the assumption of constant air flow is generally valid.

The heat transfer coefficient on the air side is calculated based on design data and a given heat transfer coefficient on the water side from the following procedure:

The UA-product at design conditions is given:

$$UA_0 = \frac{\rho_a \cdot c_{pa} \cdot (T_{ao} - T_{ai})}{LMTD_0} \tag{3.13}$$

The constant air side heat resistance including pipe wall and collar resistance is then given as the difference. Thus:

$$(hA)_a = \left(\frac{1}{UA_0} - \frac{1}{h_{w0}} \right)^{-1} \tag{3.14}$$

Air volume heat storage
In CHESS-ESI the dynamics of the heating coil model have been simplified by neglecting the heat capacity of the air volume. Calculations of the geometry of the heating coil show that the air heat

capacity is in the order of magnitude 0.1% of the total heat capacity of the coil. The same simplification is done in the heating coil model by Børresen and Thunem (1984). The dynamic behaviour of this model has been examined by Rikheim (1987). His measurements show that neglecting the heat capacity of the air has very little effect on the dynamic behaviour of the heating coil.

3.6 Room model

Since the main purpose of CHESS-ESI is to analyse substations and consumer heating installations, a simple model of the building is chosen.

Assumptions and simplifications
The room is modelled with the following assumptions:

- The room mass, i.e. walls, inventory and room air, is assumed to act as one heat storage of uniform temperature.
- The net heat loss from the room can be expressed as a linear function of the temperature difference between the indoor and outdoor temperature.

The first assumption implies that the temperature difference between air and walls is neglected. Parametric studies done by Grétarsson et al (1990) showed that a 5°C temperature difference between the walls and the ambient air gave a 1.3% decrease in heat output from the radiator heat surface.

Model description
The net heat loss is given as a linear function of the outdoor temperature determined by two points as shown in Figure 3.2. Constant gains are represented by shifting the heat loss curve downwards, while solar gain, which increases with increasing outdoor temperature, is represented by the gradient of the curve.

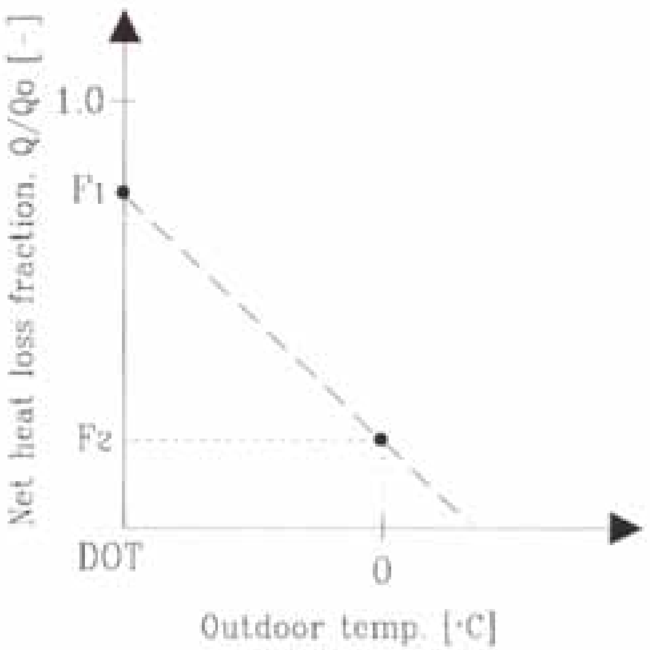


Figure 3.2 Net heat loss as a function of outdoor temperature with T_{room}=20°C (DOT- Design Outdoor Temperature)

The net heat loss can be written:

$$\dot{Q}_{net\ loss}=(F_2-cT_{out})\cdot\dot{Q}_0+\dot{Q}_{corr} \tag{3.15}$$

where the gradient c is given by the two points:

$$c=\frac{F_1-F_2}{DOT-0} \tag{3.16}$$

and the heat loss correction due to room temperature different from 20°C:

$$\dot{Q}_{corr}=c\cdot(20-T_{room})\cdot\dot{Q}_0 \tag{3.17}$$

Taking the total heat capacity of the room into account, the heat balance for the room becomes:

$$C_{room} \cdot \frac{\delta T_{room}}{\delta t} = \dot{Q}_{rad} - \dot{Q}_{net\ loss} \qquad \textbf{(3.18)}$$

3.7 Actuator with hysteresis

The actuator model in CHESS-ESI is principally the same as in CHESS. However an adjustment is done in the actuator algorithm to avoid shifts in the actuator position when the controller output is stable. For details see Volla (1996).

3.8 Three- way mixing valve in serial connection

General description

Two of the system models in CHESS-ESI have serial connection from the return water flow from the radiator system to the ventilation air heating coil.

The idea of this connection is to utilize the remaining temperature level of the radiator return water to heat the ventilation air. Usually there exists a potential for this since the room temperature, which is the limit of the radiator return temperature, is higher than the inlet air to the heating coil. The principle of serial connection of radiator and heating coil system has previously been used in a similar connection presented by Nilsen (1994a).

Figure 3.3 shows the secondary side of a system with a detailed scheme of serial/parallel connection. Since this connection introduces flow variations that are not possible to describe with the two-way valve model, a separate model that describes the flow in the coupling is described.

The flow in the coupling is controlled by the controller R3, with the ventilation outlet air temperature as control parameter. This controls the two-way valve and the three-way mixing valve in the following sequence:

1) The water temperature from the three-way mixing valve is changed by changing the flow ratio between the control and shunt port. This mode of operation continues until the control port is fully open. In this mode the serial coupling is fully utilized.

2) When the combination of radiator return temperature and radiator flow is no longer sufficient to heat the ventilation air to meet the reference temperature, the two-way valve starts to open. The opening of the two-way valve will gradually lead the operation from serial to parallel connection of radiator system and heating coil.

The sequential control leads to three basic modes of operation:

1) Serial connection of radiator system and heating coil, where all water to heating coil is supplied by the three-way valve.

2) Mixed serial and parallel connection, where the flow from the three-way valve is gradually displaced by the flow from the two-way valve.

3) Parallel connection of radiator system and heating coil, where the total water flow to the heating coil is supplied by the two-way valve.

Assumptions and simplifications

The following assumptions and simplifications are introduced to model the flow through the coupling as a function of valve openings:

- **In operation mode 1, flow 7 in the heating coil circuit, is assumed to be determined by the circulation pump and to be constant.**
  The circulation pump draws a constant water flow (flow 5) through the three-way valve. The ratio between control port and shunt port flow varies as a function of the valve position.

- **The pressure in front of the shunt port and control port is assumed to be equal.**
  This will be the case if the length of tubes between the two ports is very short.

- **In mode 2, flow 5 from the three-way valve is assumed to be gradually displaced by flow 6, while flow 7 through the heating coil is constant.**
  The difference in pressure between the inlet of the three-way and two-way valve lead to a displacement of flow 5 by flow 6. The rate of this displacement depends on the rate change in flow 6 determined by the two-way valve.

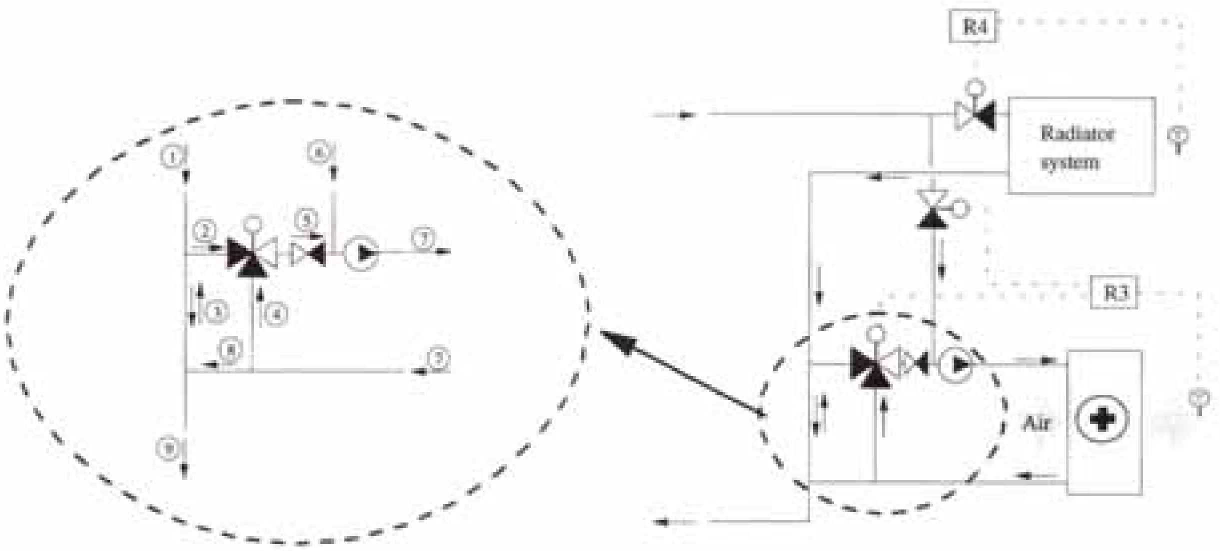


Figure 3.3 Three-way valve coupling for serial/parallel connection of radiator system and ventilation air heating coil.

Model description

Mode 1 and 2 operation:

The circulation flow through the heating coil is determined by the pump and given as input. The flow through the tree-way valve is then given as:

$$\dot{V}_5=\dot{V}_7-\dot{V}_6 \quad \textbf{(3.19)}$$

Since the pressure on the inlets of the three-way valve is equal, the distribution of the flow is given by the valve opening:

$$\dot{V}_2=f(z)_{cp}\cdot\dot{V}_5 \quad \textbf{(3.20)}$$

Flow 3 will function either as bypass of superfluous radiator flow (positive) or as a complementary flow to flow 2 (negative):

$$\dot{V}_3=\dot{V}_1-\dot{V}_2 \quad \textbf{(3.21)}$$

The flow through the bypass port of the three-way valve is given by:

$$\dot{V}_4=\dot{V}_5-\dot{V}_2 \quad \textbf{(3.22)}$$

and the return flow from the heating coil circuit, 8:

$$\dot{V}_8=\dot{V}_7-\dot{V}_4 \quad \textbf{(3.23)}$$

The total secondary return flow from both radiator system and heating coil, 9, is:

$$\dot{V}_9=\dot{V}_8+\dot{V}_3 \quad \textbf{(3.24)}$$

Mode 3 operation:

In mode 3 the flow through the two-way valve, 6, has displaced the flow 5, and the coupling works as a flow-controlled parallel connection. A further increase in flow 6 results in a corresponding increase in flow 7, 8 and 9. In this mode the total radiator return flow bypasses the heating coil circuit. The system flows are:

$$\dot{V}_8 = \dot{V}_7 = \dot{V}_6 \tag{3.25}$$

$$\dot{V}_3 = \dot{V}_1 \tag{3.26}$$

$$\dot{V}_9 = \dot{V}_1 + \dot{V}_6 \tag{3.27}$$

This model of the flow in the serial/parallel connection of radiator system and heating coil fulfils its purpose by giving the desired transition between the serial coupled and parallel coupled system. It is therefore suitable for demonstrating the potential to increase the performance of the system by additional cooling of the radiator water. However, the simplified approach to describe the changing pressure conditions in mixed mode operation will result in a deviation between calculated and observed valve position.

3.9 Thermodynamic properties

CHESS-ESI uses temperature variable thermodynamic properties for the fluids, and constant properties for solids. Details are found in Volla (1996).

3.10 Outdoor temperature excitation model

The outdoor temperature input can be varied in two modes:

1) Step increase, where the outdoor temperature can be increased in steps that are set by the user.

2) Sinusoidal variation, where the outdoor temperature is varied as a sine function with period and amplitude set by the user. By setting the period to 24 hours the sine can be used to simulate the outdoor temperature variation of a day.

3.11 Service hot water flow excitation model

The service hot water flow can be varied in two modes:

1) Step increase, where the flow can be increased in steps that are set by the user.

2) File input, where input data on a specified file on TS-format is read by the program. The program TS-HEAD.EXE is used to convert data from ASCII-format to TS-format.

4. Simulated consumer heating systems

CHESS-ESI consists of three system models. One is the Reference system, which is similar to the systems used in modern consumer heating systems connected to district heating. The two others, called System 2 and System 3 attempt to improve the performance of the consumer systems according to the objective of this project.

In all three connections, the instantaneous hot water preparation system is initially connected as an ordinary two-step scheme with preheating of the service hot water. This is in accordance with the recommendations of Frederiksen (1995). If wanted, however, the models can also be simulated as parallel schemes by setting the height or width of the preheater to zero.

The configuration of the two latter systems has been chosen according to the recommendations of Frederiksen (1995), which can be found as Part 2 of this report, and the discussions in the Experts Group which has monitored the project. Figures 4.1 to 4.3 show the three systems. The major difference between the new systems and the reference system is the serial connection between the heating coil and the radiator system. The idea is to utilize the low temperature potential of the inlet ventilation air to cool the radiator return water additionally.

The same idea was introduced by Nilsen (1994 a), who has shown an example of a serial connection of the radiator system and heating coil. In a paper presented by Ahonen et al. (1995) an improved consumer heating system that utilizes this principle is also discussed. Although few details of the system and the analyses of this system are described, the results show that the system gives improved performance.

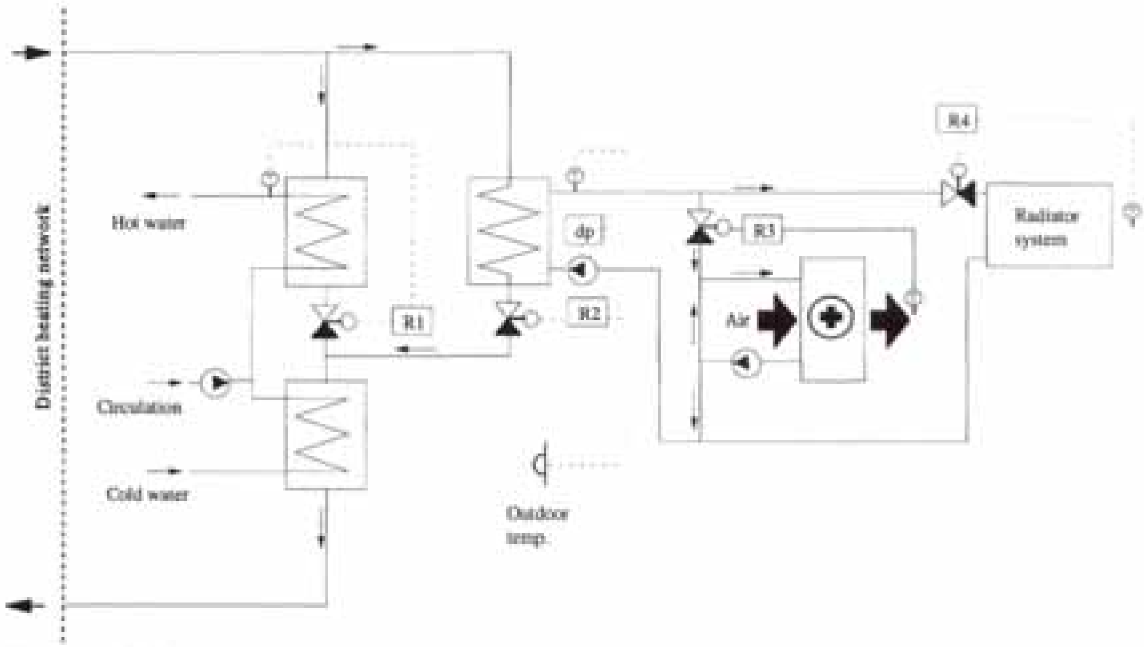


Figure 4.1 System 1, Reference system. Ventilation heating coil and radiator system connected in parallel on the secondary side.

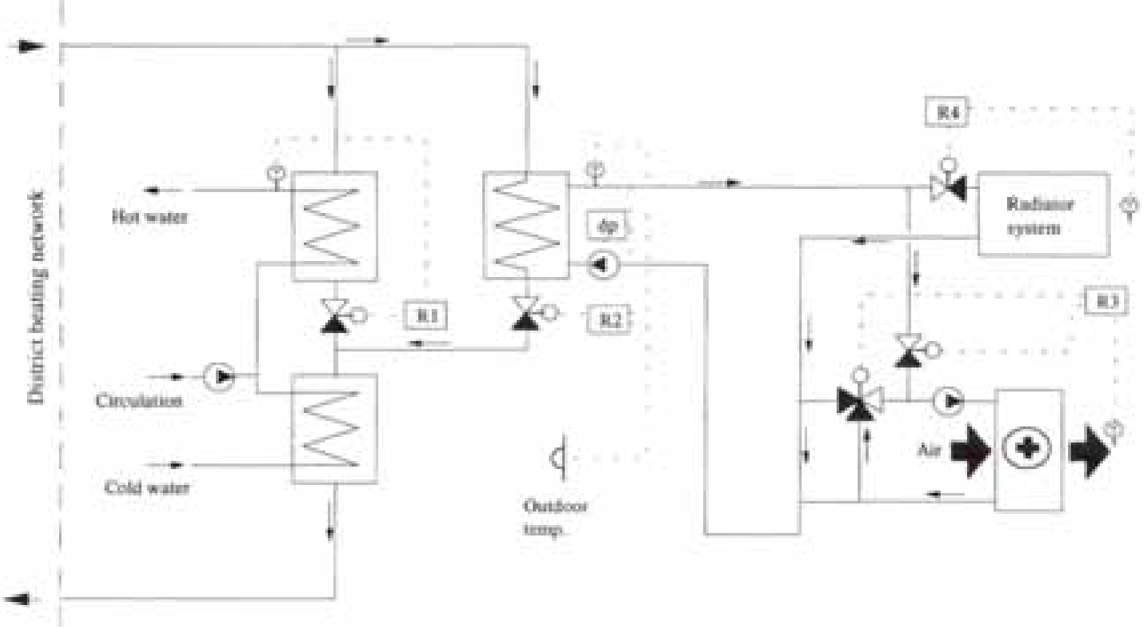


Figure 4.2 System 2. Ventilation heating coil and radiator system connected in series on the secondary side.

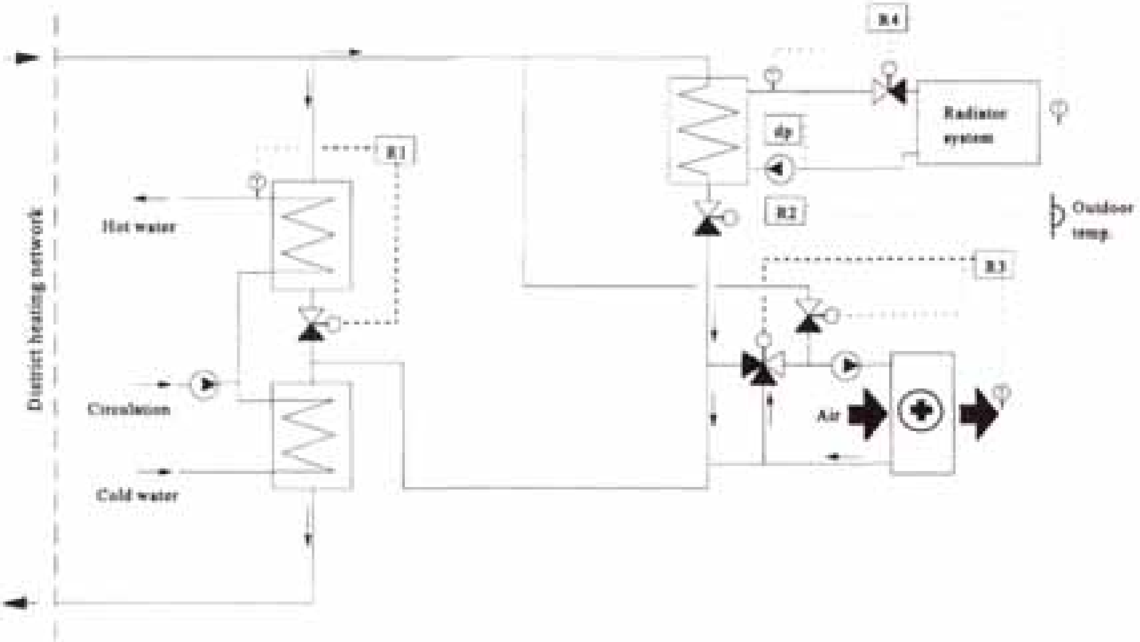


Figure 4.3 System 3. Ventilation heating coil and radiator heat exchanger connected in series on the primary side.

The Reference system is, as noted, a conventional type of heating system commonly used today. On the secondary side the radiator circuits and the ventilation air heating coil are connected in parallel. The heat load from the radiators is controlled with radiator control valves with the room temperature as parameter, while the heating coil is controlled with a special type of recirculation circuit with the outlet air temperature of the heating coil as parameter. The recirculation circuit involves mixing of water of different temperature levels that is thermodynamically unfavourable. Even so, this coupling is generally preferred by HVAC-designers, due to the risk of freezing on the water side of the coil in a flow controlled system.

In System 2, the ventilation air heating coil is connected in series with the outlet of the radiator system. Water to the heating coil is drawn from the radiator return pipe with a circulation pump. The temperature of the water into the heating coil is controlled by mixing the radiator outlet water with recirculated water in a three-way valve. If the outlet flow from the radiator system is larger than the flow needed in the heating coil, the excess water is bypassed the heating coil. If the temperature or the flow of radiator outlet water is insufficient to cover the demand of the heating coil, the parallel connection is opened to supply higher temperature. In this case the parallel connection will gradually replace the serial connection until all the water is supplied in parallel with the radiator circuit.

In System 3, the ventilation air heating coil is connected on the primary side to the outlet of the radiator heat exchanger. The connection of the heating coil is otherwise similar to System 2. This direct connection of the heating coil is possible in cases where the air handling unit is situated close to the district heating substation. The pressure in the district heating network is generally higher than in the secondary system. Commercial heating coils are, however, designed for a pressure of 16 bar, which is equal to or higher than the design pressure of most district heating networks in Scandinavia.

The additional investment costs introduced by changing from the Reference system to Systems 2 and 3 are evaluated to be small compared with the total costs of the consumer heating system. If the same components can be used unchanged, the resulting extra costs will be marginal. As the results will show, the heating coil will for some cases need to be redesigned. This can result in an increase in the number of tube rows. According to Nilsen (1994 b) the resulting increase in investment costs is about 1%.

5. Simulations of selected systems

In this chapter the performance of the three systems is evaluated through three case studies denoted Case A, B and C. The three cases are briefly described as:

- Case A. High temperature system - reference design conditions
- Case B. High temperature system - adapted ventilation heating coil design
- Case C. Low temperature system - adapted ventilation heating coil design

In the following sections the evaluation criteria are defined and the chosen building parameters are described.

5.1 Evaluation criteria

The different system scheme combinations are evaluated against the following criteria:

- Primary temperature difference for space heating system over the total range of operation.
- Yearly average performance for total consumer installation including hot water preparation scheme.
- Design primary volume flow for total consumer installation system.

The primary temperature difference for the space heating system is analysed as a qualitative evaluation of the improvements achieved by the new space heating system schemes. The systems can be compared for each point of operation.

For a total quantitative evaluation, the yearly average performance of the total consumer system is estimated by calculating the volumetric mean temperature difference, with the volume flow as weighting factor. In this way the temperature difference in periods with high heat loads counts more than periods with a low heat load. The volumetric mean temperature difference is commonly used for this purpose (See e.g. Winberg and Werner 1987, Gummerus 1989).

The volumetric mean temperature difference is expressed by the following equation:

$$\Delta\bar{T}_w = \frac{\int_t (\dot{V}\cdot\Delta T)}{\int_t \dot{V}} \tag{5.1}$$

If density and specific heat capacity are assumed to be constant, the volumetric mean temperature difference can also be estimated from the recordings of the heat meter in the district heating substation:

$$\Delta\bar{T}_w = \frac{\int_t (\dot{V}\cdot\Delta T)}{\int_t \dot{V}} = \frac{\rho\cdot c_p\cdot\int_t (\dot{V}\cdot\Delta T)}{\rho\cdot c_p\cdot\int_t \dot{V}} = \frac{Q}{\rho\cdot c_p\cdot V} \tag{5.2}$$

where V is the water volume and Q is the energy consumption of a period.

Besides the total average performance of the system, the operation at design conditions is interesting. The volume flow at design conditions determines the size of tubes and valves on the primary side, and must therefore be included in the total comparison of system schemes. Increased capacity of the district heating network is one of the most valuable improvements in a district heating system, due to the high investment costs in the network (Volla 1994).

5.2 Chosen building parameters

The following constant parameters have been chosen to represent modern occupational buildings located in the northern parts of the world:

Design outdoor temperature:	DOT = -20°C
Reference indoor temperature:	T_{room} = 20°C
Heated area:	A = 5000 m²
Design heat load to radiator:	$\dot{Q}_{rad}$ = 162 kW

Ventilation air flow:	$\dot{V} = 10\ m^3/m^2\ h$
Heat recovery efficiency:	$\eta = 0.65$
Inlet air temperature :	$T_{ai} = 17.0°C$
Design load heating coil:	$\dot{Q}_{vent} = 195\ kW$

Usually the ventilation air flow varies somewhat between different building categories.
The effect of differences in service hot water consumption is evaluated by varying between office building with low consumption and hospital building with high consumption.
To limit the number of variations the given parameters are chosen to represent both hospital and office buildings.

Radiator heat load variation

The heat load that must be supplied by the radiator system is given as the sum of heat losses minus the the part covered by internal and solar gains. Due to increasing solar gain with increasing outdoor temperature, the radiator heat load will cease before the temperature difference between indoor and outdoor temperature is zero. The outdoor temperature where the radiator heat load ceases, is called the balance temperature. This temperature depends on the building standard and design. In the case studies the net heat loss is simplified in the following way:

- The net heat loss from the building decreases linearly as a function of the outdoor temperature from design load at design outdoor temperature to zero load at outdoor temperature 10°C.

Ventilation heat load variation

The heat load to the ventilation air heating coil is a function of the outdoor temperature. It is assumed that the ventilation system is provided with a heat recovery unit. The inlet temperature to the heating coil is calculated from the recuperator efficiency:

$$T_{ai} = T_{out} + \eta \cdot (T_{room} - T_{out}) \tag{5.3}$$

Figure 5.1 shows the resulting heat loads to radiator system and ventilation air heating coil.

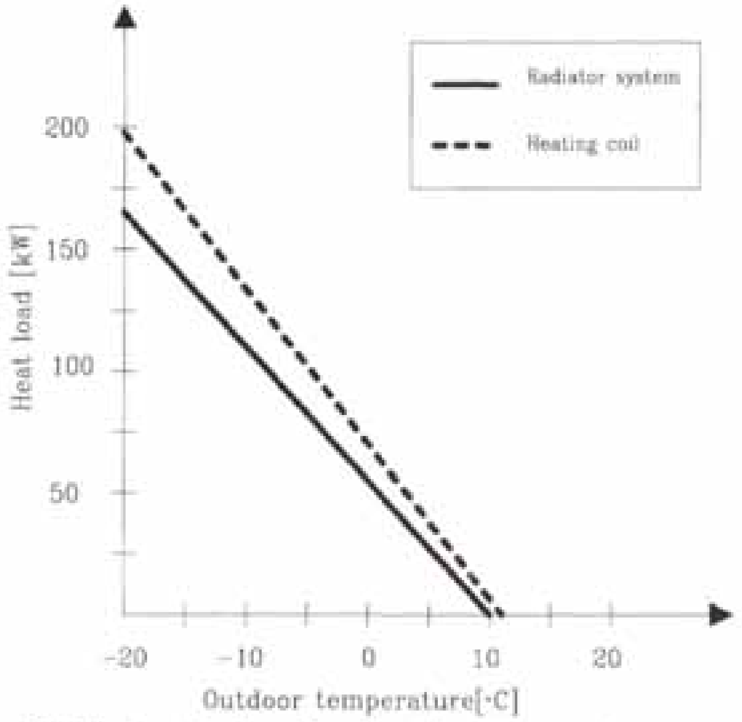


Figure 5.1 Heat load as function of outdoor temperature

The outdoor temperature is assumed to vary according to the climate of Trondheim, Norway. For more details see Volla (1996).

Hot water flow variation

The hot water flow variation model is based on measurements presented in Volla (1996). An evaluation of these measurements shows that a simple hot water flow variation model is sufficient to evaluate the effect of the service hot water on the average performance of the total consumer heating system. The variation in hot water flow is reduced to two points of operation:

1) No hot water flow.

2) Average hot water flow (zero flows excluded)

The time fraction of the respective situations is used to calculate an average weighted performance. This simplification includes the total hot water consumption, but evens out the differences between the flow magnitudes.

The design flow is calculated according to the Scandinavian standard formula, which was originally developed by Rydberg (1946).

Table 5.1 sum up the characteristic data for the 5000 m² building used in the system simulations.

Table 5.1 Characteristic hot water flow data in the 5000 m² building used in the case studies

	Design flow [l/s]	Circulation flow [l/s]	Average flow (zero flow excluded) [l/s]	Time fraction with flow [%]
Office building	0.60	0.06	0.073	6.5
Hospital building	1.5	0.15	0.16	44

5.3 Simulated cases

Case A. High Temperature System - Reference Design Conditions
The design temperatures are chosen according to the conventional district heating systems in Scandinavia.

The system configuration is altered from the Reference system to the new Systems 2 and 3 without changing the design of any of the components.

The space heating system is designed for DOT with the following temperatures for the respective components:

Space heating heat exchanger:

	Inlet [°C]	Outlet [°C]
Primary	120	57.5
Secondary	52.5	80.0

Radiator system:

	Inlet [°C]		Outlet [°C]
Water	80.0		60.0
Room		20.0	

Ventilation air heating coil:

	Inlet [°C]	Outlet [°C]
Water	80.0	40.0
Air	6.0	17.0

The least temperature difference (LTD) of 5°C for the heat exchanger is chosen according to the Swedish demands that is also widely used in the rest of Scandinavia.

The service hot water heat exchangers is designed for summer conditions, with the following temperatures:

	Inlet [°C]	Outlet [°C]
Primary	80.0	25.0
Secondary	5.0	55.0

At design conditions the preheater and supplementary heater operates as one heat exchanger area. The preheater and supplementary heater are given equal size according to general practice.

Primary supply temperature in the high temperature cases of this study is chosen according to Figure 5.2. The design supply temperatures for winter conditions, 120°C, and summer conditions, 80°C, are similar the ones used in Scandinavian high temperature systems. The gradient of the heating curve, which is chosen to reach summer conditions at an outdoor temperature of 0°C, depends on the consumer in the district heating system with the highest temperature demand.

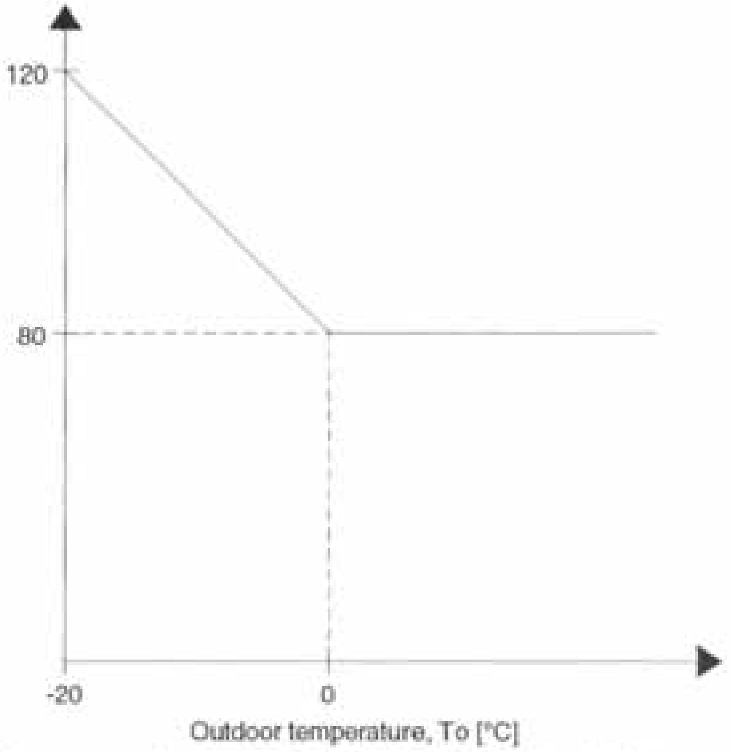


Figure 5.2 Primary supply temperature for the conventional high temperature system

The secondary supply temperature is optimized to give maximum primary temperature difference across the space heating heat exchanger.

Case B. High Temperature System - Adapted Ventilation Heating Coil Design

In Case B the ventilation air heating coil is redesigned to a lower temperature demand to increase the utilization of the serial connection between radiator and heating coil. Otherwise the systems are similar to the ones in Case A.

Design temperatures for the heating coil is chosen to:

	Inlet [°C]	Outlet [°C]
Water	60.0	30.0
Air	6.0	17.0

Lowering the design water temperatures from 80-40 to 60-30 will sometimes, depending on the type of heating coil, result in an increase from one to two tube rows in the heating coil. According to Nilsen (1994 b) the resulting increase in investment costs is about 1%.

Case C. Low Temperature system - Adapted Ventilation Heating Coil Design

This case examines the potential of the serial connections for a low temperature system.
Both primary and secondary systems are designed for low temperature. The radiator system is designed according the Swedish standard, which has been a legal demand since 1982. The design temperatures of the ventilation air heating coil are adapted to the serial connections of System 2 and System 3.

With low temperature secondary system lowering the temperatures on the primary side is possible. In a recent Dutch study (Vrins and Versteeg 1995) where many aspects of heat production and distribution are taken into account, the optimum primary supply temperature for a low temperature system is evaluated to be 70°C.

The space heating system is designed for DOT with the following temperatures for the respective components:

Space heating heat exchanger:

	Inlet [°C]	Outlet [°C]
Primary	70.0	43.2
Secondary	38.2	60.0

The somewhat odd design temperatures for the heat exchanger, is due to the resulting return temperature of the heating coil at 60°C secondary supply temperature.

Radiator system:

	Inlet [°C]		Outlet [°C]
Water	60.0		45.0
Room		20.0	

Ventilation air heating coil:

	Inlet [°C]	Outlet [°C]
Water	45.0	25.0
Air	6.0	17.0

Compared with a more conventional low temperature design, with water temperatures similar to the radiator system, the size of the heating coil will increase from one to two tube rows.

The service hot water heat exchangers is designed for summer conditions, with the following temperatures:

	Inlet [°C]	Outlet [°C]
Primary	70.0	25.0
Secondary	5.0	55.0

The primary supply temperature is constant 70°C at all outdoor temperatures. This temperature is close to minimum due to service hot water preparation. The secondary supply temperature is optimized to give maximum primary temperature difference over the space heating system.

5.4 Simulation Results

5.4.1 Case A. High temperature System - Reference Design Conditions

Optimal secondary supply temperature

The optimal supply temperature for the three systems was found by simulating the operation of the heating system over a range of secondary supply temperatures. Figures 5.3 to 5.5 show the resulting primary return temperatures from the space heating system.

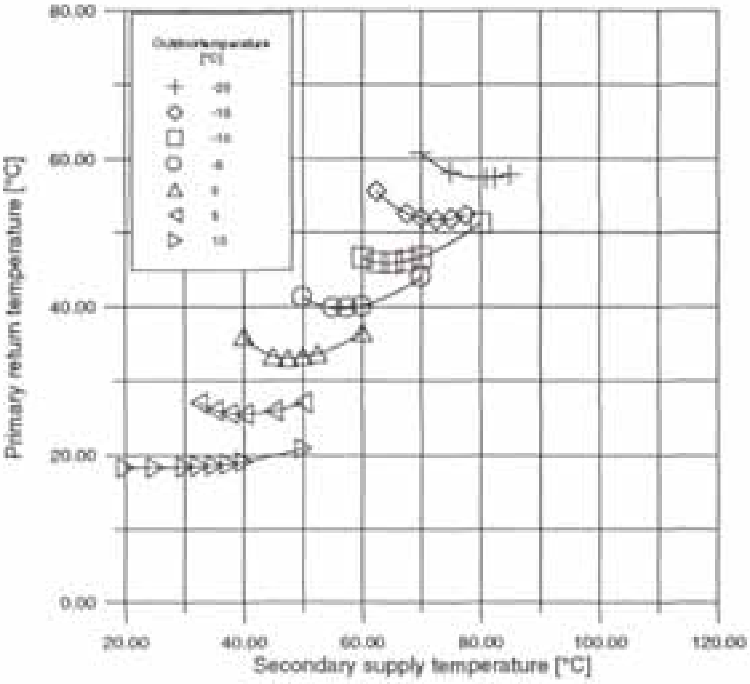


Figure 5.3 Primary return temperature from space heating heat exchanger as function of secondary supply temperature for the Reference system

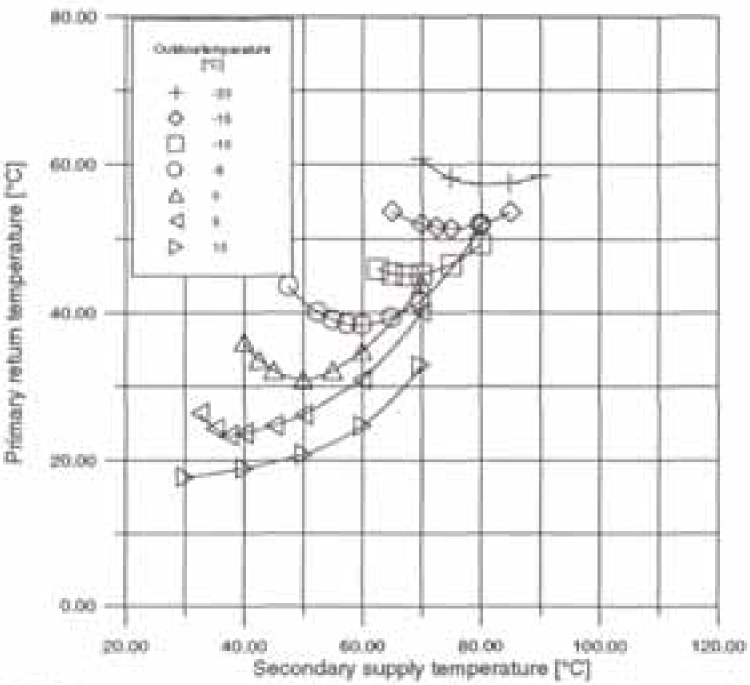


Figure 5.4 Primary return temperature from space heating heat exchanger as function of secondary supply temperature for System 2

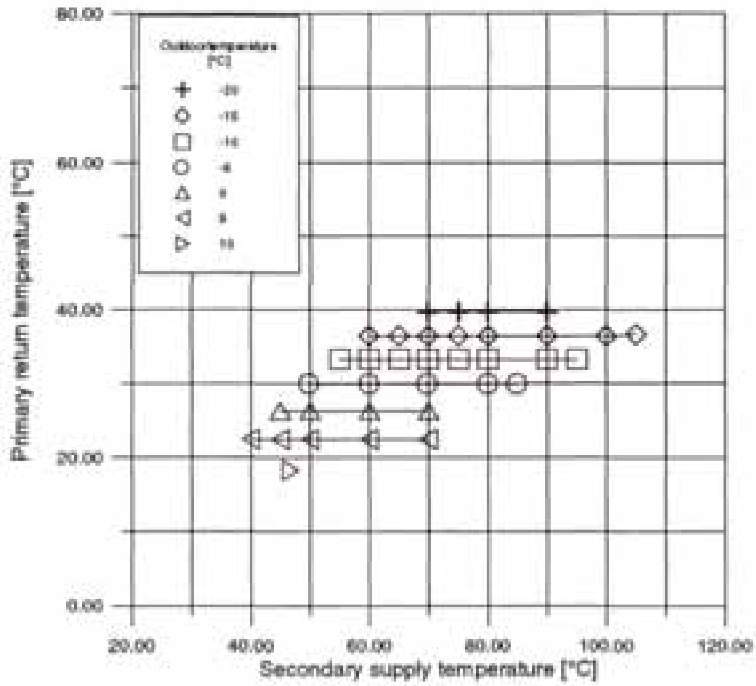


Figure 5.5 Primary return temperature from space heating heat exchanger as function of secondary supply temperature for System 3

The optimal heating curves for the Reference system and System 2 are shown in Figure 5.6. For System 3, the primary return temperature is given by the return temperature of the heating coil as long as no part of the radiator flow is bypassed the heating coil. When water is bypassed, the primary return temperature increases. As shown in Figure 5.5 this situation will not occur for realistic secondary supply temperatures.

<u>Space heating system performance</u>

Figure 5.7 shows a comparison between the primary temperature difference of the space heating system with optimized heating curves.

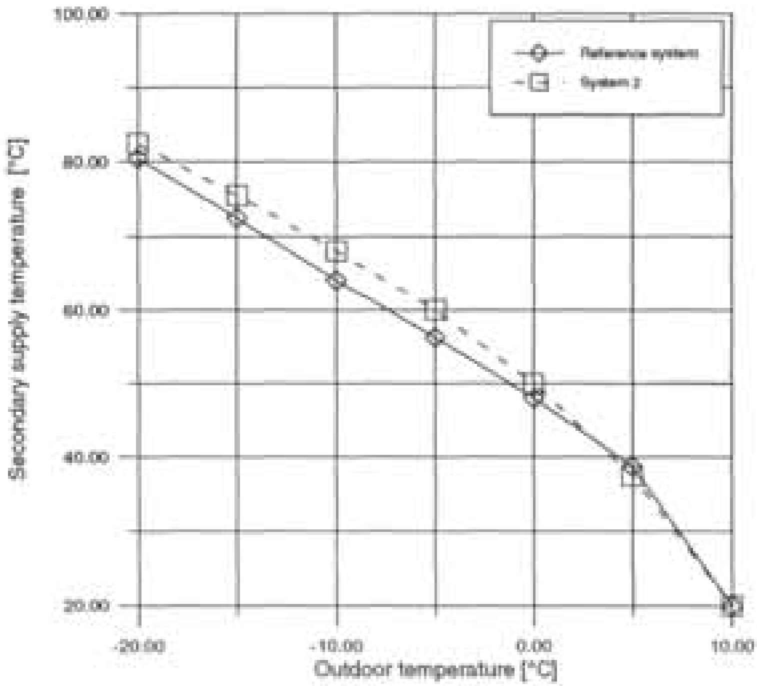


Figure 5.6 Optimized heating curves, Case A, high temperature system

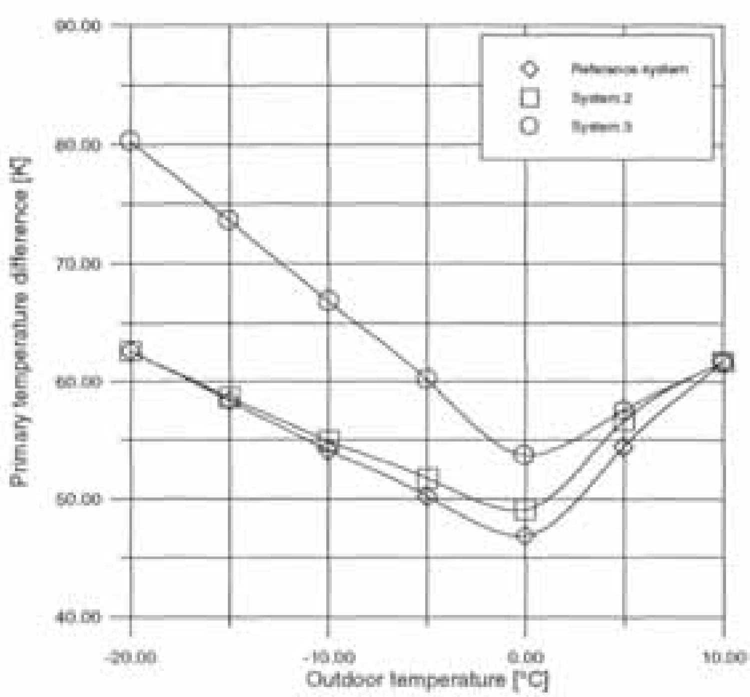


Figure 5.7 Primary temperature difference for space heating system, Case A, high temperature system

Yearly average performance of total system

To make an evaluation of the yearly performance of the three systems, the two-step service hot water system is included. To examine the influence of variations in the hot water consumptions between building categories both hospital and office building are evaluated.

Tables 5.2 and 5.3 show the resulting volumetric mean temperature differences.

Table 5.2 Yearly volumetric mean temperature difference, Case A

Location	Reference system	System 2	System 3
Office building	50.1	51.9	56.8
Hospital building	52.2	53.6	57.5

Table 5.3 Relative improvement in volumetric mean temperature difference [%] compared with the Reference system, Case A

Location	System 2	System 3
Office building	3.6	13.4
Hospital building	2.8	10.2

Design primary flow for substation

The primary side of the substation and the close parts of the district heating network are normally designed to give sufficient primary volume flow at design outdoor temperature and design hot water heat load with design supply temperature and design pressure difference. The design volume flow is the sum of primary water through the space heating system and the service hot water preparation system.

As noted, the primary supply temperature curve is, in theory, placed to keep the primary volume flow constant. In these simulations, as in reality, the primary temperature curve is given as input to the district heating substation. The maximum primary flow is therefore found either at design outdoor temperature or at the outdoor temperature from which the primary supply temperature is kept constant, in this case 0°C.

Figures 5.8 and 5.9 show the resulting primary flows for Case A at the two critical outdoor temperatures. For the hospital building the maximum primary volume flow appears at outdoor temperature 0°C. If this situation occurred at a time when the capacity was fully utilized, it would result in an increase in primary supply temperature from the district heating supplier.

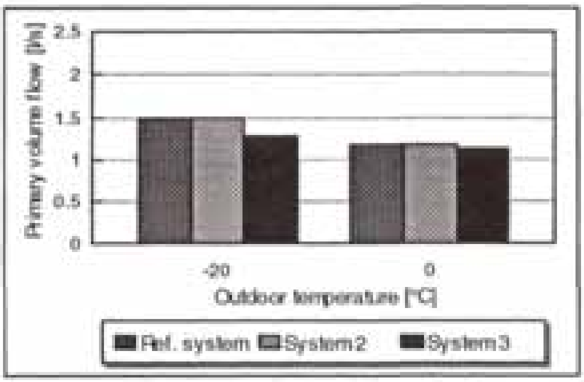


Figure 5.8 Primary volume flow for the office building at the critical outdoor temperatures, Case A

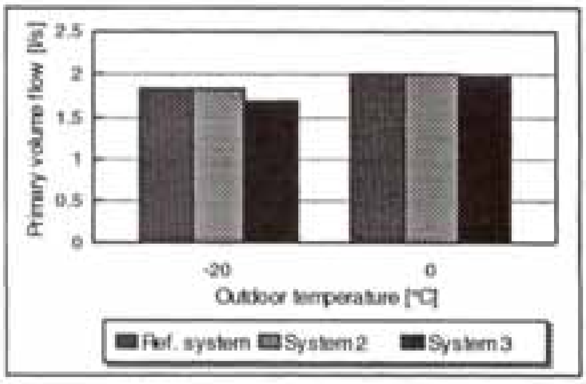


Figure 5.9 Primary volume flow for the hospital building at the critical outdoor temperatures, Case A

5.4.2 Case B. High Temperature System - Adapted Ventilation Heating Coil Design

In Case B the three systems are simulated with a new heating coil design. Otherwise the systems are equal to Case A.

<u>Optimal secondary supply temperature</u>

Figure 5.10 shows the optimized heating curves for the Reference system and System 2.
The corresponding curves that show the optimal secondary supply temperature as a function of primary return temperature are similar to the ones presented for Case A. They are therefore omitted from the text.

<u>Space heating system performance</u>

Figure 5.11 shows the primary temperature difference over the space heating system after the heating coil is redesigned to 60/30 instead of 80/40.

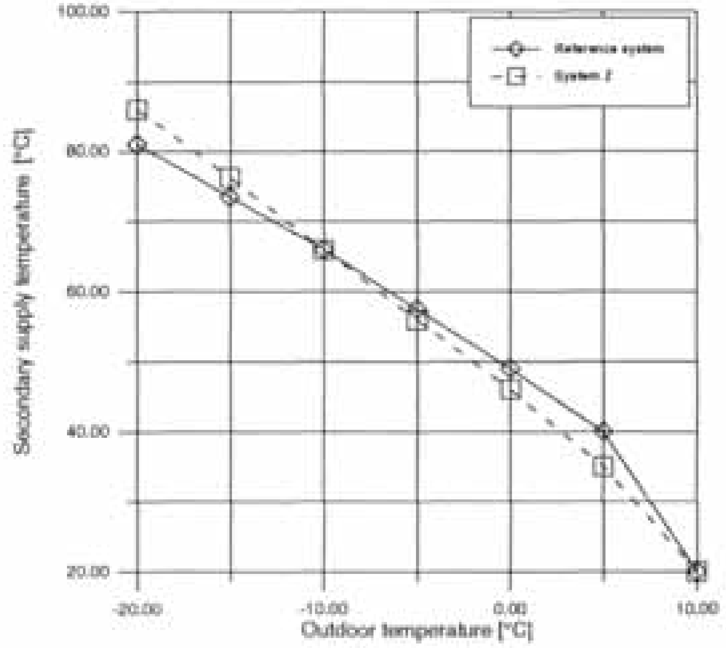


Figure 5.10 Optimized heating curves, Case B, high temperature system with new heating coil design

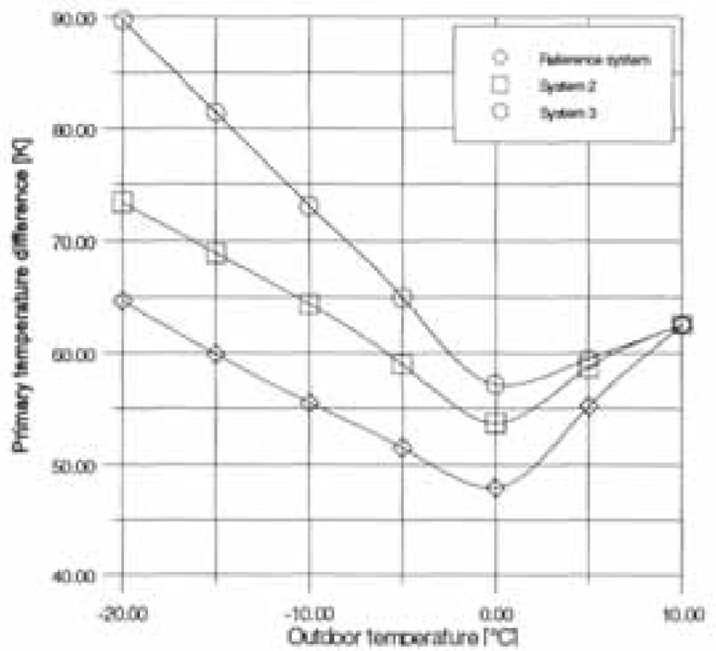


Figure 5.11 Primary temperature difference for space heating system, Case B, high temperature system with new heating coil design

Yearly average performance for total system

The hot water preparation system is included to evaluate the yearly mean performance of the systems. Tables 5.4 and 5.5 show the resulting primary volumetric mean temperature difference for the systems.

Table 5.4 Yearly volumetric mean temperature difference, Case B

Sub	Reference system	System 2	System 3
Office building	51,1	56.6	60.1
Hospital building	53	57.4	60.1

Table 5.5 Relative improvement in volumetric mean temperature difference [%] compared with the Reference system, Case B

	System 2	System 3
Office building	10.8	17.8
Hospital building	8.3	13.4

Design primary flow for substation

Figures 5.12 and 5.13 show the primary volume flow at critical outdoor temperatures for the office and hospital building respectively.

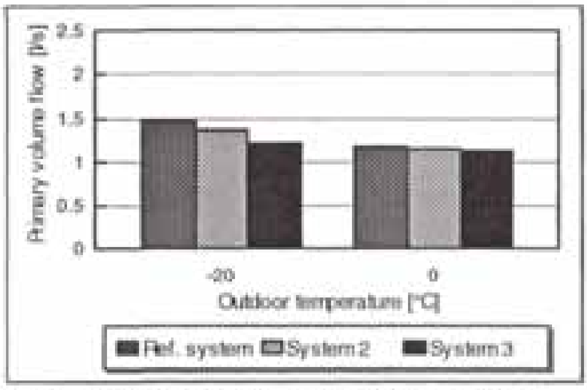


Figure 5.12 Primary volume flow for the office building at critical outdoor temperatures, Case B

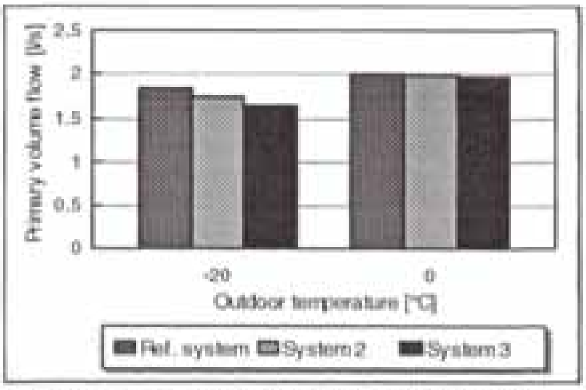


Figure 5.13 Primary volume flow for the hospital building at critical outdoor temperatures, Case B

Like in Case A, the maximum primary volume flow for the hospital building is found at outdoor temperature 0°C. This shows that the primary supply temperature curve is somewhat low to cover the total heat load at 0°C if the system was designed for outdoor temperature conditions.

5.4.3 Case C. Low Temperature System - Adapted Ventilation Heating Coil Design

In simulation Case C a low temperature system is simulated. To be comparable, all three systems are designed with the same components. This means that the Reference system is designed with lower heating coil temperatures than are normal in a conventional low temperature system.

Optimal secondary supply temperature

Figure 5.14 shows the optimal heating curve for the Reference system and System 2 in Case C.

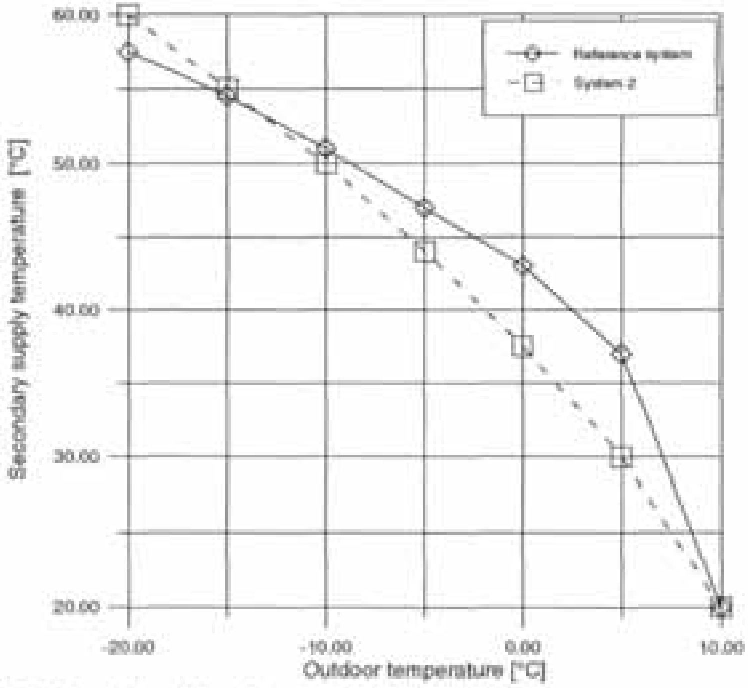


Figure 5.14 Optimized heating curves, Case C, low temperature system

Space heating system performance

Figure 5.15 shows the resulting primary temperature difference of the heating system with optimal heating curves.

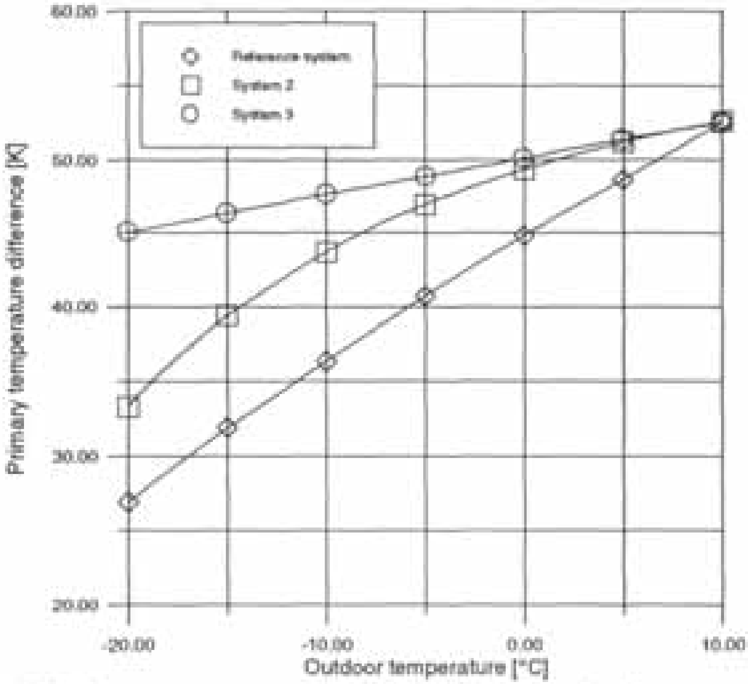


Figure 5.15 Primary temperature difference for space heating system, Case C, low temperature system

Yearly average performance for total system

The yearly average performance is evaluated in the same way as for the cases above. Tables 5.6 and 5.7 give the yearly volumetric mean temperature difference for the low temperature system.

Table 5.6 Yearly volumetric mean temperature difference, Case C

	Reference system	System 2	System 3
Office building	42.8	47.4	48.6
Hospital building	43.1	46.6	47.6

Table 5.7 Relative improvement in volumetric mean temperature difference [%] compared with the Reference system, Case C

	System 2	System 3
Office building	12.0	15.7
Hospital building	8.1	10.4

Design primary flow for substation

Because the primary supply temperature is constant in Case C, the maximum volume flow will appear at design outdoor temperature. Figures 5.16 and 5.17 show the design primary volume flow for the office and the hospital building.

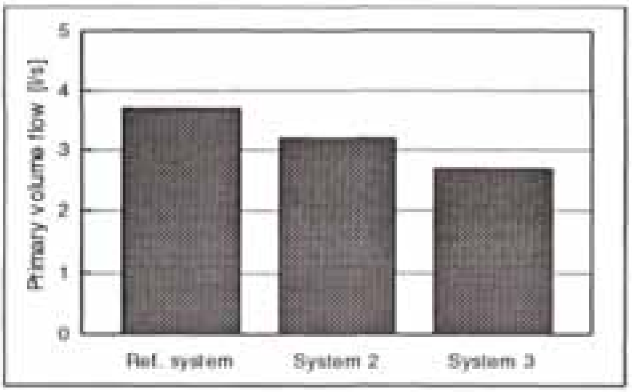


Figure 5.16 Primary volume flow for the office building at design outdoor temperature, Case C

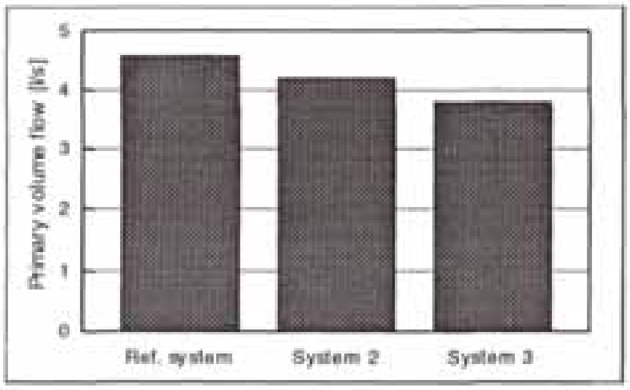


Figure 5.17 Primary volume flow for the hospital building at design outdoor temp. Case C

5.5 Result analysis

The following analyses are structured according to the following aspects:

- Optimization of the heating curves.
- Primary temperature difference for space heating system over the total range of operation.
- Yearly average performance for total consumer installation including hot water preparation scheme.
- Design volume flow for total consumer installation system.

5.5.1 Optimal heating curves

Figures 5.3 and 5.4 in Case A show that for the Reference system and System 2, an optimal secondary supply temperature exists, which gives minimum primary return temperature. The optimal points of the reference system and System 2 arise as a result of somewhat different mechanisms. This results in a sharper optimal point for System 2, which makes the optimization procedure most critical for this system. The mechanisms that determine the optimal points are explained in the following:

For the reference system the secondary return temperature is determined by the mixing of return water from the radiator and heating coil systems. As previously explained, only one combination of secondary temperatures and flow give the minimum primary return temperature. Figure 5.3, shows that the optimal point of the reference system is relatively flat. This indicates that the heat exchanger is not very sensitive to small changes in the secondary temperatures.

The optimal secondary supply temperatures for System 2, which is shown in Figure 5.4, depend on the utilization of the serial connection. When the serial connection is fully utilized, the optimal point is reached at the minimum secondary supply temperature where the total radiator water flow is used in the heating coil. At this point a decrease in secondary supply temperature results in an increase in radiator flow, which will be bypassed the heating coil system to give a rapid increase in secondary return temperature. An increase in secondary supply temperature from the optimum point will change the temperature differences over the radiator and the heating coil, but not affect the secondary output temperature.

If the serial connection in System 2 is not fully utilized, the system is operating in a mixed mode between serial and parallel connection. This gives an optimum caused by similar conditions as for the reference system . The effect of a not fully utilized serial connection in System 2 can be seen for Case A in Figure 5.4. For the lowest outdoor temperatures, the return radiator temperature is lower than the demand of the heating coil. The system therefore changes to operate as a parallel connection. The resulting optimal temperature curves can be seen to be equally flat to the ones for the reference system . At higher outdoor temperatures, the match between radiator return temperature and heating coil supply temperature is better. This is seen to result in a sharper optima, as described above.

Figures 5.3 and 5.4 show that for the highest outdoor temperature simulated, i.e. 10°C, the minimum primary return temperature appears at minimum secondary supply temperature. At this point there is only ventilation heat load. Since the return temperature from the heating coil is constant, the primary return temperature will only be influenced by the secondary supply temperature.

At lower outdoor temperatures this effect will be decreasing according to the ratio between ventilation heat load and radiator heat load. The result is a dip in the optimal heating curves in Figure 5.6, when approaching the highest outdoor temperature. A similar effect can also be observed for Case B and C in Figures 5.10 and 5.14. This effect will generally depend on the ratio between ventilation and radiator heat loads as a function of the outdoor temperature. The chosen radiator and ventilation heat loads give a ratio close to one in most of the heating season. This is evaluated to be a realistic situation with the present demands of ventilation air volume and insulation standard. In cases where the difference between radiator and ventilation heat load is greater, the individual effect on the optimal heating curve of each heat load will be more distinct.

The results show that the optimal heating curve is generally not linear like the traditional setting. The flat optimum of the Reference system opens the possibility to linearize the curve without increasing the primary return temperature considerably. With the sharper optima of System 2, however, it is more important to use modern control equipment that can handle non-linear heating curves.

Figure 5.5 for System 3 shows that no optimal secondary supply temperature is found. This is due to the serial connection on the primary side, and is explained by the following:

> For this system, the secondary supply temperature supplies water only for the radiator. By changing the radiator supply temperature, the primary radiator return temperature can be minimized as for Systems 1 and 2. As long as the return flow from the radiator heat exchanger is fully utilized in the heating coil, however, the total primary return flow is returned with constant temperature from the heating coil circuit. For System 3, the radiator return temperature is sufficiently high to utilize the serial connection in all three cases.
>
> If the radiator supply temperature is increased beyond the optimum for the radiator heat exchanger, the primary return temperature from the radiator heat exchanger is increased to a point where water is bypassed the heating coil. At this point the primary return temperature will increase. In these cases, however, the radiator supply temperature is varied within realistic range without affecting the primary return temperature.

Since a wide range of secondary supply temperatures can be chosen for System 3, the heating curve should be set to ensure good working conditions for the valves in the system.

5.5.2 Primary temperature difference for the space heating system

The primary temperature difference for the space heating system for Case A, B and C are shown in Figures 5.7, 5.11 and 5.15 respectively.

<u>Case A. High temperature system - Reference design conditions</u>

In Case A, the components are designed for the conventional system. The system variation is created by connecting the same components in alternative ways. Figure 5.7, shows that both System 2 and System 3 have improved the performance compared with the Reference system. The gain is, as one could expect, highest for System 3, where the heating coil is connected in series with the radiator heat exchanger on the primary side. This indicates that the reference heating coil design is well suited to utilize the primary radiator return temperature. The total primary temperature difference is, however, limited by the return temperature of the heating coil. The difference between the return temperature of the heating coil and the inlet air temperature indicates that there is a further potential to decrease the design return temperature. Even with extra investment cost this may be profitable because the return temperature affects the primary temperature difference directly.

With System 2, a small improvement in primary temperature difference compared with the Reference system is found. As noted above, the serial connection is not fully utilized due to the temperature demand of the heating coil. Without redesigning the heating coil, the improvements that can be made by changing from the reference system to System 2 are therefore limited.

Case B. High temperature system - Modified heating coil design

Figure 5.11 shows that the new heating coil design used in Case B improves the Systems 2 and 3 considerably compared with the Reference system.

System 2 has in this case a better match between the return temperature of the radiator system and the temperature demand of the heating coil. This improves the utilization of the serial connection which gives the decrease in return temperature from the heating coil a direct influence on the resulting secondary return flow and temperature.

The improved temperature difference for System 3, is mainly caused by the decrease in heating coil design return temperature. As noted, the direct influence on the primary temperature difference will make a low return temperature from the heating coil specially favourable for this system.

For the Reference system, the decrease in design supply and return temperature for the heating coil has only a small effect on the resulting primary temperature difference, because the water flow from the heating coil is only a part of the total secondary water flow.

Case C. Low temperature system - Modified heating coil design

In Case C, the heating coil is designed to match the radiator return temperature for the serial connection. Figure 5.15 shows that compared with the Reference system a considerable improvement can be made with both System 2 and System 3.

In Case C the primary supply temperature is presumed to be constant 70°C. The supply temperature is usually set to a minimum, but the temperature demand of the service hot water limits the possibility for a further decrease. The resulting temperature difference for the space heating system therefore decreases with increasing outdoor temperature.

A major disadvantage with low temperature systems compared with high temperature systems is that the temperature difference potential is lower due to the low primary supply temperature. This results in high volume flows and larger pipeline dimensions. The results of Case C show a design temperature difference over the space heating system for the Reference system of 26.9K. With the same components and radiator design temperatures the design temperature difference increases 23.7% for System 2 and 67.3% for System 3. In buildings where the space heating load is dominant at design flow conditions, this could lead to the possibility of a potential decrease in network dimensions.

5.5.3 Yearly average performance

The yearly volumetric mean temperature difference is used to evaluate the yearly performance of the total consumer heating system and substation. The influence of the service hot water consumption is evaluated by analysing both an office building (low-use) and a hospital building (high-use).

Case A. High temperature system - Reference design conditions
The results in Table 5.3 indicate that the total performance of the consumer installation improves more for the office building than for the hospital building when the serial connection is introduced. This can be explained by the difference in hot water consumption. In the hospital building the service cold water flow provides a low temperature potential for a relatively large part of the time that is utilized in the two-step scheme. In the office building the hot water flow is small and has short duration. Thus, the improvements in the temperature difference for the space heating are more pronounced in the average performance of the total system for the office building.

The limited improvement for System 2 compared with the Reference system is caused by the high temperature demand of the reference design heating coil. System 3, on the other hand, shows an improvement that is high considering that no change of components has been done.

Case B. High temperature system - Modified heating coil design
In Case B the heating coil design is modified to match the radiator return temperature. The resulting improvements are clearly shown in Table 5.5. Compared with Case A, the performance of Systems 2 and 3 is considerably improved, while the Reference system is only slightly improved. This indicates that the relatively small extra cost, that the new heating coil design represents, serves two purposes that both increase the primary temperature difference:

1) By lowering the temperature demand, the utilization of the serial connection is increased in System 2.

2) The lowering of the design return temperature from the heating coil gives lower primary return temperatures for both System 2 and 3.

Case C. Low temperature system - Modified heating coil design
In the low temperature Case, C, the same effects as for Case B can be observed in Table 5.7. The serial connections in System 2 and 3 is utilized to give a good improvement in average temperature difference that is highest for the office building.

5.5.4 Design primary flow

Case A. High temperature system - Reference design conditions
Figures 5.8 and 5.9 show the primary flows in Case A for the critical outdoor temperatures for the office and hospital building. The maximum flows appear at design outdoor temperature for the office building and at 0°C for the hospital building. This difference indicates that the primary supply temperature curve is slightly higher than necessary for the heat loads of the office building.

There is practically no difference in design flow between the Reference system and System 2 in Case

A. The design flow of System 3, however, is reduced with 14.3% for the office building and 7.9% for the hospital building. The office building has most to gain in reduced design flow from the improved space heating system due to the small hot water consumption.

Case B. High temperature system - Modified heating coil design
The results from Case B, in Figures 5.12 and 5.13, shows design flow for the same temperatures as in Case A.
Comparing the three systems, it can be seen that the office building gives the largest reductions in design flow when using Systems 2 and 3, with a decrease of 8 and 17% respectively. A similar improvement, although only 4 and 9%, can be seen for the hospital building.

Case C. Low temperature system - Modified heating coil design
The design flow for the low temperature Case, C, is shown in Figures 5.16 and 5.17. In this case the primary supply temperature is constant.

The low temperature case gives, as expected, the greatest reductions in the design flow for System 2 and 3 compared with the Reference system. The design flow is reduced 9% for System 2 and 21% for System 3 compared with the Reference system. Corresponding results for hospital building are 13% and 27%.

Thus, by utilizing the low temperature of the inlet ventilation air, it is possible to reduce the design flow considerably without significant extra investment costs. A further increase in temperature difference for the well working Reference system, can only be achieved by increasing the radiator surface.

5.6 Conclusions

From the discussion of the case studies above it seems reasonable to draw some conclusions that are valid for these system configurations in general.

- In district heating substations with indirect connection, the heating curve, which sets the secondary supply temperature should be optimized to give minimum primary return temperature. If a serial connection is used on the secondary side, like in System 2, the optimization is specially profitable due to the sharp optima.

- The optimal heating curve is generally not linear. In conventional parallel systems, however, a linear heating curve can be used without any major increase in primary return temperature due to the relatively flat optima.

- Due to the difference in return temperatures from the radiator system and ventilation heating coil system, the optimal heating curve will depend on the ratio between the heat loads. This is specially the case when the recirculation coupling is used on the heating coil as in System 1, 2 and 3. Since the return temperature is constant from this connection, it will favour low secondary supply temperatures.

- Serial coupling of radiator system and heating coil has a potential to give a considerable increase in primary temperature difference for the space heating system. To maximize the gain from this connection the temperature levels of radiator and heating coil systems need to be harmonized.

- If the serial connection is to be used on the secondary side, as in System 2, a lower design supply temperature necessitates a larger heating coil to match the system. When the heating coil is connected in series on the primary side, as in System 3, the design supply temperature for the heating coil can be allowed higher.

- The improved return temperature from the heating coil coupling is more profitable for the serial coupled systems than for the conventional system in parallel. For the serial coupling on the primary side, the return temperature has direct influence on the primary return temperature. Since the marginal costs of decreasing the return temperature from the heating coil are small, the serial connection provides a cost efficient way to maximize the temperature difference.

- The annual average performance of the district heating substation can be considerably improved by introducing a serial connection between radiator and ventilation air heating system.

- Improvements on the space heating system are most profitable for buildings with low service hot water consumption. For buildings with high hot water consumption, the primary water can be cooled against the cold service water temperature.

- The design flow from the substation is decreased by using the serial connection. For low temperature systems with constant primary supply temperature, the potential decrease in design flow is considerable. For the simulated low temperature case the design flow was reduced in the range 20% to 30% compared with the Reference system.

- For most systems, the return temperature from the space heating system will be sufficiently high at design conditions to give a preheating of the service hot water that reduces the design flow with a two-step hot water preparation scheme.

6. References

Ahonen, M, Kosonen, R. and Kekkonen V. 1995. "The new low temperature HVAC-System that reduces combined heat and power plant's costs", *Proceedings of 5th International Symposium on Automation of District Heating Systems*, Helsinki.

Børresen B.A., and Thunem A.J. 1984. "HVAC-control models", SINTEF, STF48 F84019, Trondheim.

Chapman, A.J. 1987. "Fundamentals of heat transfer", Macmillian Publ. Comp., New York.

Frederiksen, S. 1995. "District Heating Substations for Low Temperature Operation", *Draft report*, International Energy Agency - District Heating and Cooling, Annex IV, Efficient Substations and Installations project.

Gnielinski, V. 1976. "New equations for heat and mass transfer in turbulent pipe and channel flow", *Int. Chem. Eng.*, Vol.16, no.2, p.359.

Grétarsson, S.P., Valdimarsson, P. and Jónsson, V.K. 1991. "Heat transfer modelling of a plate radiator for district heating applications", *Int. Journ. of Energy Research*, Vol.15, p.301.

Gummérus, P. 1989. "Analys av konventionella abonnentcentraler i fjärrvärmesystem", *Doctoral thesis*, Department of Energy Conversion, Chalmers University of Technology, Göteborg.

Hjorthol, E.M., and Ulseth, R. 1992. "Consumer Heating System Simulation (CHESS)", International Energy Agency - District Heating and Cooling Project, Annex III, Report-P5, Sittard

Johnson A. and Besant R. 1995. "Validation of the heating coil model used in the Consumer Heating System Simulation program", *Draft report*, International Energy Agency - District Heating and Cooling, Annex IV, Efficient Substations and Installations project, EnEff Technologies Inc, Saskatchewan.

Jonsson, G. 1990. "Parameter estimation in models of heat exchangers and geothermal reservoirs", *Doctoral Thesis*, Department of Mathematical Statistics, Lund Institute of Technology, Lund.

Nilsen, Ø. 1994 a. "System for fjernvarmetilpasset kobling på sekundærsiden", *Norsk VVS*, no.4, 1994, p.29.

Nilsen, Ø. 1994 b. "90/70-anlegg lite egnet til utnyttelse av spillvarme", *Norsk VVS*, no.8, 1994, p.14.

Rikheim, H. 1987. "Dynamiske egenskaper for varmluftsbatterier - praktiske målinger og datasimuleringer", *Diploma thesis*, Department of Heating and Ventilating, Norwegian Institute of Technology, Trondheim.

Rydberg, J. 1945. "Beräkning av maximala tappningseffekten för varmvattens-beredningsanleggningar", *VVS-Tidsskrift för Värme- Ventilasjons- och Sanitærteknik*, Vol.16, no.5, p.70.

Seider, E.N. and Tate E.G. 1936. "Heat Transfer and Pressure Drop of Liquids in Tubes", *Ind. Eng. Chem.*, Vol.28, p.1429.

Steiner M. 1989. "Low order dynamic models for heat exchangers", *Proceedings of International Symposium on District Heating Simulation*, Reykjavik.

Volla, R. 1994. "Gevinster ved økt temperaturdifferanse i fjernvarmeanlegg", The Norwegian District Heating Association, RE-35/1994, Oslo.

Volla, R. 1996. "Consumer Heating Systems for District Heating - Development by System Simulations and Service Hot Water Measurements", *Doctoral Thesis*, Department of Refrigeration and Airconditioning, Norwegian University of Science and Technology, NTH report no.1996:3, Trondheim.

Vrins, E. and Versteeg H. 1995. "Low Temperature Systems (LTS)", Report summary, Novem, Sittard.

Winberg, A. and Werner, S. 1987. "Avkylning av fjerrvärmevatten i befintliga abonnentcentraler", Thermal Engineering Research Institute (VärmeForsk), Stockholm.

Part II

DISCUSSION OF LOW TEMPERATURE SUBSTATIONS: MOTIVES, STATE-OF-THE-ART & SOME KEY ISSUES

Svend Frederiksen

Lund Institute of Technology
Department of Heat and Power Engineering

February 1996

Table of Contents

1. Objective

Substations provide the interface between district heating (DH) networks and internal distribution systems in buildings. This part report indentifies some types of technological solutions which have favourable thermodynamic properties, i.e. they are in accordance with an overall move in DH practice towards operation with low network temperatures.

The substation types considered here are modifications of well-established heat exchanger assembly types found in Scandinavian low-temperature hot water DH networks. This tradition has also been the starting point for an IEA-study performed at NTH/SINTEF in Trondheim, Norway, to which this work paper is related.

Before entering into the discussion of the selected technological solutions, the basic premise of low-temperature solutions is discussed, in order that the focus of interest is seen in a proper perspective.

2. Motives for low temperature operation

There are many arguments in favour of low DH network temperatures. As a first classification the following main arguments can be listed:

1. Improved generation plant performance
2. Reduced heat losses
3. Reduced circulating water flowrate (at lower return water temperature)
4. Cheaper pipeline technology

Low network temperatures can be achieved by a combination of various choices and measures. Some of the associated decisions may cause increased investment cost, while other decisions may result in lower network temperatures without added cost. For instance, network temperatures can be lowered by installing bigger radiators in connected buildings, a measure which clearly will increase investment costs. In contrast, more thermodyanically efficient substation connection schemes may result in lower network temperatures without necessarily calling for more expensive equipment. Ultimately, of course, investment costs will increase below certain network temperatures, so that a trade-off must be made when deciding on the appropriate temperature level.

Below, each of the 4 main attractions listed above will be commented on shortly.
It will be seen that in some instances arguments pertain both to forward and return temperatures. In other cases, a certain argument is clearly linked to, either a low forward temperature, or a low return temperature.

Ad 1 (Improved plant performance):

The strongest argument in favour of low network temperatures probably is that low network temperatures can be utilized for improved CHP plant performance. In the next section this fact will be dealt with separately, along with temperature considerations for centralized heat pump plants.

Even with heat-only generation, however, there may be benefits. A particularly interesting instance of this occurs when temperatures become low enough to make recovery of latent heat from combustion gas water vapour possible. Depending on the sort of fuel and the type of generation plant, this becomes feasible at temperatures below typically 40 - 60°C. Such a facility, which is primarily made possible by a low DH return temperature, may result in a boiler efficiency in excess of 100%, based on the lower calorific value of the fuel, as is customary in Europe when specifying boiler efficiency.

Ad 2 (Reduced heat losses):

For a given DH network operating at varying water temperature level, as a first estimation it can be assumed that heat losses are proportional to the difference between the ambient temperature and the arithmic mean of the forward and return water temperatures, i.e.:

$$Q_{HL} = const.(t_{fr} - t_a) \tag{1}$$

defining: $$t_{fr} = (t_f + t_r)/2 \tag{2}$$

Thus, the relative influence of changed mean temperature t_{fr} can be expressed as:

$$\frac{1}{Q_{HL}} \frac{dQ_{HL}}{dt_{fr}} = \frac{1}{t_{fr} - t_a} \tag{3}$$

If, e.g., $t_{fr} - t_a = 50°C$, this gives:

$$\frac{1}{Q_{HL}} \frac{dQ_{HL}}{dt_{fr}} = 2\%/K$$

When assessing variations in heat losses at different temperature levels in a design situation, things become a little more complicated, although as with differing operating temperatures the general tendency will be that lower temperatures cause reduced heat losses.

If t_f is lowered at constant t_r the associated increased flowrate for a given heat load requires bigger pipeline diameters to restrict pressure losses. This in turn will increase the surface area of the pipes. In the extreme, the net result could be a higher heat loss.

Ad 3 (Reduced water flowrate):

If the return temperature is lowered, e.g. due to better cooling of DH water passing substations, the circulating flowrate becomes smaller, for a given forward temperature and for a given heat load. This will reduce pumping costs, due to smaller pressure drops in pipelines.

The economic value of such reduced pumping costs will depend very much upon the type of DH network. Even in networks serving big cities the pumping power demand may be no greater than a fraction of a percentage of the heat load served, typically representing 10% of the size of the heat losses, i.e. P_{pump} = typically 0.1 x Q_{HL}. In such a case the economic value of reduced pumping power will be only marginal.

However, in long transmission lines pumping power demand may amount to several percent of the heat load, in which case a reduced pumping power becomes more significant from an economical point of view.

Apart from reduced pumping costs there may be further benefits to achieve from reduced flowrates because of better primary water cooling. When in a given system distribution pipes are already utilized to a maximum, better primary water cooling in buildings already connected to the network can create possibilities for connecting further buildings, without installing new distribution pipeline capacity.

Ad 4 (Cheaper pipeline technology):

In classical optimization studies of DH network temperatures, specific investment network costs for pipes were usually being related only to the pipe diameter, while the temperature in itself was considered to have only minor influence on pipeline investment costs. The influence of the forward temperature on the costs would only be indirect in that a higher forward temperature for a constant return temperature would reduce the flowrate and thereby the diameter (for given pressure losses).

This way of representing pipeline investment costs in optimization models is reasonable when the classical type of mains technology can be presupposed, i.e. pipes are installed in concrete ducts surrounded by mineral wool thermal insulation and are allowed to perform free thermal expansion.

Costs for modern plastic-shielded, bonded polyurethane insulated pipes, and other types of mains technology, in contrast tend to become lower when operating temperatures are lowered for a given pipe dimension. This point will be developed in section 4 below.

A practical example of the economic significance of lower temperatures:

Malmö is the third largest Swedish town with a population of 230,000 people. A majority of the town's buildings are connected to the DH network. The connected heat load is around 1500 MW.

The Malmö Energy Utility, as a rule of thumb, estimate [ref. 2] that a lowered mean network water temperature can be attributed a value of around 1 M Swedish Crowns per °C per year (equivalent to around 130,000 USD per °C per year). This gain is mainly attributable to reduced heat losses. It is also estimated that the value of each degree's lowered temperature level will increase in the years to come, due to gains in power output from a growing installed CHP capacity.

3. Implications for CHP and heat pump operation

Lower DH temperatures can result in improved plant performance, in the sense of either the First, the Second, or both, Laws of Thermodynamics, depending on the type of plant. The benefits from this may be, e.g., smaller fuel consumption for the same output, bigger electricity output from a CHP (Combined Heat and Power) plant, or lower electricity demand for a central heat pump plant.

In the case of a CHP, two commodities, electric power P and useful heat Q, are produced. Consequently, at least two types of efficiences must be used to characterize plant performance [ref. 3]. Very often the following two efficiency measures are adopted:

The Energy Utilisation Factor:

$$EUF = \frac{W+Q}{F} \tag{4}$$

and the Second Law Efficiency:

$$\eta = W / F \tag{5}$$

For a fossil fuelled plant the fuel energy input F is given as the product of the fuel mass flow and the calorific value of the fuel:

$$F = m_f CV \tag{6}$$

In the expressions for EUF and η the electric power output, P, is sometimes used directly in place of the mechanical work output W. P is related to W by the electric generator efficiency η_g:

$$P = \eta_g W \tag{7}$$

In some instances it is profitable to replace the Second Law Efficiency by the

Power to Heat Ratio:

$$PHR = P / Q \tag{8}$$

(sometimes instead termed the coefficient of performance).

When in CHP plant the EUF is affected by network temperatures, it is usually the return temperature, t_r, which has the strongest influence. E.g., in an open-cycle gas turbine plant the hot exhaust gases normally give up heat to a water circuit in a counter-current flow arrangement heat exchanger. In cases when this heat exchanger is connected directly to the DH network without recirculation, a lowered t_r will cause a lowered stack temperature, thereby increasing Q without affecting the size of P.

In the case of steam plant CHP, in contrast, a lowered t_r may have little or no influence on the EUF, depending on the type of arrangement employed for boiler air preheating. If all preheating is performed by a flue-gas driven, recouperative or regenerative heat exchanger, t_r may have very little influence on the flue gas stack temperature.

In the case of closed-type CHP plant, the network forward water temperature, t_f, and sometimes also the return temperature, t_r, may have significant influence on the power output P, and thus on the Second Law Efficiency.

In the case of an ideal Carnot cycle, cf. fig. 1, the efficiency is given by the absolute tempeatures of heat reception, T_1, and of heat rejection, T_2:

$$\eta = \frac{T_1 - T_2}{T_1} \tag{9}$$

Thus, if T_2 is lowered to a value T_2' at a constant T_1, η improves. With the simple and theoretical type of plant given by the figure, t_f influences plant performance exactly as does T_2, whereas t_r has no influence at all on η.

For a given T_1 and given network temperatures t_f and t_r, the ideal, reversible closed cycle is given by a sliding temperature T_2, instead of a constant temperature, to match the heat-up of network water, as shown in fig. 2. Here, the effective mean temperature of heat rejection will be (converting degrees Centigrade to absolute temperatures):

$$T_2 = (t_f + t_r) / 2 \tag{10}$$

In this case, changes in t_f and in t_r will have equally large influence on plant efficiency.

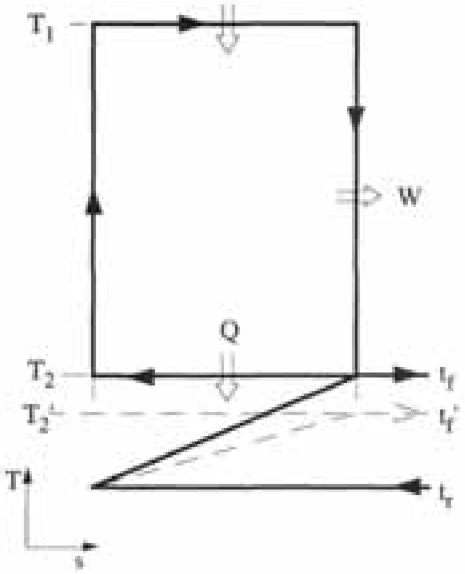


Fig. 1. *Ideal, simple Carnot CHP closed cycle.*

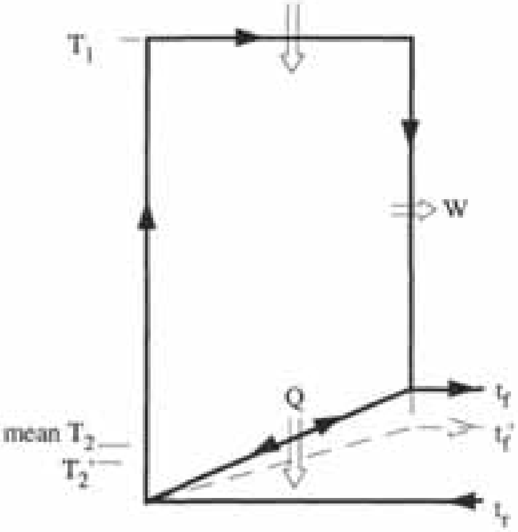


Fig. 2. *Ideal Carnot CHP closed cycle with sliding heat rejection temperature.*

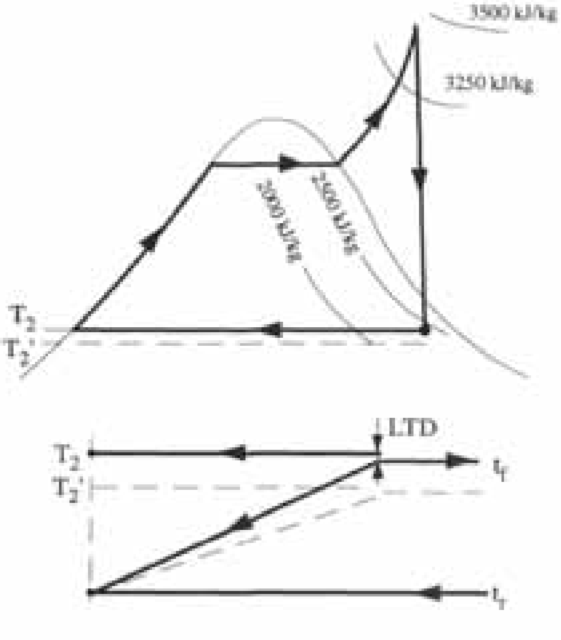


Fig. 3. *Simple Clausius-Rankine steam back-pressure CHP plant.*

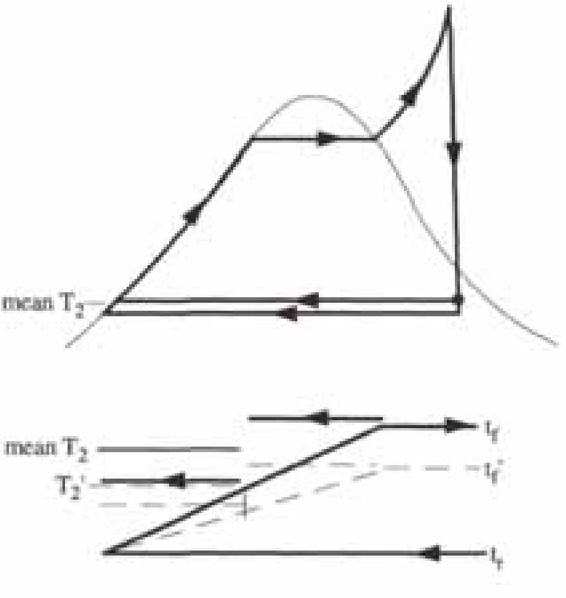


Fig. 4. *Clausius-Rankine steam back-pressure CHP plant with 2-stage heating.*

The most common type of closed CHP cycle is the classical Clausius-Rankine steam cycle. Figs. 3 and 4 depict simple Clausius-Rankine steam cycles resembling the ideal cycles in figs. 1 and 2. For simplicity, renegerative feedwater heating (normally adopted) has been omitted here. Temperatures of heat rejection are given by steam condensation at the bottom of the cycles. The two condensation temperatures of fig. 4 represent a first step towards the theoretical limiting case of infinitely many heat-up steps for DH water, and a corresponding infinite number of condenser stages.

In the real world heat transfer surface areas are not infinitely large, so that the steam condensing temperature, T_2, will typically be some degrees higher that the DH forward water temperature, i.e. there is a Least Temperature Difference, LTD. For a given plant, a lower incoming return water temperature tends to make LTD little smaller. This means that in fig. 3 t_r will exert only a small influence on η.

In the case of two-stage DH water heating (fig. 4) the influence of t_f on Second Law efficiency is reduced compared to single-stage heating (fig. 3), whereas the influence of t_r is increased.

At high DH heat loads, the heat output Q from a CHP plant is often supplemented by heat from other sources, typically by a contribution Q_{HOB} from a Heat-Only Boiler, cf. fig. 5. In this case a lowering of t_f will not have full impact on the CHP plant, since it is the intermediate water temperature, t_m, not t_f, which is linked to the cyclic temperature of heat rejection, T_2.

The size of network temperature influence on steam plant CHP can be assessed by inspecting how isothermals and isenthalps intersect below the saturation line in a Mollier steam-water state diagram. Cf. fig. 3. Following a typical steam expansion line of a turbine, a sensitivity of around 5 kJ /kgK is found. For moderately advanced initial steam data and no steam reheating the turbine steam enthalpy drop is in the order of 900 KJ /kg. This gives a size of the

Work to Temperature Sensitivity:

$$\frac{1}{W}\frac{dW}{dT_2} = 5.5\%/\ K$$

Regenerative feedwater heating and steam reheat both cause a rise of the mean temperature of heat reception for the cycle, whereby the Work to Temperature Sensitivity is reduced somewhat.

Accurate calculations of this sensitivity will have to take into account further considerations, e.g. the fact that the steam condensation enthalpy depends somewhat on temperature. For a given Q this introduces a variation of steam flow, whereby the size of P is in turn affected. Normally, however, this effect will not be large.

In detailed considerations a distinction must also be made between effects of network temperature variation in a design stage and effects of variations caused by varying operating conditions for a given plant.

For instance, at small variations in condensation temperature, the turbine expansion end point in the Mollier diagram can with reasonable accuracy be assumed to shift upwards or downwards in the same direction as the expansion line itself. This applies both to different design cases and to performance variations. But at large temperature variations a given turbine will usually respond in a highly non-linear way instead, caused by the end-point moving towards higher entropy values in the diagram.

Figs. 3 and 4 refer to back-pressure CHP plants. The influence of DH temperatures on the performance of pass-out steam turbine plants bears resemblance to this, that is, also this case there will be a shift of a turbine end-expansion point in the Mollier diagram.

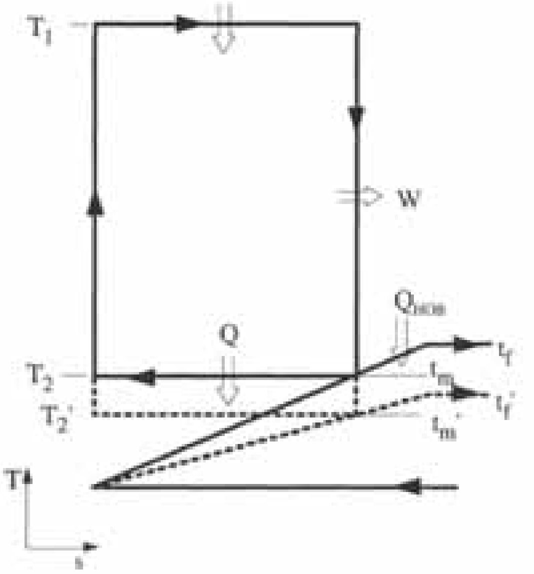


Fig. 5. *Ideal simple Carnot CHP closed cycle plant in series with topping Heat-Only-Boiler.*

In general, combined cycles based on open-cycle gas turbines and heat recovery steam cycles are less sensitive to DH network temperatures, compared to pure steam plants. This is because only part of the total work output is affected by varying network temperatures.

In the case of diesel and gas engines, the work output is rather unsensitive to network temperatures. However, maximum lube oil temperature for the engine usually poses an important temperature restriction.

In the case of a central heat pump serving a DH network, cf. fig. 6, plant performance is strongly influenced by network temperatures [ref. 4].

The Coefficient Of Performance:

$$COP = \frac{Q}{W} \tag{11}$$

may be written as:

$$COP = \eta_c \frac{T_1}{T_1 - T_2} \tag{12}$$

in which the Carnot-factor, η_c, accounts for Second-Law deviations of a realistic cycle from the idealized Carnot-cycle depicted in the figure. According to one reference [ref. 5], the Carnot-factor of a heat pump typically varies between 0.2 - 0.63 for heat pumps, ranging from 1 to 10,000 kW's.

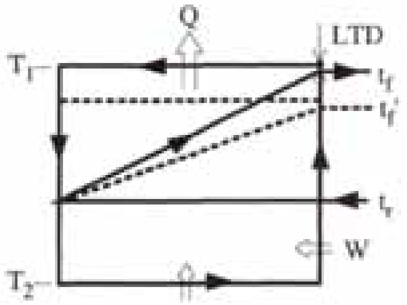


Fig. 6. *Ideal Carnot-cycle heat pump plant, lowered DH forward temperature.*

Relative variations of work input for constant heat delivery Q, caused by varying temperature T_1, may be calculated as:

$$-\frac{1}{W}\frac{dW}{dT_1} = \frac{1}{COP}\frac{dCOP}{dT_1} \tag{13}$$

Assuming that η_c is constant, (12) yields:

$$\frac{1}{COP}\frac{dCOP}{dT_1} = -\frac{T_2}{T_1}\frac{1}{T_1 - T_2} \tag{14}$$

As a numerical example,

$T_1 = 343$ K (70°C) and $T_2 = 293$ K (20°C) yield:

$$\frac{1}{COP}\frac{dCOP}{dT_1} = 1.7\%/K$$

Neglecting variations in Least Temperature Difference (LTD), this is identical to the sensitivity for variation in forward temperature t_f.

With the simple type of heat pump plant assumed here, variations in return temperature, tr, only has minor influence on plant performance, cf. fig. 7. A lowering to the level of t_r' only causes a minor influence, due to a reduction in LTD.

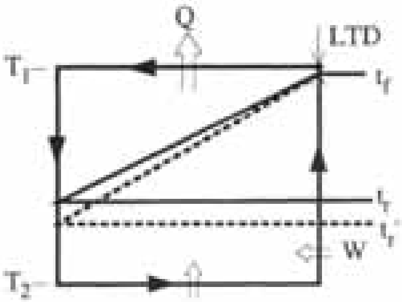


Fig. 7. *Ideal Carnot-cycle heat pump plant, lowered DH return temperature.*

Sometimes a central heat pump only produces a minor part of the total heat delivery to a DH system. In such a case the plant is typically connected to a return pipe. Thus, there will be a strong influence of t_r on W and on COP.

4. Temperature practices in various countries

Apart from the basic distinction between steam and hot water distribution technologies, there are a number of technological variations in DH technology which are reflected in the various types of substations utilized in different countries. In part those differences result from differing traditions in temperature practice.

By tradition, various countries generally adopted different network design supply temperatures as follows:

* Former Soviet Union: 160 - 200°C
* Eastern Europe: Around 160°C
* Western Germany: 130°C
* Finland and Sweden: 120°C
* Denmark: 80 - 120°C

Those North American networks which have been based on hot water DH technology, in general seem to have adapted more or less to the Finnish-Swedish design practice [ref. 6].

In many countries the general trend has been towards lower temperature levels. Thus, current European piping standardization work seems to favour a practice of a maximum of 110°C.

For a number of years now Swedish DH utility standards prescribe that in general substations be designed to be capable of satisfying all heating demands at a maximum supply temperature of 100°C, although safety requirements should still be fulfilled at DH temperatures up to 120°C [ref. 7].

Denmark seems to be the country in which the average network temperatures generally are the lowest, although significant differences exist between practices at various utilities [ref. 8]. For a number of years State Building authorities have worked on a possible extension of the

Building Code, prescribing an extreme low temperature practice of a maximum supply temperature of 70°C.

The international trend towards lower network temperatures stems primarily from reduced costs in piping technology and from improvement in thermodynamic system performance.

A maximum supply temperature of around 130°C permits the use of preinsulated pipes with an outer plastic shield pipe and polyurethane insulation foam between the shield pipe and the medium carrying pipe, which is normally made of mild steel or copper. In Western Europe this type of mains has become the general first choice since a number of years and has replaced older mains types based on steel pipes placed in underground concrete culverts. The widespread use of preinsulated pipes has brought about a substantial reduction in DH installation costs.

As a result of strength of materials investigations in the last few years, an agreement is now emerging that a maximum supply temperature in the order of 100 - 110°C opens up for unproblematic adoption of installation practices which omit stress reduction by medium pipe preheating prior to fixation in the ground. This reduces the time necessary for installing the pipes.

Still lower maximum supply temperatures in the order of 70 - 90°C make possible the use of mains designs in which medium carrying pipes are made of plastic, which due their flexibility contribute to further installation cost reductions.

Lower limits to DH network temperatures are set by temperature levels existing in heat and water distribution systems in buildings. These levels are in turn determined according to differing legal and professional standards in various countries. Also, engineering design practices varied over the years. Thus, great variations exist between various buildings.

Older and newer standards for radiator systems in some important DH countries are shown in fig. 8.

In Sweden a legal requirement created in the early 1980ies [ref. 9], stipulates that radiator systems in most buildings connected to DH should be designed for a maximum supply temperature of 60°C. Outside existing DH areas, 55°C design temperature is required. A main motive behind this legislation has been an ambition to facilitate a gradual introduction of solar building heating, in addition to a general purpose of creating favourable conditions for the utilization of low-temperature heat sources for building heating.

It is clear that low design temperatures for radiators call for comparatively big surface areas and thus imply higher installation costs in buildings.

Inventories made in various countries have established that radiator systems often possess low temperature potentials when real sizing is compared to minimum sizing according to legal demands or standards. Typically, there is a reserve in the order of 5 - 20°C, depending on the age and type of building.

Domestic hot water temperatures distributed to individual taps generally are within a temperature span of 37 to 65°C, depending on a number of more or less conscious decisions and requirements. A hot tap water temperature around 50°C is normally considered suitable for most household applications of hot water. Traditionally, the hot water temperature has been selected so as to provide general comfort, to minimize the risk of scalding accidents, and to avoid excessive scale formation in hard potable waters.

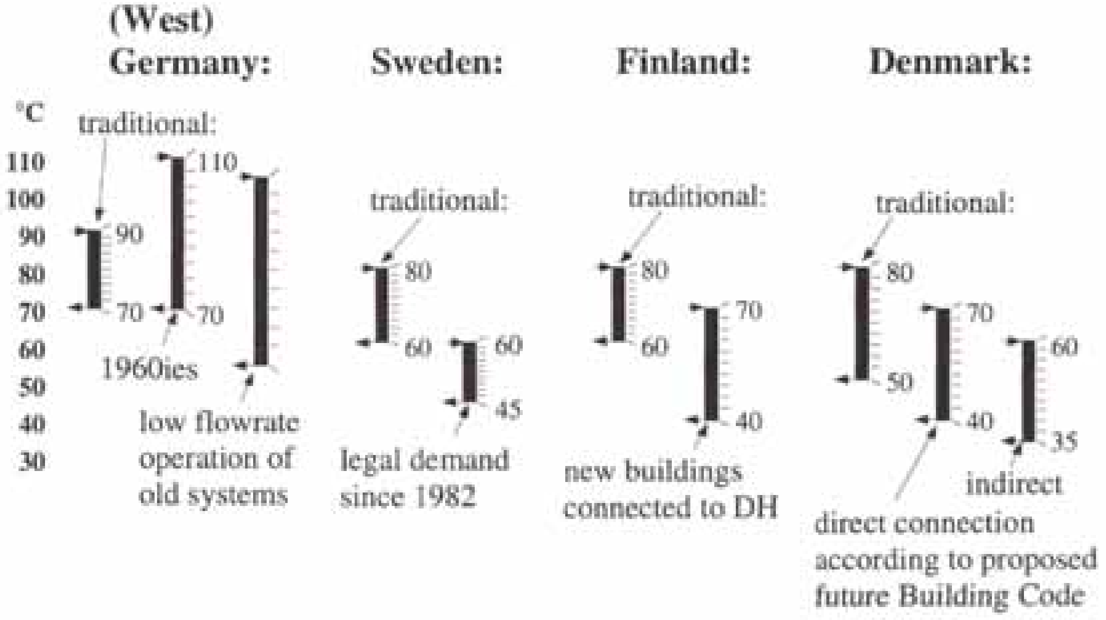


Fig. 8. *Older and newer radiator water temperatures according to standards in various countries.*

In the wave of energy savings following the first internal oil crises in 1973/74, domestic hot water temperatures were sometimes lowered. Nowadays most countries instead prescribe higher temperatures, in order to minimize the risk of Legionella bacteria multiplication, a hazard that was not known at all in the early energy saving days [ref. 14]. Prescribed levels differ from one country to another and are sometimes graduated according to the type of installation. Typically, it is nowadays required that potable water be heated to a minimum of 50 - 60°C.

5. Main types of substation technology

A main dividing criterion between different types of substation schemes relates to the degree of hydraulical separation between the various types of water networks which interact in the substation.

Most DH systems are closed in the sense that the open potable water system is separated from the DH network by some kind of heat transfer surfaces. In open networks, which are e.g. used in some Russian towns, hot water is instead drawn off from the network and used as domestic hot water.

In closed systems heating of potable water may be performed in various types of equipment. A basic distinction is made between instantaneous water heaters and heaters which incorporate some kind of hot water storage. These in turn can be subdivided into calorifiers with internal heat transfer, and systems in which a separate storage tank is fed with hot water from an external heat exchanger.

Below we shall restrict our considerations to solutions with instantaneous water heaters, without any detailed discussion. Many relevant arguments can be made for or against the choice of a certain heater type, and it is a fact that on this point DH practices differ in various countries.

From a pure thermodynamic point of view it may be claimed that storage solutions with built-in heat transfer, although possessing other advantages, suffer from the basic difficulty of mixing heat transfer and storage, a restriction which tends to hamper possibilities for favourable temperature characteristics.

Another basic line of division between various types of substation equipment is the distinction between direct connection of radiator systems to DH networks and indirect connection, where heat exchangers provide hydraulical separation. Again, differing national traditions determine the choice of technology on this point. Below we shall mainly be concerned with indirect installations, although many considerations pertaining to connection schemes will be of relevance for both direct and indirect connections.

Thermodynamically, direct connection should be preferred, since this type of solution avoids the inevitable temperature losses associated with a separating heat exchanger. A main safety argument against direct connection is the risk that unintended pressure transients created in the DH network are more readily transmitted to the building heating system. This may cause damage in the form of radiator blasting and big water leakages.

In the last few years new types of indirect connection schemes have been suggested, incorporating pressure separation equipment which promises a higher degree of safety against such accidents [refs. 15 & 16].

In some types of substations, hydraulical separation is carried a step further, so that heat transfer from the DH network to domestic hot water passes two heat transfer surfaces. Such double heat exchange can be realised in several alternative ways.

In some types of installation, the two heat transfer surfaces are accomodated in the same heat exchanger. Tube heat exchangers may be designed according to the tube-within-a-tube principle. Alternatively, tubes or plates may be arranged adjacent to eachother, e.g. in the form of double wall heat exchangers. The two heat transfer surfaces may be arranged to be in direct mechanical contact, or a sweet water loop may separate the two surfaces. In the latter type of substations, radiator circuit water is normally used as the intermediate medium.

Double heat exchange provides an additional safety against leakages between DH water and potable water. In some instances local safety demands stipulate double heat exchange, but so far the principle has not been generally prescribed. The additional safety must be weighted against generally higher installation costs.

Double heat exchange is also associated with a double temperature loss, which may be more or less significant for the overall substation operating characteristics, depending on the type of heat exchange technology and on the primary and secondary temperature levels.

The higher the average temperature level in the radiator circuit is, compared to the average temperature level in the domestic hot water circuit, the smaller is the thermodynamic penalty associated with double heat exchange. Therefore, it seems natural that double heat exchange is more common in Germany [ref. 13], in comparison to Scandinavia.

Also, double heat exchange lies closer at hand when DH network temperatures are relatively high. For one thing, the necessary heat exchanger sizes are not as big.

Another important consideration relates to the risk of scale formation in the case of hard potable water. If the primary side temperature is high in a water heater, this tends to create a high local surface temperature close to the domestic hot water outlet, increasing the risk of scale precipitation. Although the local maximum surface temperature can be minimized by proper heat exchanger design, it is impossible to completely avoid temperature peaks. Double heat exchange reduces the primary side temperature in water heaters and thus reduces the risk of scale formation.

In the following sections we shall not consider double heat exchange substations any further.

6. First key issue: Avoid unnecessary mixing devices!

In substations mixing valves or other types of mixing devices are sometimes quite in place. But in other cases mixing represents an unnecessary thermodynamic loss and should be avoided.

This thermodynamic loss relates to the Second Law of Thermodynamics. Theoretically speaking, mixing of two flows of different temperature is associated with an exergy loss, i.e. a loss of the potential for converting heat into mechanical work. Sometimes such a loss in a substation may be associated with less mechanical work performed in a combined heat and power plant, due to an unnecessarily high temperature level in the DH network.

Fig. 9 is a simple schematic of a heat exchanger arrangement with mixing valves in both primary and the secondary water circuits.

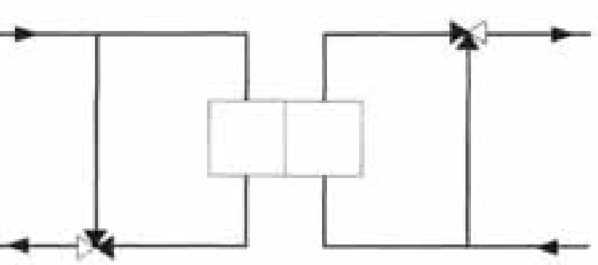


Fig. 9. *Substation heat exchanger with mixing valves, both in primary and secondary water circuits.*

The mixing valve on the primary side causes an increased primary return temperature, for a given return temperature leaving the heat exchanger. Analogously, the mixing valve causes a degradation of supply temperature, which could be unnecessary, depending on the context. At least, for a given secondary supply temperature after the mixing valve, a higher primary supply temperature is needed when there is a temperature drop in the mixing valve.

By tradition, primary side mixing devices have been used in many east European DH systems with the aim of keeping DH network flowrates constant, when variations in heat demands call for varying primary flowrates passing heat exchangers.
This practice has caused deteriorated primary water cooling in substations. When modern control equipment is available, above all variable speed controls for network circulation pumps, mixing valves maintaining a constant flow are superfluous and thermodynamically harmful.

When buildings are heated by an individual boiler instead of a DH network, it is common practice to have a secondary side mixing valve, in which the radiator supply temperature is automatically adjusted to the actual outdoor air temperature, according to a prescribed controller curve.

Sometimes, when buildings formerly heated by individual boilers are connected to a DH network, this type of temperature control is taken over without modification. However, retrofitting should include replacement of the mixing device by a two-way valve on the primary side, in series with the heat exchanger.

Mixing valves are sometimes in place as safety devices, e.g. when taking down the temperature level of hot domestic water leaving a heat exchanger, in the event of a faulty function of the two-way thermostatic control valve.

It might be argued that even in normal operation a mixing valve represents no thermodynamic loss, if it lowers a domestic hot water temperature to a prescribed level, and if the primary supply temperature is also fixed. When heat is passed from the DH network to the domestic hot water circuit, an exergy loss will take place any way. When no mixing is performed on the secondary side, a greater exergy loss is instead incurred in the heat exchanger, when heat is passed from the higher to the lower temperature level.

This rather loose way of reasoning is correct in the theoretical limiting case of an infinitely effective heat exchanger. Such a device would cause the primary water to be cooled down to the incoming cold town´s water temperature level, irrespective of whether mixing takes place or not on the secondary side.

However, with a real-world, finite heat exchanger the situation is different. When mixing takes place, the primary to secondary water temperature difference between supply temperatures becomes smaller. From heat exchanger theory (e.g. utilising the logarithmic mean temperature concept) it then follows that the temperature difference at the other end of the heat exchanger increases. I.e., for a given cold water temperature, the primary return temperature goes up.

The size of this effect depends upon the actual load case and heat exchanger size. Still, from a theoretical point of view no mixing on the secondary side is the best choice.

7. Second key issue: Space heating radiator flowrate optimization

In modern radiator systems for space heating the supply temperature is normally adjusted to variations in heat load demand according to a controller curve. In many buildings there is no speed control of circulation pumps, and the flowrate is more or less constant.

By selecting a proper controller curve, the heat rate emitted from radiators may vary in such a way that a more or less constant indoor temperature is maintained. In its basic form this kind of control is a pure feedforward type of control.

As a refinement, individual radiators may be equipped with thermostatic valves, which add a feedback type of control. For instance, such valves may prevent excessive heating of rooms exposed to solar heat gain. As a further refinement, variable speed pumping is sometimes employed.

In the case of individual boiler heating of a building the question of which control philosophy should be selected is mainly a question of balancing comfort and energy savings.

When instead buildings are heated by sources which for their efficiency are temperature dependent, further requirements may be formulated on the control strategy. Examples of such heating sources are, solar heating, heat pumps, and DH.

In the case of DH a reasonable criterion for the radiator supply temperature t_{rf} is that the primary return temperature t_{r1} should be as low as possible. We shall discuss consequences of this criterion, first for the case of direct connection of a radiator system, and next for the somewhat more complicated case of indirect connection, where the performance of the intersecting heat exchanger must be taken into account.

When space heating systems are connected directly, the lowest return temperature is simply achieved by passing the incoming primary supply temperature directly into the radiator circuit. Degradation of the temperature level by admixing return water causes the return temperature to increase. This is seen by regarding the radiators as heat exchangers and considering the LMTD (logarithmic mean temperature difference) for the radiators, with radiator water as the primary medium and the indoor air (of constant temperature along the radiator surface) as the secondary medium.

With high DH forward temperatures, this type of operation leads to low water flowrates in radiators. We shall therefore term it: *Low flowrate operation.*

Safety demands and hygienic considerations may not permit DH forward temperatures to enter radiator circuits uncooled in all instances. In networks where a too high DH supply temperature may occur, the substation should be equipped with a mixing device which limits the radiator supply temperature. At lower DH supply temperatures the automatic system controlling the mixing device may allow the incoming supply water to pass on without any admixing.

For many years low flowrate operation has been practised in directly connected radiator systems in Germany and Denmark. In Germany the method is commonly combined with usage of so-called 'Thermostatische Feinregulierventile' [ref. 17], i.e. radiator control valves which are in principle normal thermostatic valves, but of such a quality that they allow for a stable control of the rather small flowrates which pass the valves.

In Danish single-family dwellings, directly connected to DH, low flowrate operation is commonly combined with usage of a further type of thermostatic radiator control valves. These valves are fitted into the return pipe from a radiator and maintain a preset return water temperature. Usually the DH tariff is such that the building owner has an economic incentive to make such hand adjustments of these return valves that a lowest possible return temperature is achieved.

The case of indirectly connected radiator circuits is discussed in relation to fig. 10. We here assume that the heat rate q_r, passing both the heat exchanger and the radiators, is fixed, as is the incoming primary supply water temperature t_f. The radiator supply temperature t_{rf} is variable (within limits, of course), together with the flowrate m_r in the radiator circuit. When a certain value is selected for t_{rf}, m_r is also determined to a certain value (dynamic effects are not considered here). It is a system with one degree of freedom.

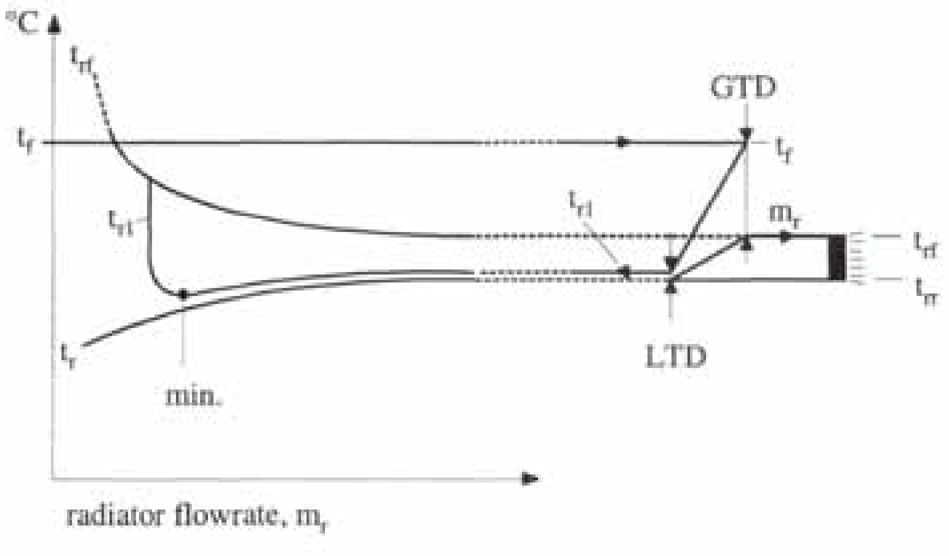


Fig. 10. *Varying primary circuit return temperature t_{r1} at varying radiator flowrate m_r for constant heat load.*

If t_{rf} is increased, t_{rr} goes down, and m_r becomes smaller. To assess the effect of this on the primary return temperature, t_{r1}, it is necessary to consider the LMTD of the heat exchanger. The LMTD is a weighted mean temperature based on the two terminal temperature differences, the greater temperature difference, GTD (between primary and secondary supply temperatures) and the least temperature difference, LTD (between the return temperatures). In Germany and Scandinavia, LTD is sometimes termed 'Grädigkeit'.

Depending on changes in heat transfer coefficients, LMTD may change somewhat (to maintain a constant heat rate q_r), but in general not very much. Therefore, the smaller GTD following the higher t_{rf} normally results in a bigger LTD.

We thus have two opposing effects on the primary return temperature t_{r1}: The secondary return temperature t_{rr} is lower, but the temperature difference LTD is bigger. The net result may be a lowering or an increase, depending on the load situation and the starting point for the variation.

This implies that there will be an optimum secondary supply temperature t_{rf} which gives a minimum t_{r1}. In this optimum, an infinitesimal change of t_{rf} will produce no change in t_{r1}. I.e., the first derivative of t_{r1} is 0 at a minimum. Cf. fig. 10.

The theoretical limit case of an infinitely effective heat exchanger with LTD = 0 is equivalent to the direct connection case, where t_{rf} should be equal to t_f for minimum t_{r1}. In the real case of a finite heat exchanger, the optimum value of t_{rf} becomes higher if a more effective heat exchanger is chosen.

A consequence of this is that the choice of heat exchanger size should be reviewed if low flowrate operation is considered. With conventional radiator flowrates, heat exchangers are normally designed to give an LTD of only a few degrees. Therefore, the marginal benefit from choosing an even bigger heat exchanger is small. With low flowrate operation, however, the marginal benefit becomes bigger and should therefore be considered. The consequence may be that a substantial increase in heat exchanger size is called for.

For a given installation and a given temperature vs. load curve for the primary supply temperature, a certain controller curve for t_{rf} vs. load represents the thermodynamical optimum, producing a minimal t_{r1} at all loads.

In practice it may be advisable to choose a t_{rf}-controller curve which is somewhat lower than the theoretical optimum. This consideration may be particularly relevant in big buildings, where low flowrates may cause problems with time lags and big temperature gradients in supply pipes, caused by heat losses. As with all differentiable mathematical functions, the minimum variable changes only a little close to optimum. Still, the theoretical optimum controller curve will be of value to establish, as a reference for making a rational choice of control strategy.

Depending on the way low flowrate control of radiator systems is realized, there may be a problem in that high secondary supply temperatures tend to make it difficult for the utility to lower the primary supply temperature in the network. This may be unfortunate, since in some cases (depending in type of heat producing plants etc) a lowered supply temperature may be of greater value than a lowered return temperature.

Therefore, before adopting low flowrate radiator controls, it should be considered how to make it as flexible as possible, so that a modification of the controller curve can be made easily in future situations. If low flowrates are achieved by hand operation of balancing valves in the radiator system, the modification back to higher flowrates may be a tedious procedure. Thus there is a need for equipment which can perform the modification in a simple way. The ideal would be an automatic control which selects the optimal supply temperature t_{rf}, independently of how the DH company chooses the primary supply temperature t_f at various heat loads.

8. Third key issue: Combined heat exchanger and flowrate optimization

The size of substation heat exchangers basically relies on a trade-off between running costs and investment costs: A bigger heat transfer surface area can be utilised for improved system performance and may therefore lessen net running costs. On the other hand a bigger heat exchanger usually represents a higher investment cost.

In addition, it is customary to take into account various aspects of heat exchanger size:

- A reasonable margin should be allowed for deteriorated performance caused by fouling.

- The primary and secondary circuit flow areas should not be too large, i.e. the heat exchanger should not be too 'wide', since this tends to cause low heat transfer coefficients.

- On the other hand, the heat exchanger should not either be too 'long', since this will cause pressure drops to become too large.

This section focusses on gains in substation temperature performance to be achieved by optimizing secondary flowrates in space heating systems, and the size of the intersecting heat exchanger in the case of indirect connection of a radiator circuit to the DH network.

When temperature drops in radiator circuits are small or moderate, the possible thermodynamic gains from increasing the size of the heat exchanger may seem very modest. For instance, in Sweden it has been general practice to design and operate radiators for ciculating water cooling in the order of 5 - 15°C at maximum heat load. DH radiator heat exchangers in substations are typically sized to cause a 5°C difference between primary and secondary return temperatures.

With such temperatures it is seen that even drastic increases of heat exchanger size will only cause a few of degrees' improved cooling of primary circuit water. This observation is underlined by the fact that the 5°C temperature differential at design heat load becomes smaller at average load, which occurs more frequently.

However, this applies to changes made under the assumption of constant radiator circuit flowrate. When flowrates are optimized, the situation changes, as illustrated by the example shown in fig. 11.

Here, consequences of different flowrates and different heat exchanger sizes have been calculated, starting out with a reference temperature graph (shown in the right-hand side of the figure), given by a radiator forward temperature of 60°C. For all cases considered the primary forward temperature is kept constant at 100°C. In the reference case the least temperature difference, LTD, between primary and secondary return temperatures is 5°C, referring to a typical maximum heat load case. The heat rate Q transferred in the radiators and in the heat exchanger is kept constant throughout the figure.

In the left-hand side of the figure optima are derived for t_{r1}, that is, minimal primary return temperatures are derived for varying secondary forward temperature t_{rf}. Successively lower t_{r1}-curves are found for increasing heat exchanger size, expressed both i terms of LTD at maximum load (5, 2°C etc) and in terms of relative heat transfer surface area A/A_o (1, 1.30 etc).

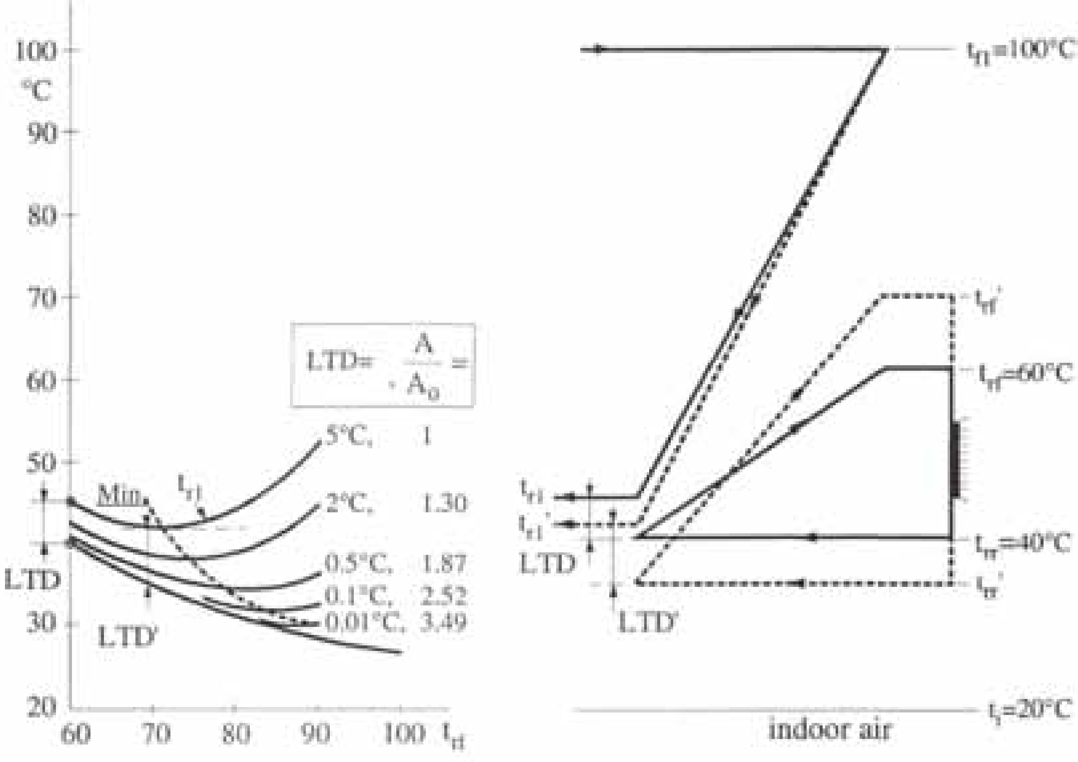


Fig. 11. *Minimum primary return temperature t_{rl} when varying radiator forward temperature t_{rf} for successively bigger heat transfer area A serving indirect connection to a DH network.*

Throughout the entire figure, the overall heat transfer coefficient U of the heat exchanger is kept constant, not only when moving from one heat exchanger size to another, but also along each t_{rl}-curve. I.e., primary and secondary flow areas are thought to be adjusted to keep convective heat transfer coefficients constant in spite of varying flowrate. Thus, the entire diagram refers to true design variations, not to differing load cases for a certain heat exchanger.

The results shown were calculated by keeping LMTD´s constant for both radiators and heat exchangers, a consequence of keeping both Q and U constant.

From the curves it can be seen that with increasing heat exchanger size A, the optimal forward temperature t_{rf} moves towards higher temperature levels. At the same time the optimal LTD decreases, but not as dramatically as LTD does when the flowrate is kept constant at varying A.

The corresponding gains Δt_m in primary water cooling as a function of heat exchanger size are plotted in fig. 12. At the top of this diagram a horizontal line represents the limiting gain achieved with direct connection (or infinitely large heat exchanger) and no forward temperature drop across the heat exchanger, i.e. the t_{fl} = 100°C temperature is transmitted directly into the radiator system.

Looking first the case of m_r = constant, we can see that the law of deminishing returns rapidly manifests itself when the heat exchanger size A is increased.

In contrast, when the flowrate m_r is optimized for each selected heat exchanger size, the temperature gains become 2 - 3 times larger, and although there clearly is an asymptotic value at large A's, the curve does not flatten out as readily as does the curve for constant m_r.

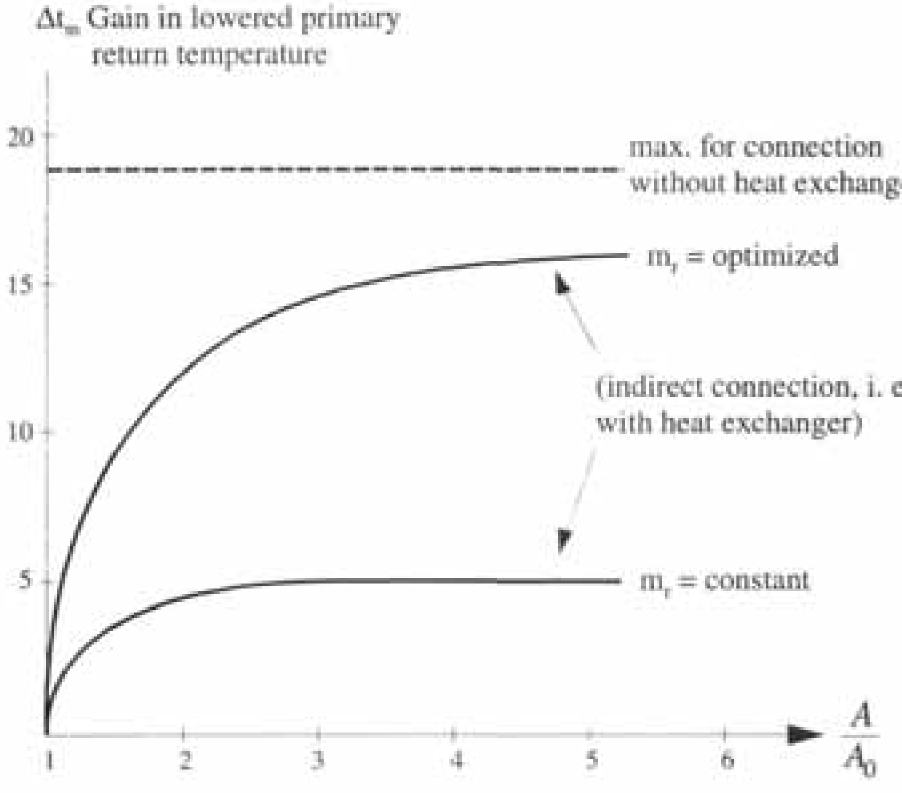


Fig. 12. *Gain in lowered primary DH return water temperature t_{r1} with increasing heat transfer surface area A. Two cases:*

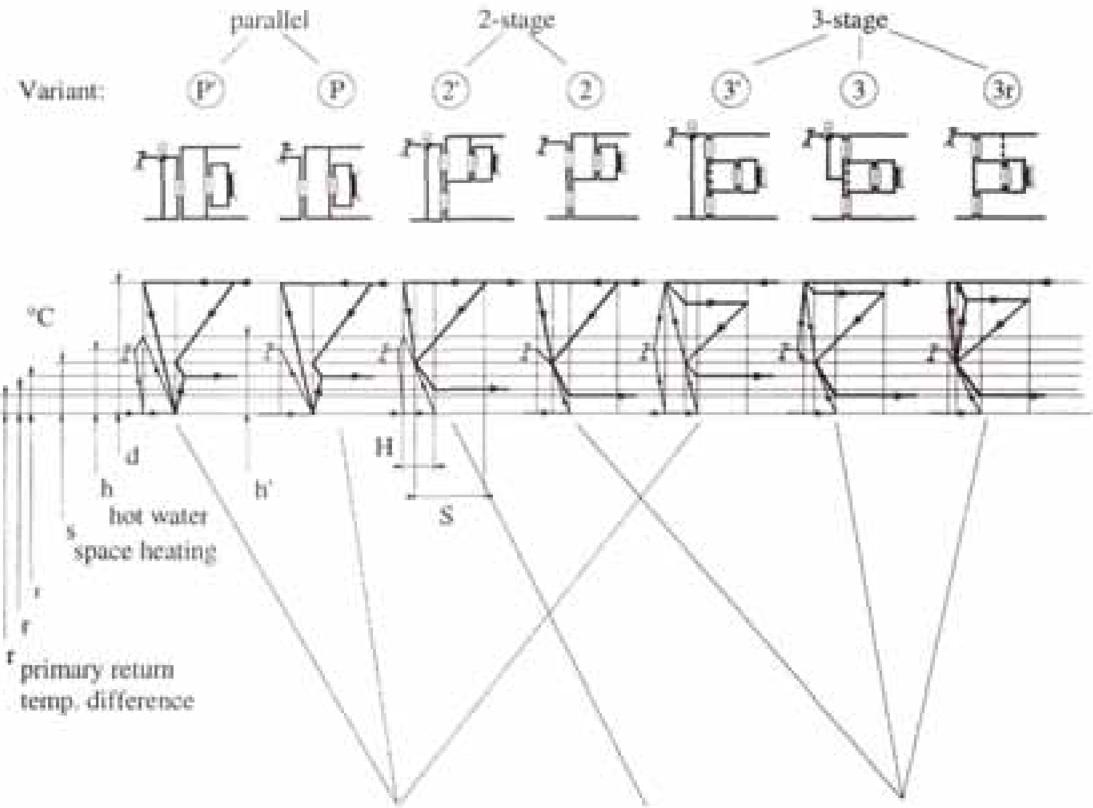
- *Radiator flowrate m_r kept constant*
- *Radiator flowrate m_r optimized to minimize t_{r1} (as in fig. 11).*

This hints at the possibility that when flowrates are optimized, economical optima for heat exchanger sizes may increase. It seems probable that this could result in economical gains of primary water cooling in the order of 10°C for the conditions underlying the numerical example analyzed here. Of course, since calculations are made at maximum heat load, average gains in primary water cooling will become smaller. But even a 5°C improved average cooling would be a large enough potential to justify a more extended analysis of the suggested optimization concept.

9. Fourth key issue: Selection of the best type of connection scheme

In this section we shall consider a number of alternative connection schemes in which heat exchangers are combined in different ways to accomodate for radiator space heating and for provision of domestic hot water. If further types of heat requirements are present, like floor heating, heating of air for laundry drying, ect., many further types of schemes are possible. In the next section we shall thus deal with connection schemes including a heat exchanger for ventilation air heating.

Fig. 13 [ref. 20] gives a systematic overview of a number of possible combinations and provides a thermodynamic comparison of the alternatives.



$$r = \frac{s}{1+\frac{H}{S}\left(1-\frac{s}{d}\right)} \qquad r = \frac{s\left(1+\frac{H}{S}\left(1-\frac{d}{h'}\right)\right)}{1+\frac{H}{S}\left(1-\frac{s}{h'}\right)} \qquad r = \frac{s\left(1+\frac{H}{S}\left(1-\frac{d}{h}\right)\right)}{1+\frac{H}{S}\left(1-\frac{s}{h}\right)}$$

Fig. 13. *Various types of connection schemes (with no hot water recirculation). Temperature graphs and primary return temperature r difference, analytically derived for the idealized limit case of infinitely effective heat exchangers. Source: Frederiksen et. al. UNICHAL 1991.*

Some combinations are more complicated than others. Of course, a solution with more heat exchangers tends to represent a higher installation cost, which in practice must be justified by sufficiently great advantages.

The 7 combinations shown in the figure are grouped into three main types: Parallel, 2-stage, and 3-stage schemes. In all types of 2- and 3-stage schemes a preheater section is provided for the incoming potable water.

Below the connection schemes temperature graphs indicate cooling of primary water flows, heating of potable water, and effects of flow mixings. In the diagrams the abscissa is transferred heat. Assuming that all water flows have a specific heat value which is constant with temperature changes, all curves for heating and cooling become straight lines.

To gain an overview, some simplifications are made in this comparison of connection schemes. To exclude the effect of variations in heat exchanger effectiveness, all heat exchangers are assumed to be infinitely effective, i.e. all LTD´s are 0. Another simplification is that there is no recirculation of domestic hot water.

In the lowest part of the figure, formulae are given for the primary return temperature, r, expressed as temperature level in excess of the incoming potable cold water temperature. The (excess) return temperature from the space heating system is termed s. The desired (excess) domestic hot water temperature is h, while h´ denotes an excess hot water temperature which in some of schemes is lowered to the value h in a mixing device in the hot water circuit. d spans the difference between the incoming primary supply temperature and the potable cold water temperature.

In addition to the temperature differentials, the heat rates S and H for space heating and domestic hot water provision, respectively, are found in the formulae. In general, a bigger H/S value results in a lower return temperature r. For all connection schemes including a preheater section, r becomes 0 at a certain value of H/S.

In terms of low-temperature ranking, the 7 connection schemes fall into 3 groups, each characterized by a certain analytical expression for r. The lowest r-value is achieved by the 3 schemes in the group associated with the far right expression for r.

Two of the general conclusions to be drawn from the comparison are the following:

* Mixing devices are sometimes detrimental from a thermodynamic point of view, in some cases neutral, and sometimes inherently neccessary for a certain connection scheme to avoid excessive domestic hot water temperatures. Thus, fig.13 provides several examples in line with our previous discussion in section 5.

* Preheating of cold water in general contributes to an extra cooling of primary water. All thermodynamically best schemes include preheating.

In Swedish DH practice all variants 2´, 2, 3´, and 3 can be found [ref. 7]. In Finland 2-stage solutions are the common choice for bigger buildings [ref. 11]. Both in Sweden and Finland parallel connections are sometimes chosen instead for smaller multi-family houses.

3-stage schemes are sometimes used in Sweden in geographical zones with soft potable water. When potable water is hard, 3-stage schemes of the 3´or 3-sub-type are not suitable, since all primary water here passes the afterheater, even when there is no hot water load. This causes a high secondary side surface temperature in the afterheater, resulting in precipitation of scale.

With the idealized assumptions underlying the analysis in fig. 13, the best 2- and 3-stage schemes are exactly equal in terms of r-values. When practical, finite sizes of heat exchangers are introduced, a clear-cut general ranking becomes more difficult. Several Scandinavian investigations considered this question. Gummerus in his thesis [ref. 21] ended up in favour of the 3-stage scheme.

If hot water recirculation is provided for in the domestic hot water circuit, and if both supply and return temperatures in the radiator system lie between the cold and hot water temperatures, this tends to favour a thermodynamic preference for 3-stage schemes.

However, unless the potable water is very soft, the risk of scale formation will have to be considered. In this context it is interesting to notice that the last sub-type of 3-stage scheme shown in fig. 4, the one termed 3r, seems to combine the thermodynamic advantages associated with 3-stage, at the same time allowing for a close-down of primary water supply to the afterheater in the event of no hot water demand, thus reducing the risk of scale formation with hard potable water.

From literature, e.g. [refs. 22 & 23] the 3-stage sub-type 3r is known to have been used commonly in Russia (and to some extent in eastern Europe as well), although at quite different operation temperatures, as compared to Scandinavian low temperature practice. Nevertheless it seems worthwhile to consider in this context as well.

Fig. 14 shows a further type of substation connection scheme, which originates from Holland [ref. 24]. It has not been used commonly in international DH practice, but it appears very interesting from a thermodynamical point of view. The scheme may be termed a hybrid of 2- and 3-stage connections. A main characteristic is that both potable water and radiator circuit waters are heated in 2 stages. The dotted line indicates a bypass pipe in which primary water should flow in situations where there is a low radiator heat load and a high hot water demand.

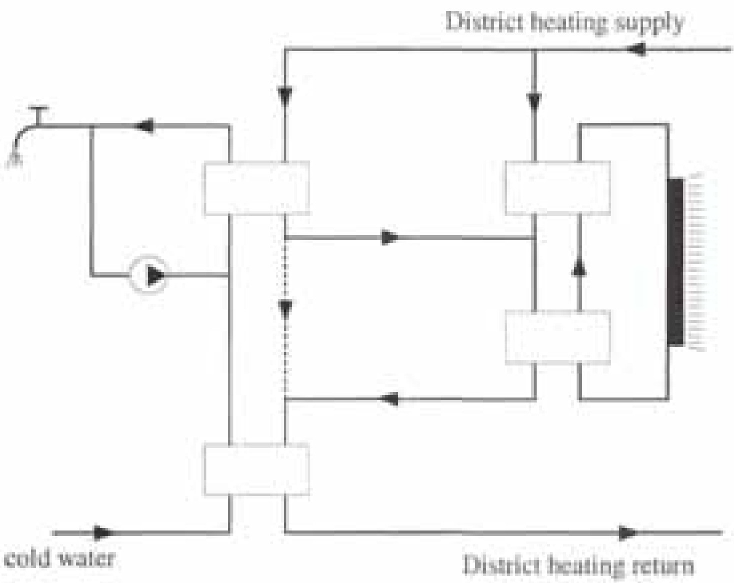


Fig. 14. *Substation in which both potable water and radiator circuit water is heated in 2 stages. Source: Koot 1990.*

The 2-stage heating of radiator water seems to make this solution fit for a low flowrate control strategy for the radiator circuit, provided the heat exchangers are sufficiently large.

From a practical point of view, a solution with 4 heat exchangers may appear expensive. However, if modern plate exchanger technology is utilized, it may be possible to fit it into a compact assemblage.

A full evaluation of different connection schemes will have to consider a number of aspects. One such aspect is dynamic control performance. Since preheaters tend to lessen variations in primary water flowrate at varying hot water demand (higher primary water cooling at big hot water loads), preheating ameliorates fast hot water thermostatic control.

10. Fifth key issue: Low temperature potential of ventilation air

Compared to radiator space heating systems, ventilation systems may be said to be generally low-temperature oriented, for two basic reasons:

1. In the water-to-air heat transfer process, the mean air temperature is generally lower, since the supply air temperature, to be heated from, is lower than the indoor air temperature.

In cold climate zones (like e.g. Sweden) supply air is normally heated in two stages: First in a heat recovery unit which transfers part of heat content of the return air to the supply air, either in a simple heat exchanger or in a heat pump unit. In the second stage, the afterheater, external heat is added to the supply air, e.g. from a DH network.

At low outdoor air temperatures, the air temperature entering the afterheater will typically be some degrees above the freezing point, to avoid freezing in the afterheater. If this temperature is e.g. 6°C, and the indoor air temperature is 22°C, the average air temperature 'felt' by the DH system is (22 - 6) /2 = 8°C below the indoor air temperature.

2. In an air heater the air-side heat transfer is a forced convection process. In contrast, heat transfer on the air-side of radiators is a combination of radiation and natural convection. Both these mechanisms rely on a greater temperature difference to become efficient.

Current practice of incorporating ventilation air heating into DH substations bears much resemblance to solutions developed for applications where there is no significant low-temperature demand. This question can e.g. be discussed in relation to a generic scheme which is found in current Swedish utility recommendations, cf. fig. 15 [ref. 7].

In modern systems, the recommended design supply / return water temperatures for the interface ventilation circuit heat exchanger, separating the intermediate water loop from the DH circuit are:

100 / 35°C on the primary side
60 / 30°C on the secondary side

First it deserves mention that the scheme is low-temperature friendly insofar as the air heater is not equipped with any bypass, allowing hot water to pass the air heater without any cooling.

Still a number of low-temperature orientered modifications deserve consideration:

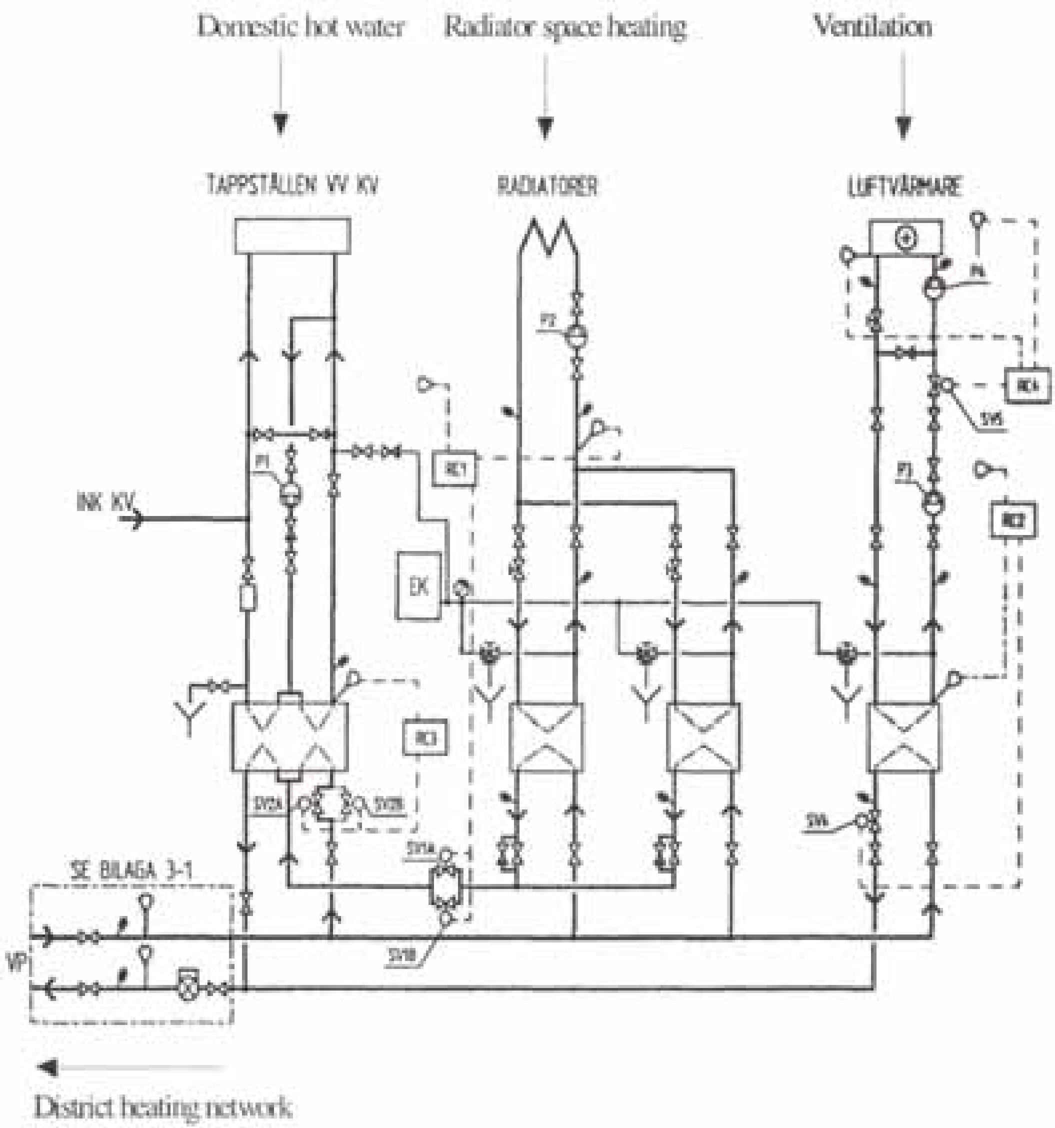


Fig. 15. *Standard connection scheme according to Swedish utility practice, including domestic hot water provision, radiator space heating, and ventilation air heating. Source: Swedish District Heating Association, Stockholm 1994.*

1. Although the secondary side return temperature of 30°C cited above certainly is low, compared to design temperatures for radiator systems, there could be economical potential for further lowerings. It may be that this would require compact air heater units with rather big air-side pressure drops. If so, the consequent bigger air fan power demand will have to be taken into account. Also, higher flow-generated noise levels should be given careful consideration.

2. In an analogy to low-flowrate considerations for radiator circuits, the control curve for the supply temperature to the air heater may be optimized to give the lowest primary side water temperature. In some cases this may imply a high supply temperature to the air heater and thermal air stratification, which will have to be handled in the design of the air heater.

3. In part of the heat load interval, operation with no recirculation of the supply water to the air heater may be considered.

Such an operation is given as an option in German utility recommendations. From a thermodynamic point of view it would be ideal to do away with all mixing. On the other hand, according to some air heating practices, varying water flow control of air heaters is considered problematic, because of the risk of local freezing in the heater. However, if the air heater is equipped with a freeze protection thermostat and an emergency recirculation loop, it should be possible to minimize the freezing risk.

4. Even when a DH utility prefers indirect connection of radiator systems, direct connection of air heaters may be considered. This would eliminate one source of temperature drop in transferring DH heat to secondary circuits.

This option seems particularly close at hand in such buildings where there is a single air heating unit which can be fitted into the basement, the normal place to fit in a substation. In cases where one or more air heaters are located remotely from the substation, long secondary water distribution pipes may be needed. Safety demands on such a circuit probably differ substantionally from one country to another, and will depend on the pressure and temperature levels of the DH network.

5. Traditionally, air heaters are connected in parallel with radiator circuits in DH substations, as is the case in the scheme of fig. 15. When air heaters are designed for low-temperature operation, a higher degree of primary water cooling can be achieved with a lower-temperature end connection, as shown in fig. 16.

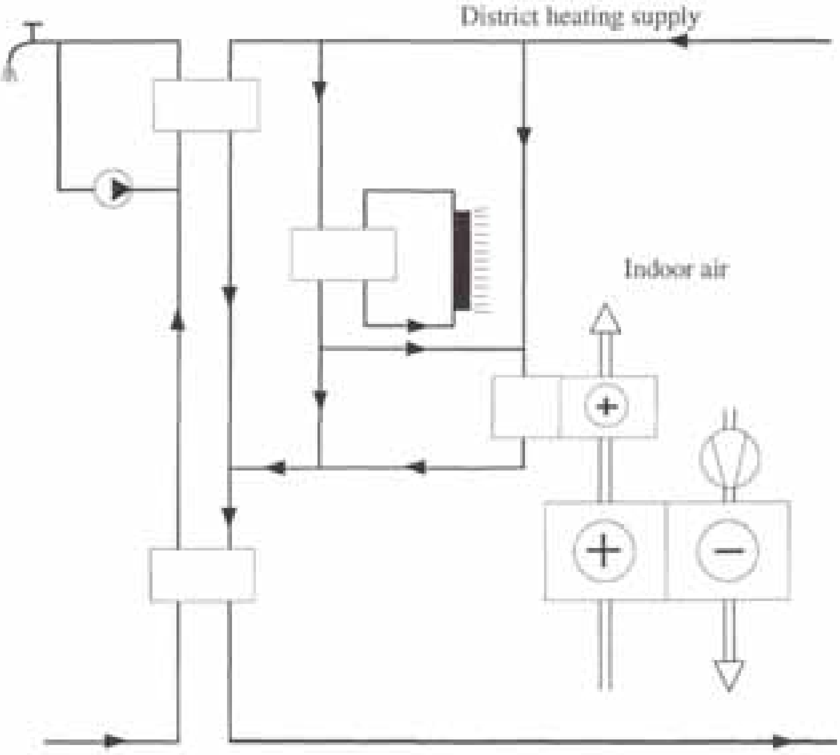


Fig. 16. *Example of low-temperature connection of buildings with combined domestic hot water provision, radiator space heating, and heating of ventilation air (after preheating in heat recovery unit).*

6. When air heaters are designed and connected for decidedly low-temperature operation, it may be considered to shift over as big a proportion as possible of the total heat delivery to the ventilation system, cutting back on radiator heating. Due to lower radiator water temperatures, an additional low-temperature effect may be achieved in this way, apart from the effect of generally lower temperatures in the ventilation system.

It may be that such a shift will cause an increased heat demand for the building, due to higher air exchange. Only a closer analysis can tell whether the Second Law benefits from lower network temperatures will outbalance such a higher energy demand.

11. Literature

[1] *Hjorthol, E. & Ulseth, R.:* Consumer Heating System Simulation.
SINTEF, the Norwegien Institute of Technology, Trondheim
IEA District Heating Project Report published by
NOVEM, Sittard, The Netherlands, 1992.

[2] *Personal communication:* Malmö Energy Utility.
Malmö, Sweden.

[3] *Horlock, J.H.:* Cogeneration: Combined Heat and Power, Thermodynamics and Economics. Pergamon Press, U.K. 1987.

[4] *Volla, R.:* Gains From Increased Temperature Difference in District Heating Plants (in Norwegian), Report from SINTEF, the Norwegian Institute of Technology, Trondheim 1994.

[5] *Stene, J. (editor):* Basic Heat Pump Technology (in Norwegian) Textbook, the Norwegian Institute of Technology, Trondheim 1990.

[6] *Sleiman, A.H., Spurr, M., Rydaker, A. Karnitz, M.:* Guideline for Converting Building Heating Systems for Hot Water District Heating. District Heating St. Paul, Inc. (USA) under contract to Prince Edward Island Energy Corp. (Canada), IEA District Heating Project Report published by NOVEM, Sittard, the Netherlands, 1990.

[7] *Swedish District Heating Association:* District Heating Substations (in Swedish). Technical Guideline published 1994 in Stockholm, Sweden.

[8] *Andersen, N.B., Lawaetz, H.:* Energy Savings Achieved by Low Temperature Operation of Danish District Heating Plants (in Danish). Dansk Teknologisk Institut, Taastrup, Denmark 1992.

[9] *Statens Planverk:* Electric Resistance Heating of Single-Family Dwellings, Low Temperature heating of Buildings and Further Items (in Swedish). Statens Planverks Författningssamling 1982:3 Statute, Stockholm, Sweden 1982.

[10] *Boverket:* Byggregler 94 (in Swedish). Boverkets Författningssamling BFS 1993:57 Statute, Stockholm, Sweden 1993.

[11] *Finnish District Heating Association:* District Heating in Buildings. Summary of Publication (English version) K1/1992 Helsinki, Finland 1992.

[12] *Arbeitsgemeinschaft Fernwärme:* Fernwärmeversorgung aus Heizwerken. 2. Auflage, Frankfurt/Main, Germany 1981.

[13] *Arbeitsgemeinschaft Fernwärme:* Technische Anschlussbedingungen Heizwasser. Information für Mitglieder Nr. 62 Frankfurt/Main, Germany 1994.

[14] *Brundrett, G.W.:* Legionella and Building Services. Butterworth-Heinemann Ltd, Oxford, U.K. 1992.

[15] *Munser, H.:* Verfahrenstechnische Lösung zur Erhöhung der Betriebssicherheit bei der direkten Einspeisung in Fernwärmesystemen.
Energietechnik, 38. Jg., Heft 10, 1988.

[16] *Gustafson, B.-A.:* The Pressure Separator.
Report from the Department of Thermo- and Fluid Dynamics, Chalmers University of Technology, Gothenburg, Sweden 1995.

[17] *Schelosky, H.U.:* Möglichkeiten der Erzielung niedriger Rücklauftemperaturen in Fernwärmenetzen.
Fernwärme international - FWI, Jg. 9 (1980), Heft 4, pp 275-280.

[18] *Frederiksen, S. & Wollerstrand, J.:* Performance of District Heating House Stations in Altered Operational Modes.
Paper to the 23. UNICHAL-Congress in Berlin, Germany 1987.

[19] *de Wit, J. & Paulsen, O.:* Control of Substations in Single-Family Dwellings Based on Indirect Connection to District Heating Networks.
Report from Teknologisk Institut, Taastrup, Denmark 1989.

[20] *Frederiksen, S., Nikolic, D. & Wollerstrand, J.:* District Heating House Stations for Optimum Operation.
Paper to the 25. UNICHAL-Congress in Budapest, Hungary 1991.

[21] *Gummérus, P.:* Analysis of Conventional District Heating Substations.
Dissertation, Chalmers University of Technology, Gothenburg, Sweden 1989.

[22] *Zinger, N.M., Belova, N.P. & Burd, A.L.:* Comparison of Different Systems of Connecting Hot Water Supply Plants to the Heating Network.
Thermal Engineering, 26 (2), 1979, pp 69-73.

[23] *Valasek, J.:* Massnahmen zur Senkung der Rücklauftemperaturen durch Übergabestationen in der CSFR.
Energietechnik, 41. Jg, Heft 1, 1991, pp 7-10.

[24] *Koot, M.H.M.:* Return Temperature Control and Heat Exchanger Configurations.
Report from the Measurement & Systems Department, Twente University, published by NOVEM, Sittard, the Netherlands 1990.

[25] *Nielsen, Ø.:* Combined Series-Parallel Connection: System for District Heating Adapted Connection (in Norwegian).
Norsk VVS 4/1994, pp. 29-31.

[26] *Ahonen, M., Kosonen, R. & Kekkonen, V.:* The New Low Temperature HVAC-System That Reduces Combined Heat and Power Plants' Costs.
Paper to the 5th International Symposium on Automation of District Heating Systems,Nordic Energy Research Programme, Helsinki, Finland 1995.

Part III

VALIDATION OF THE HEATING COIL MODEL USED IN THE CONSUMER HEATING SYSTEM SIMULATION PROGRAM

Allan Johnson Robert Besant

EnEff Technologies Inc.

May 1995

Table of Contents

SUMMARY

This report is concerned with the design and performance of typical air-heating coils used in buildings supplied by a district heating system.

First the method of analysis and design for typical finned-tube heat exchanger coils is presented, and measured data is compared to simulation results. Uncertainties in the heat rate of less than 5% are expected.

Second the existing CHESS program heating coil model is reviewed and compared with the validated simulation model coil simulation results. If the CHESS model is modified to model more accurately the heat exchange processes in a typical coil, it shows good agreement with the U of S simulation program. The current assumed values of heat transfer coefficients, air side heat transfer area, and air volume are not the same as calculated using the U of S simulation model. A simulation program has been produced that will calculate the input values required in the CHESS model for a given heating coil design. It is shown that if corrected values are not introduced as input data for the CHESS program, errors of up to 50% can occur in the overall heat transfer coefficient.

Finally, an optimized heating coil design based on minimizing the Life-Cycle Cost of the heating coil has been determined. This design has been compared to a conventional heating coil design to show the opportunities available using an optimal design. This optimized coil design resulted in a coil selection that was 65% more expensive, but the system Life-Cycle costs were more than 4 times smaller. The main benefit of using a better designed heating coil is that liquid flow rates can be reduced which is beneficial to the building owner in lower pumping costs, and beneficial to the District Heating utility which would also see the benefit of lower pumping rates.

1. Introduction

The purpose of this report is to validate the heating coil model in the CHESS program developed by SINTEF for the purpose of dynamic simulation of building HVAC systems connected to a district heating system. A simple 3 cell model using basic energy equations is used in the CHESS program to simulate a heating coil.

The problems involved in the incorporation of a heating coil into the dynamic program is that the input values (flow rates, temperatures, fluid and heating coil mass and volumes) must be easy to specify. Also the mathematical description of the heating coil must not be too complex in the CHESS program.

This report validates the use of the heating coil model used in the CHESS program by comparing results to a steady state simulation model developed at the University of Saskatchewan (U of S). There are four distinct steps in the validation process.

1. The simulation program developed at the U of S is compared to monitored results to show that the model used is accurate when compared to actual finned, staggered-tube heat exchangers.

2. A conventional designed heating coil specified by a manufacturer to meet the requirements of the simulation example shown in Appendix A of the CHESS manual is used as the basis for comparison of the input values.

3. The CHESS program heating coil model is compared to the simulation results from the University of Saskatchewan using input values calculated from the U of S program.

4. An optimized heating coil design is found for minimum Life-Cycle Cost at the same design conditions as used by the manufacturer for a conventional heating coil design. This optimum heating coil design should provide the most economical design for the building owner.

2. Methodology

2.1 Steady State Simulation Program for Heat Exchanger Coils

The procedure used in the heat exchanger coil simulation program at the University of Saskatchewan involves solving three simultaneous energy equations that describe the heat transfer within the heat exchanger core. The three energy balance equations used in the simulation are:

$$q = \varepsilon C_{min} (T_{l,in} - T_{a,in}) \quad (1)$$
$$q = C_a (T_{a,out} - T_{a,in}) \quad (2)$$
$$q = C_l (T_{l,in} - T_{l,out}) \quad (3)$$

In this analysis, the temperatures,

$T_{l,in}$ = the liquid temperature entering the coil, and

$T_{a,in}$ = the air temperature entering the coil

are known variables.

The variables,

C_a = air capacitance rate

C_l = liquid capacitance rate

C_{min} = minimum of air and liquid capacitance rate

ε = the coil effectiveness,
are calculated, and considered known variables.

This leaves the variables,

q = the heat transfer rate,
$T_{a,out}$ = the outlet air temperature, and
$T_{L,out}$ = the outlet liquid temperature,

as the unknowns that are solved in the simulation model.

The air and liquid capacitance rates are determined knowing the flow rates, as well as the density and specific heats of both fluids. The coil effectiveness, ε, is determined using an ε-NTU relationship such as:

$$\varepsilon = \frac{1-\exp[-NTU(1-C_r)]}{1-C_r\exp[-NTU(1-C_r)]} \tag{4}$$

where C_r is the capacitance rate ratio, and the number of transfer units, NTU is given by:

$$NTU = UA/C_{min} \tag{5}$$

This ε-NTU relationship is accurate for 4 or more tube passes, and $C_r < 1$.

The overall thermal conductance of the heat exchanger, UA, is calculated from:

$$\frac{1}{UA} = \frac{1}{(hA)_{liquid}} + \frac{\ln(D_o/D_i)}{2\pi kL} + R_c + \frac{1}{(\eta_o hA)_{air}} \tag{6}$$

where

h = the heat transfer coefficient of the liquid or the air
A = the surface area on either the liquid or air side
η_o = the overall fin efficiency (Methodology given by Schmidt [1])
R_c = the collar resistance (contact resistance between fins and tubes)

and the second term (on the right hand side of the equation) accounts for the wall resistance where,

D_o = the outer tube diameter
D_i = the inner tube diameter
k = the thermal conductivity of tube material
L = the total length of tubing in the heat exchanger.

2.2 Geometric and Property Parameters Used in the Chess Program

In the CHESS program, certain geometric and fluid parameters are required in the modeling of the heating coil. These parameters include:

liquid density
air density
liquid heat capacity
air heat capacity
liquid side heat transfer surface area
air side heat transfer surface area
liquid side heat transfer coefficient
air side heat transfer coefficient
liquid volume in heat exchanger
air volume in heat exchanger
tube volume
fin volume

Currently many of the parameters used in the CHESS program are either assumed values or values taken from an "average" heating coil. All of these variables can be accurately calculated using the known geometry of the heat coil, and the properties of the air and liquid. A computer program has been provided that simulates a given heat exchanger coil and provides the parameters required in the CHESS program as well as the simulated heat transfer rate and outlet fluid temperatures.

Figure 1 and Figure 2 describe the geometric variables associated with a heating coil. The geometric variables are for a staggered tube arrangement common for many heating coils.

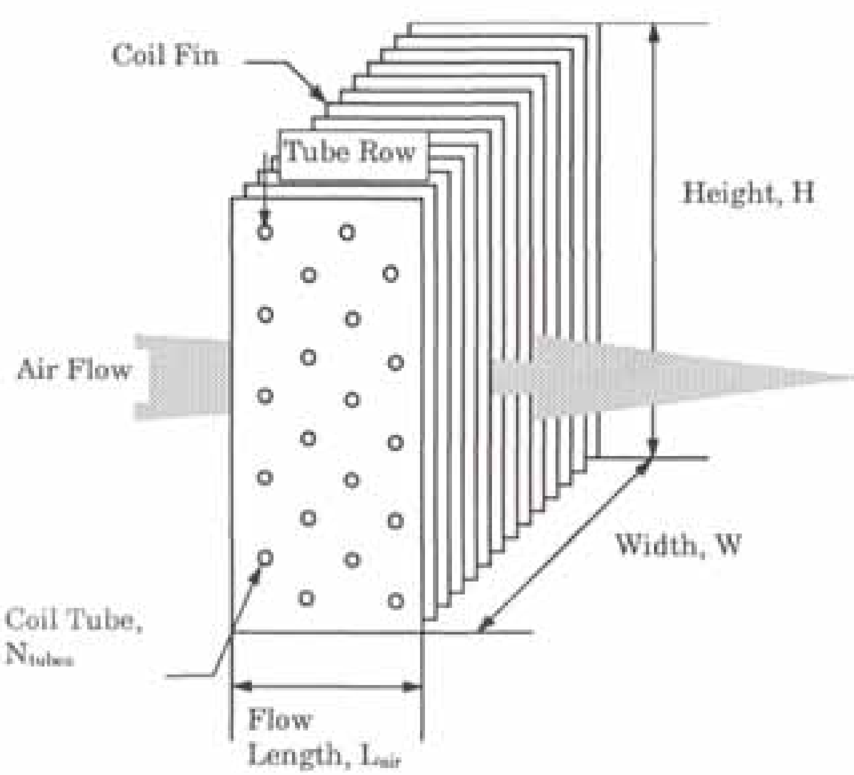


***Figure 1:** Staggered Expanded Tube Heat Exchanger Coil*

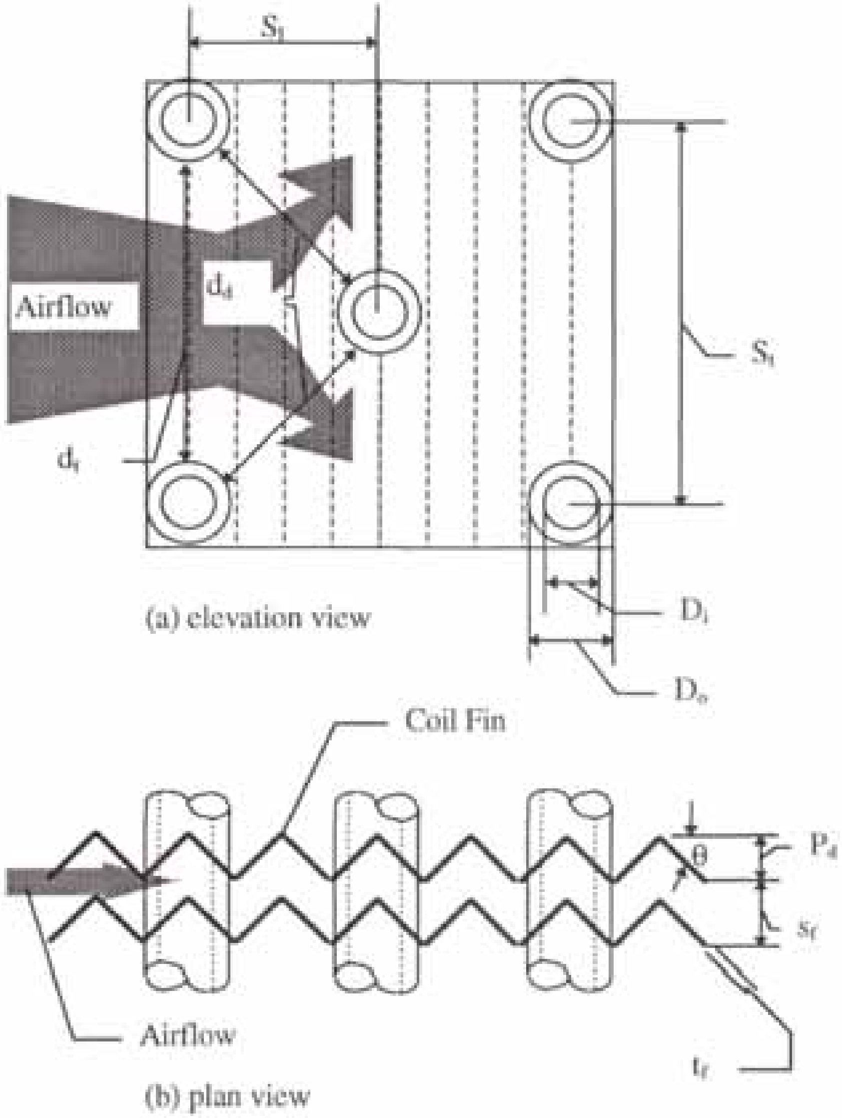


Figure 2: *Geometric Parameters of Fins and Tubes for a Staggered-Tube Compact Heat Exchanger*

2.2.1 Fluid Property Equations

In the CHESS program, the density and specific heats for the liquid are for water only. Usually ethylene glycol is used to provide freeze or burst protection in the heating coil. Also the properties are temperature dependent, and this should be considered in the energy analysis. For this reason, the following equations have been provided to calculate the liquid properties.

The density of ethylene glycol solution in lb_m/ft^3 is given by:

$$\rho_{fl} = 62.316944 \times (A + (B/10000)\times T - 1\times10^{-6}\times C\times T^2) \qquad (6)$$

where
$A = 0.9996 + 0.0023925\times(vol\%) - 8.790645\times10^{-6}\times(vol\%)^2$
$B = 1.346693 - 0.1331125\times(vol\%) + 0.001049024\times(vol\%)^2$
$C = 2.0795536 - 0.07934069\times(vol\%) + 0.0008072\times(vol\%)^2$

and T is the bulk mean temperature of the liquid in °F. Vol% is the percent glycol in water by volume.

The specific heat of the liquid, C_p, in Btu/lb-°F is given by:

$$C_p = A + B\times(0.0001)\times T + C\times1\times10^{-7}(T^2) \qquad (7)$$

where
$A = 1.008581 - 0.0039757\times(vol\%) - 1.26235\times10^{-5}\times(vol\%)^2$
$B = -1.76776 + 0.1975521\times(vol\%) - 0.001418548\times(vol\%)^2$
$C = 8.21/(1+440.34\times(vol\%))$

and T is the bulk mean temperature of the liquid in °F. Vol% is the percent glycol in water by volume.

The liquid properties in Equations 6 and 7 are given in Imperial units. To convert temperature from Metric units to Imperial units use the equation:

$$T\ (°F) = T(°C) \times 1.8 + 32 \qquad (8)$$

To convert the density from Imperial units to Metric units use the equation:

$$\rho_{fl}\ (kg/m^3) = \rho_{fl}\ (lb_m/ft^3) \times 16 \qquad (9)$$

To convert the specific heat from Imperial units to Metric units use the equation:

$$C_p\ (kJ/kg°C) = C_p\ (Btu/lb\text{-}°F) \times 4.19 \qquad (10)$$

The density of moist air in kg/m^3 is given by the equation,

$$\rho = \frac{P}{R\cdot T\cdot(1+1.6078\cdot W)} \qquad (11)$$

where P is the absolute static pressure in Pa, T is the absolute temperature in °K, R is the specific gas constant of air (= 287.1 J/kg°K), and W is the air humidity ratio in kg/kg. The specific heat, C_p, of air in kJ/kg°C is given by,

$$C_p = 4.19\times(0.240 + 0.444(W)) \tag{12}$$

2.2.2 Geometric Parameters Required in CHESS Program

Liquid Volume

$$V_{liquid} = \pi\, D_i^2/4 \cdot N_{tubes} \cdot W \tag{13}$$

where,

D_i = the inner tube diameter
W = the coil width
N_{tubes} = the number of tubes in the heat exchanger coil.

$$N_{tubes} = N_{vr} \cdot N_r \tag{14}$$

where,

N_{vr} = the number of vertical tube rows in the heat exchanger coil
N_r = the number of tube rows (horizontal) in the heat exchanger coil.

$$N_{vr} = H/S_t - 1/2 \tag{15}$$

where,

H = the heat exchanger coil height, and
S_t = the transverse tube spacing (vertical).

Liquid Side Heat Transfer Area

$$A_{liquid} = N_{tubes} \cdot \pi \cdot Di \cdot W \tag{16}$$

Air Side Heat Transfer Area

The primary air side heat transfer area, A_p, is the outside area of the tubes minus the area covered by the fins and is given by,

$$A_p = N_{tubes}\pi D_o W\left(1 - \frac{s_f t_f}{\cos\theta}\right) \tag{17}$$

where D_o is the outside tube diameter, s_f is the fin spacing, and t_f is the fin thickness (see Figure 2). The header plate surface is not included in this calculation because there is no direct heat transfer from the air to the liquid through the header in the coil. Cos θ is the fin wave angle and is calculated using:

$$\cos\theta = \frac{X_f}{\sqrt{(X_f^2 + P_d^2)}} \tag{18}$$

where
$$X_f = \frac{S_l}{2N_p} \tag{19}$$

and P_d is the depth of fin wave, S_l is the longitudinal (horizontal) tube spacing, and N_p is the number of fin wave cycles between each longitudinal tube row as shown in Figure 2 for the case of two cycles per tube row. In the simulation program provided, it is assumed that N_p is 2.

The secondary air side heat transfer surface is the fin surface area, A_f, and it is given by,

$$A_f = \frac{2s_f W}{\cos\theta}(HL_{air} - N_{tubes}\pi \, D_o^2/4 + t_f H) \tag{20}$$

where

$L_{air} = N_r \cdot S_l$ = the air side flow length

This includes the area of the flat surface of the fins plus the leading and trailing edges of the fins. The total air side heat transfer area, A, is the sum of the primary and secondary air side heat transfer areas,

$$A = A_p + A_f \tag{21}$$

Tube Volume

$$V_{tube} = N_{tubes} \cdot \pi \cdot W(D_o^2 - D_i^2)/4 \tag{22}$$

The tube volume can be used to calculate the mass of the tubes using the density of copper.

Fin Volume

$$V_{fin} = \frac{s_f W t_f}{\cos\theta}(N_r S_l H - N_{tubes}\pi \frac{D_o^2}{4}\cos\theta) \tag{23}$$

The fin volume can be used to calculate the mass of fins by using the density of aluminum. From this, the heat storage term for the shell wall can be calculated using the mass of the tubes, and the mass of the fins.

Air Volume

$$V_{air} = L_{air} \cdot H \cdot W - (V_{fin} + V_{tube} + V_{liquid}) \tag{24}$$

The air volume is calculated using the total volume of the heat exchanger, and subtracting the volumes of the fins, tubes and liquid.

Heat Transfer Coefficients

The heat transfer coefficients are determined using correlations for both liquid and air sides. The equations for the heat transfer coefficients are probably too complex to be incorporated into the CHESS program, so it is hoped that the simulation program provided can be used to calculate nominal heat transfer coefficients, and for each particular heating coil an exponent determined to account for varying flow rates.

The liquid side heat transfer coefficient is calculated using the Gnielinski [2] correlation to calculate Nusselt numbers for Reynolds numbers above 2400. For a Reynolds number below 1800, the Nusselt number is 4.364. In the simulation program, linear interpolation is used in the transitional region (i.e. 2400 > Reynolds >1800).

The air side heat transfer coefficients are determined using correlations by Gray and Webb [3] for plain fins. For wavy fins, a correlation by Webb [4] that fits data by Beecher and Fagan [5] is used.

In the CHESS program, the collar resistance, tube wall resistance, and overall fin efficiency are not accounted for. It is suggested that these terms be incorporated into the air side heat transfer coefficient by:

$$\frac{1}{h_{air-effective}} = (R_{collar} + R_{wall})A_{air} + \frac{1}{h_{air}\eta_{fin}} \tag{25}$$

The collar resistance, wall resistance, air side heat transfer area, and fin efficiency are all computed in the simulation program provided. Significant errors will result if these terms are omitted, and only the air side heat transfer coefficient is used.

An equation of the same form as used in the CHESS program can be used to account for changes in the liquid or air flow rates. This equation is (page 20 of the manual):

$$h = h_{nominal}\left(\frac{Q}{Q_{nominal}}\right)^{a} \tag{26}$$

where,

Q is the fluid flow rate,

$Q_{nominal}$ is the fluid flow rate corresponding to the nominal heat transfer coefficient, and, n is the exponent correlating the change in heat transfer coefficient with the change in fluid flow rate.

In practice, an accurate simulation program should be used for each particular heat exchanger coil to determine the exponent, n, but if this is not practical it is probably appropriate to use 0.8 for n in the case of the liquid flow. Generally, the liquid flow rate in a heating coil is kept above a Reynolds number of 5000, so an exponent of 0.8 is accurate when properties are constant. Under part load conditions, the liquid flow rate is reduced, and the heat transfer coefficient changes more proportional to the change in the liquid flow rate, so the exponent, n, is higher than 0.8.

The Reynolds number on the air side is generally in the laminar range with respect to plane surface fins, but the wavy fins and changing air flow direction around the tubes make the flow very complex. Using 0.8 for the exponent, n, will result in calculated effective air heat transfer coefficients that are too low. As is shown in the results section, for a typical heat exchanger coil, the exponent should be closer to 0.3 for the effective air side heat transfer coefficient. The effective air side heat transfer coefficient is less sensitive to changes in air flow rate because it includes contact (collar) and wall resistances which are constant.

2.3 Validation of the CHESS Program

A 3 cell model using the same equations as the CHESS program was set up to compare our simulation program to the CHESS program. A direct comparison could not be made using the CHESS program because of the use of the shape factors "M" and "K" in the program. The shape factor, "M", is used to calculate the outside surface area based on the inside surface area. "K" is used to calculate the air volume based on the volume of the tubes and lamellas. The comparison was made at steady state, so the dynamic equations (page 17 and 18 of the CHESS manual) used in the CHESS program for each cell reduce to:

Tube Liquid:

$$0 = C_l (T_{l(n-1)} - T_{l(n)}) - h_{liquid} A_{liquid} (\frac{T_{l(n-1)} + T_{l(n)}}{2} - T_{t(n)}) \qquad (27)$$

Shell Wall:

$$0 = h_{liquid} A_{liquid} (\frac{T_{l(n-1)} + T_{l(n)}}{2} - T_{t(n)}) - h_{air-effective} A_{air} (T_{t(n)} - \frac{T_{a(n-1)} + T_{a(n)}}{2}) \qquad (28)$$

Air Side:

$$0 = C_a (T_{a(n)} - T_{a(n-1)}) - h_{air-effective} A_{air} (T_{s(n)} - \frac{T_{a(n-1)} + T_{a(n)}}{2}) \qquad (29)$$

where, n, is the cell number, and T_s is the shell temperature. It should be noted that the average of the inlet and outlet fluid temperatures was used when calculating the convective heat transfer from the liquid to the shell, and from the shell to the air. It is unclear whether this was done in the CHESS program or if only the outlet temperature from each cell was used as it appears in the program manual.

2.4 Optimization of Heating Coils

The optimization of the heating coil is similar to the optimization of run-around systems as described in Bennett et al [6] and Johnson et al [7]. In general, the expected Life-Cycle cost is minimized by varying geometric parameters of the heat exchanger coil, and varying the liquid flow rate and inlet liquid temperature. There are many constraints used in the optimization procedure. One constraint is the heat rate must be sufficient at the cold design temperature. Other constraints are mainly geometric constraints of the manufacturer. Some of the constraints imposed by the manufacturer are tube diameter, coil height and width, tube spacing, and fin series.

The Life-Cycle cost for the heating coil was approximated by:

$$LCC = C_{coil} + C_{floor} + C_{pump} + C_{fan} + C_{DH} \qquad (30)$$

where,

LCC is the Life-Cycle cost that is minimized in the optimization procedure,

C_{coil} is the capital cost of the heating coil approximated by correlations that approximate the cost of heat exchanger coils manufactured by Engineered Air Inc. Calgary, Alberta. These costs were last updated in 1991, so the actual costs may have changed, but relative costs between coils should be the same.

C_{floor} is the capital cost associated with the floor area of the hot deck. It is assumed that the hot deck width is governed by the coil width, and that the deck length is generally 10 times the width. It was assumed that the cost of the floor space would be \$300/m^2, although this may vary from \$200 - \$400/m^2. Thus the floor cost is approximated by \$3000 W^2.

The pumping cost C_{pump} is the operating cost of the circulating pump during the heating season. Pressure drops across both the heating coil, and the plate heat exchanger at the connection between the district heating loop and the building loop are considered in calculating the cost. It is assumed the pumping efficiency is about 55%, the pump motor efficiency 85%, and the pump has a power factor of 0.9.

The fan cost C_{fan} is the operating cost caused by the pressure drop across the heating coil. Typical to larger HVAC fans, a fan efficiency of 70%, motor efficiency of 90%, and power factor of 0.9 are used in calculating the operating cost.

The final operating cost, C_{DH}, is the cost passed on to the building owner by the district heating company for liquid pumping costs. In the absence of better data, it is assumed that a coefficient of performance (COP) of 100 can be used. This COP relates supply heat rate to pumping power. For a required heating coil heat rate, q_c, the pumping power, P, can be considered

$$P = \frac{q_c}{C.O.P} \tag{31}$$

For an optimal coil design the pumping power of hot water should decrease in proportion to the cube of the fluid flow rate. Because the heat rate is fixed by the air heating load of the building, the flow rate will change proportional to the temperature drop across the coil. District heating pumping cost savings resulting from running the coil with a larger temperature drop (and smaller fluid flow rate) are calculated by comparing the temperature drop across a conventionally designed coil, and the optimal coil. It is assumed that the savings resulting from larger temperature drops across the coil cannot exceed 90% of the initial cost of pumping for the district heating utility.

The electricity costs used in the analysis were the commercial building rates in Saskatoon. These rates are $0.0837/kWh for consumption, and $12.53/kW as a demand charge. It was assumed that electrical rates are increasing at the same rate as inflation.

The cost benefit of eliminating the boiler and the space it would occupy in a building supplied by a district heating system were not considered in the optimization. This is because these savings would be fixed benefits, independent of the size and design of the air heating coil.

3. Results

3.1 Validation of the Steady State Heat Exchanger Program

Monitored results from a run-around heat recovery system was used to validate the single coil heat exchanger model. The monitored data was for a heat exchanger with the following coil parameters:

Table 1: Physical Parameters for the Supply Coil on a Monitored Run-Around System Coil

Parameter	Material	Supply Coil	
Airflow, L/s (cfm)		14540	(30800)
Liquid Flowrate, L/s (US gpm)	37% Ethylene-glycol and Water Mixture by Weight	10.2	(161)
Coil Width, cm (in)		335	(132)
Coil Height, cm (in)		198	(78)
Number of Tube Passes		4	
Number of Tube Rows		4	
Tube Outer Diameter, mm (in)	Copper Tubing	16	(0.625)
Tube Inner Diameter, mm (in)		15	(0.59)
Tube Transverse Spacing, mm (in)		38	(1.5)
Tube Longitudinal Spacing, mm (in)		33	(1.3)
Fin Thickness, mm (in)	Aluminum Fin Stock	0.14	(.0055)
Fin Wave Depth, mm (in)		2.1	(0.082)
Fin Spacing, fins/mm (fins/in)		0.472	(12.0)

The liquid flow rate, inlet air temperature, humidity, and pressure, and inlet liquid temperatures were monitored for a wide range of operating conditions. These values were used as the inputs for the heat exchanger simulation. The output results from the program were the heat rates, outlet air temperature, and outlet liquid temperature. These results were compared to the measured outlet air temperature, outlet liquid temperature, and calculated heat rates. The heat rates were calculated using the measured inlet and outlet temperatures, and a calculated heat capacity rate for each fluid.

Figure 3 shows a comparison of the simulated heat rate, and the monitored heat rates. The simulated results are within ±5% of the monitored heat rates. On the air side, the maximum uncertainties in the data collected were due to uncertainties in the air flow rate. On the liquid side, the maximum uncertainties are likely the liquid property values. For example, the reason that the liquid heat rates may be slightly higher than the air side is because the calculated density, or calculated specific heat are slightly higher.

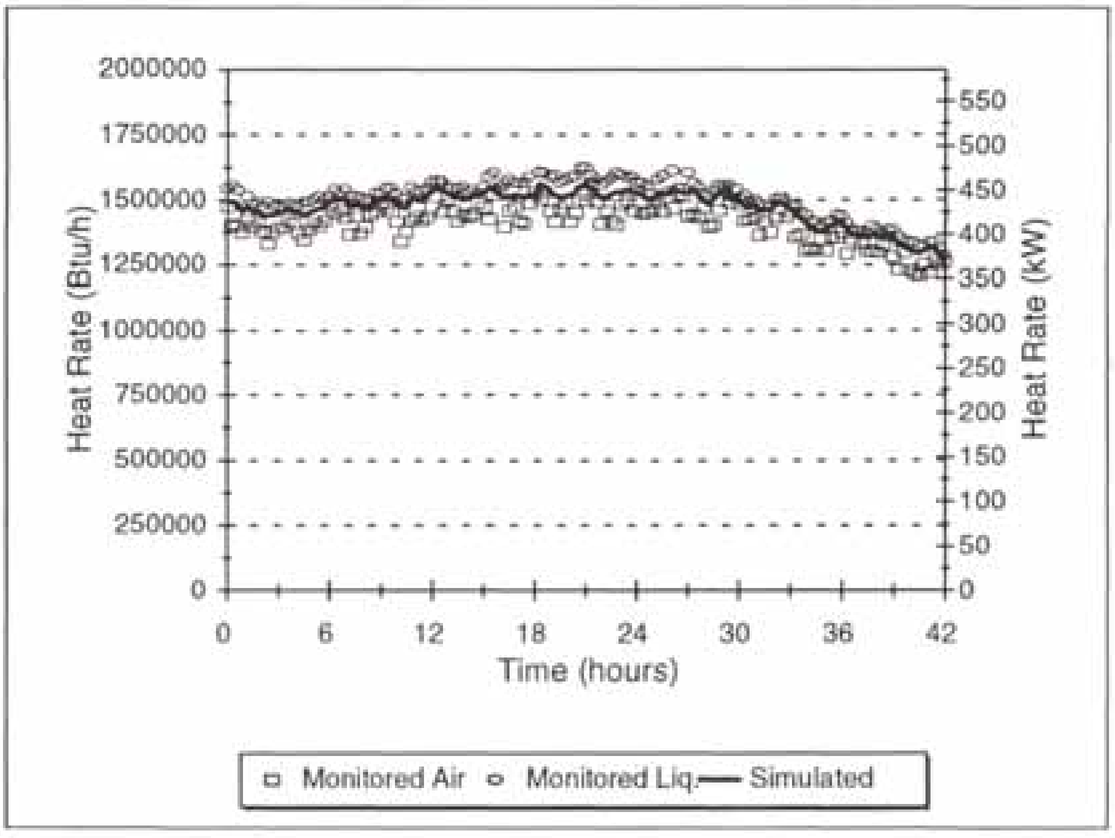


Figure 3: *Comparison of Measured Heat Rates for a Monitored Run-Around Coil compared to Simulated Heat Rates from the U of S Simulation Program.*

Transient effects were never found to be significant on any of the test results from several heating and cooling coils. That is, although operating conditions were changing continuously, the transient term would always be smaller that the uncertainties in the measured and calculated data. For this reason, the theoretical model does not include transient terms for these validation studies.

Figure 4 compares the air outlet temperature from the simulated heat exchanger to the monitored results for the same typical data set as Figure 3. The outlet air temperatures from the simulation are almost exactly what was seen in the monitored results.

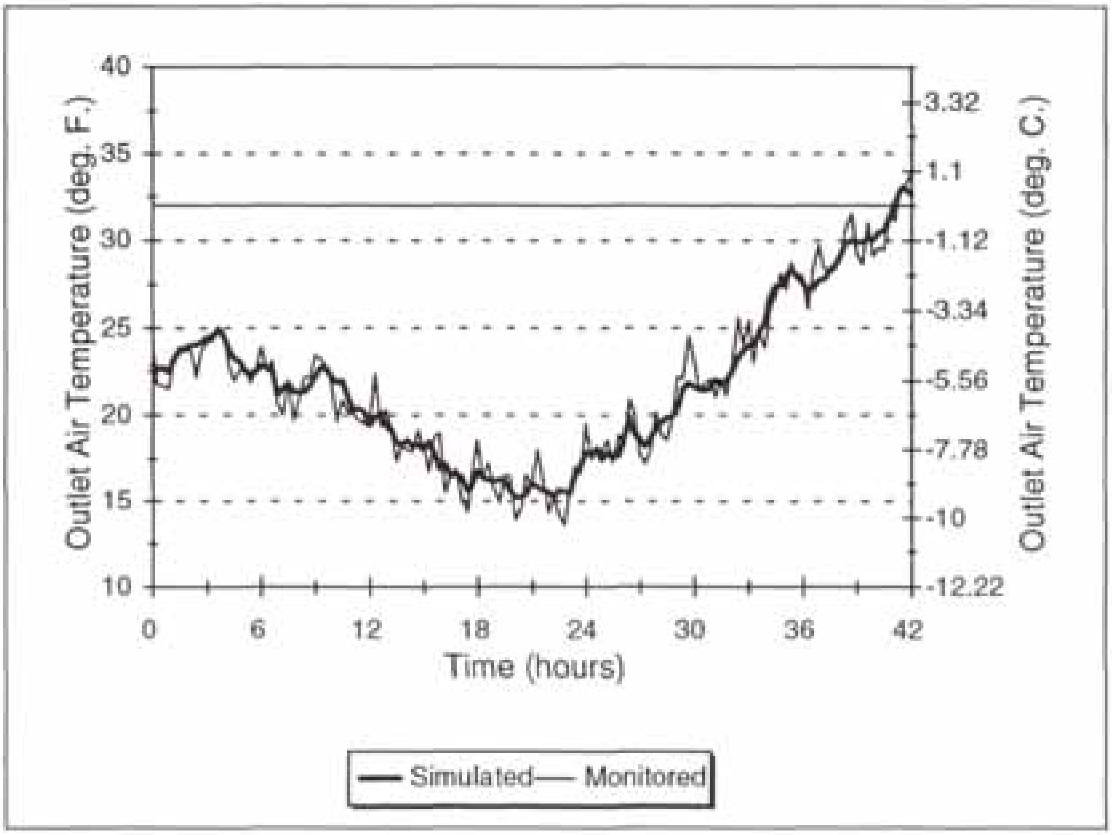


Figure 4: *Comparison Between Measured Outlet Air Temperatures and Simulated Outlet Air Temperatures from the University of Saskatchewan Simulation Model.*

Figure 5 illustrates the corresponding liquid outlet temperatures for the monitored data, and simulation results. The monitored temperatures are consistently higher than the simulated results. The likely reason for this result, as in the heat rates, is because the monitored heat capacity rates were slightly lower than what was used in the simulation, perhaps due to uncertainties in the glycol-water density and specific heat.

Figure 5: *A Comparison Between Measured and Simulated Outlet Liquid Temperatures for a Single Run-Around Heat Recovery System Heat Exchanger Coil*

3.2 Coil Design Studies of an Example Heating Coil

In the following sections, several air heating coil design studies were undertaken. Firstly, the coil design selection and expected operating conditions are compared to our simulation results for a conventionally designed heating coil. Secondly, the CHESS program simulation model was compared to the U of S validated simulation model using the operating conditions and parameters from the conventional coil design. Finally, the least Life-Cycle cost heating coil was designed and compared to the selection of the manufacturer conventionally designed heating coil.

3.2.1 Conventional Coil Design

Engineered Air supplied a heating coil design that met the example conditions in the CHESS program simulation. The heating coil in the CHESS simulation program was to provide a 20°C (68°F) outlet air temperature at the cold weather design conditions with an air flow rate of 1 m^3/s (2119 cfm). It was assumed that the cold weather design temperature should be -35°C (-31°F) based on design data for Saskatoon, Saskatchewan. The purpose of having manufacturer design is threefold. First, the expected performance

of their coil can be compared to the U of S simulation model. Secondly, the coil parameters can be used to compare the CHESS simulation model to our simulation model. And finally, the conventional design can be compared to an optimum design.
The heating coil design specifications provided by Engineered Air are shown in Table 2:

Table 2: *A Conventionally Designed Heating Coil Selected to Correspond to the Example Conditions used in the CHESS Program*

Parameter	Material	Supply Coil	
Airflow, L/s (cfm)		1000	(2119)
Liquid Flowrate, L/s (US gpm)	37% Ethylene-glycol and Water Mixture by Volume	1.71	(27)
Coil Width, cm (in)		76	(30)
Coil Height, cm (in)		53	(21)
Number of Tube Passes		4	
Number of Tube Rows		2	
Tube Outer Diameter, mm (in)	Copper Tubing	16	(0.625)
Tube Inner Diameter, mm (in)		15	(0.59)
Tube Transverse Spacing, mm (in)		36	(1.4)
Tube Longitudinal Spacing, mm (in)		32	(1.25)
Fin Thickness, mm (in)	Aluminum Fin Stock	0.14	(.0055)
Fin Wave Depth, mm (in)		2.1	(0.082)
Fin Spacing, fins/mm (fins/in)		0.394	(10.0)

This heating coil is to provide 73.3 kW (250,177 Btu/h) at an entering liquid temperature of 80°C (176°F) and an entering air temperature of -35°C (-31°F). The outlet water temperature is 68.9°C (156.0 °F), and the outlet air temperature is 25.7°C (78.3°F). The air pressure drop across the coil is 44.8 Pa (0.18 inH_2O), and the liquid pressure drop across the coil including header is 12.47 kPa (1.81 psi).

Although, a coil with 2 rows 4 passes cannot be exactly simulated in our program, this coil is thermally equivalent to a coil with 2 rows 2 passes, a coil height of 26.7 cm (10.5 in) and coil width of 152.4 cm (60 in). Table 3 shows the comparison between our simulated results and the manufacturer's data for its heating coil.

Table 3: *Comparison Between the Manufacturer's Specifications for a Conventionally Designed Heating Coil, and the Simulation Model Results*

	Manufacturer's Specification	Simulation Model
Inlet Air Temp, °C (°F)	-35 (-31)	-35 (-31)
Outlet Air Temp, °C (°F)	25.7 (78.3)	25.4 (77.8)
Inlet Liquid Temp, °C (°F)	80 (176)	80 (176)
Outlet Liquid Temp, °C (°F)	68.9 (156)	68.8 (155.8)
Air Flow Rate, m^3/s (cfm)	1 (2119)	1 (2119)
Liquid Flow Rate, L/s (US gpm)	1.70 (27)	1.70 (27)
Heat Rate kW (Btu/h)	73.3 (250,177)	74.7 (254,831)
Air Pressure Drop, Pa (inH_2O)	44.8 (0.18)	44.1 (0.177)
Liquid Pressure Drop, kPa (psi)	12.47 (1.81)	8.98 (1.3)

The outlet temperatures from the simulation model are close to the manufacturer's results. The simulation model gives slightly lower values for liquid pressure drop and slightly higher values for the heat rate. The reason for the differences in these results may be a safety factor in the coil selection process. Perhaps there are innaccuracies in the manufacturer's coil model.

3.2.2 Validation of the CHESS Program

The output parameters from our simulation program for the manufacturer sized coil was used as input in the 3 cell model used in the CHESS program. The variables required for the CHESS model are shown in Table 4.

Table 4: *Parameters Required in the CHESS model Calculated Using the U of S Simulation Model for the Conventionally Designed Heat Exchanger*

Parameters Used in CHESS Program	Value
Liquid Density, kg/m^3	1030.7
Air Density, kg/m^3	1.222
Liquid Specific Heat, kJ/(kg K)	3.7854
Air Specific Heat, kJ/(kg K)	1.0102
Liquid Side Heat Transfer Area, m^2	1.0028
Air Side Heat Transfer Area, m^2	18.74
Liquid Volume, m^3	0.00375
Air Volume, m^3	0.02034
Fin Volume, m^3	0.00123
Tube Volume, m^3	0.00047
Liquid Side Heat Transfer Coefficient, $W/(m^2C)$	6065.4
Air Side Heat Transfer Coefficient, $W/(m^2C)$	105.9
Fin Efficiency	0.692
Collar Resistance ,°C/W	1.2712×10^{-4}
Wall Resistance, °C/W	1.12×10^{-6}

The effective air side heat transfer coefficient is calculated using Equation 25 and the results from the simulation as shown in Table 4. The effective air side heat transfer coefficient is 62.32 W/(m^{2}°C) and the liquid side heat transfer coefficient is 6065 W/(m^{2}°C). These heat transfer coefficient values are very different from the assumed values currently used in the CHESS program of 150 W/(m^{2}°C) for the air side, and 2000 W/(m^{2}°C) for the liquid side. There will be errors in the CHESS simulation if the current assumed heat transfer coefficients are used.

The assumed shape values "M" and "K" currently used in the CHESS program (page 19 in the manual) are significantly different from the simulated values for the conventional coil design using the U of S program. "M" is a shape factor that sizes the outer coil area knowing the inside tube area. In the CHESS program, "M" was assumed to be 15. Using the geometric parameters for the conventionally designed (Engineered Air) heating coil, the value of "M" is equal to 18.7. "K" is a shape factor that helps calculate the air volume based on the tube and fin volume. In the CHESS program the assumed value was 4. For the conventional coil design, this value is 10.4. Significant errors will occur if the assumed values for "M" and "K" are used instead of the actual values calculated from the geometric parameters of the designed coil.

The following example illustrates the potential for error if the shape factors and assumed heat transfer values are used in the thermal analysis.

<u>Conventionally Designed Heating Coil</u>

<u>CHESS Program Calculation of Overall Heat Transfer Coefficient, UA</u>

L	=	Tube Length	= 1.524 m/pass
N	=	Total Number of Tubes in Bank	= 14
d_i	=	Tube Inner Diameter	= 0.01499 m
A_i	=	Inside Area = $\pi d_i L N$	= 1.00 m^2
A_o	=	Outer Area = $A_i M$	= 15.0 m^2
h_i	=	Inside Heat Transfer Coefficient	= 2000 W/m^2 °C
h_o	=	Outside Heat Transfer Coefficient	= 150 W/m^2 °C

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \tag{32}$$

These data result in UA = 1058.8 W/°C

ρ_t	=	Average Density	= 8819 kg/m^3
Vt	=	Volume of Tube and Lamells = m_t/ρ_t	= 0.000846 m^3
V_a	=	Air Volume = K V_t	= 0.02372

U of S Simulation of Heat Coil, Calculation of UA

h_l	=	Liquid Heat Transfer Coefficient	= 6065.4 W/m^2 °C
h_a	=	Air Heat Transfer Coefficient	= 105.92 W/m^2 °C
η_f	=	Fin Efficiency	= 0.692
A_l	=	Liquid Side Area	= 1.00 m^2
A_a	=	Air Side Area	= 18.74 m^2
R_c	=	Collar Resistance	= 1.2712×10^{-4} °C/W
R_w	=	Wall Resistance	= 1.12×10^{-6} °C/W

$$\frac{1}{UA}=\frac{1}{h_l A_l}+R_c+R_w+\frac{1}{h_a A_a \eta_f} \tag{33}$$

These data result in UA = 979.4 W/°C.

The difference in the overall heat transfer coefficient between these two simulations is 8% in this example.

V_t	=	Combined Tube and Fin Volume	= 0.00593 m^3
V_a	=	Calculated Air Volume	= 0.02034 m^3

There is a significant difference in the tube and fin volume of 86%, but because the shape factor is too small, the calculated air volume only differs by 17%.

Supply Side Education Coil (Monitored Coil)

CHESS Program Calculation of Overall Heat Transfer Coefficient, UA

L	=	Tube Length	= 3.353 m/pass
N	=	Total Number of Tubes in Bank	= 206
d_i	=	Tube Inner Diameter	= 0.01499 m
A_i	=	Inside Area = πd_i L N	= 30.59 m^2
A_o	=	Outer Area = A_i M	= 458.9 m^2
h_i	=	Inside Heat Transfer Coefficient	= 2000 W/m^2 °C
h_o	=	Outside Heat Transfer Coefficient	= 150 W/m^2 °C

These data result in UA = 32,388 W/°C

ρ_t	=	Average Density	= 8819 kg/m^3
Vt	=	Volume of Tube and Lamells = m_t/ρ_t	= 0.04421 m^3
V_a	=	Air Volume = K V_t	= 0.3165

U of S Simulation of Supply Coil. Calculation of UA

h_l	=	Liquid Heat Transfer Coefficient	= 1733 W/m^2 °C
h_a	=	Air Heat Transfer Coefficient	= 75.8 W/m^2 °C
η_f	=	Fin Efficiency	= 0.717
A_l	=	Liquid Side Area	= 30.59 m^2
A_a	=	Air Side Area	= 752.7 m^2
R_c	=	Collar Resistance	= 3.49×10^{-6} °C/W
R_w	=	Wall Resistance	= 7.0×10^{-9} °C/W

These data result in UA = 21,355 W/°C.
The difference in the overall heat transfer coefficient is 52% in this case.

V_t	=	Combined Tube and Fin Volume	= 0.07913 m^3
V_a	=	Calculated Air Volume	= 0.6832 m^3

There is a significant difference in the tube and fin volume of 44%, and because the shape factor is too small, the calculated air volume differs by 47%.

The previous example shows the potential errors that can occur by using the CHESS program as it exists. In the first example of the conventionally designed heating coil, the error is small (8% error in the overall heat transfer coefficient). This is because of compensation between the assumed heat transfer coefficient, and the heat transfer area calculated using the shape factor "M". Alternatively, with the larger coils (the monitored coils), there is a much larger error (52% error in the overall heat transfer coefficient). The calculated tube and lamellas volume and air volume will also have error in the current CHESS program. Using the simplified model of an average coil density, and a shape factor to calculate the air volume can lead to large errors (although in the heat transfer analysis, the effect of coil and air volumes on the dynamic heat transfer may be small).

The CHESS program currently uses a nominal heat transfer coefficient, and calculates the change in heat transfer coefficient resulting from a change in fluid flow rate by using an exponent, n, as shown in Equation 26. It is assumed that the air flow rate is generally turbulent, and the liquid flow rate is generally laminar. For turbulent flow the CHESS program uses an exponent, n = 0.8, which is correct for the Dittus-Boelter equation for Reynolds numbers above 10000. For laminar flow, the CHESS program uses an exponent, n = 0.33. Normally, for laminar flow, the Nusselt number is constant, so theoretically there should not be a change in the heat transfer coefficient.

We have found that for most designs the liquid flow is turbulent, and usually above a Reynolds number of 4000 (for the conventional coil design it is 29000) so using the exponent, n = 0.33 will underestimate changes in the heat transfer coefficient. Table 5 shows the comparison between the actual heat transfer coefficient found using the U of S simulation program for reduced liquid flow rates, and the calculated heat transfer coefficient using the nominal heat transfer coefficient, the exponent, n = 0.8, and Equation

26. It can be seen that there is little difference between the actual and calculated heat transfer coefficients. The difference can be attributed to property variations, and that the Nusselt number is calculated using the Gnielinski correlation which is valid above Reynolds numbers of 2300. It is suggested that the U of S simulation program provided be used to determine the heat transfer coefficient for varying flow rates, and then an exponent determined because it is possible that the liquid flow, when reduced, will change from a turbulent to laminar flow regime. In this case, the exponent n = 0.8, will not be valid.

Example of Calculating the Heat Transfer Coefficient using Equation 26.

$h = h_{nominal}(Q/Q_{nominal})^n$
$h = 6065.4\ (1.58/1.70)^{0.8}$
h = 5703 (the actual heat transfer coefficient is 5651 which corresponds to n = 0.92 not n = 0.8)

Table 5: *Variation in Liquid Side Heat Transfer Coefficient for the Conventionally Designed Heating Coil with Varying Liquid Flow Rates.*

Liquid Flow Rate L/s, (US gpm)	Actual Heat Transfer Coefficient (W/m^2K)	Exponent, n, corresponding to Actual Heat Rate	Heat Transfer Coefficient Using n = 0.8 (W/m^2K)
1.70 (27)	6065.4 (nominal)	not applicable	6065.4 (nominal)
1.58 (25)	5651	0.92	5703
1.26 (20)	4585	0.93	4770
0.95 (15)	3467	0.95	3790

The air flow is usually laminar, or in the transitional region rather than turbulent as suggested in the CHESS manual. It would be expected that the heat transfer coefficient for laminar flow would be constant. This is not exactly the case because of the complexity of the flow caused by wavy fins, and the staggered tubes. Table 6 shows the variation in the heat transfer coefficient with changing air flow rates. The actual heat transfer coefficient found from the U of S simulation program does not change significantly with varying air flow rates. If the nominal heat transfer rate, an exponent n = 0.8, and Equation 26 is used, the heat transfer coefficient will be significantly different from the actual values. It is suggested that if using the exponent, n, is required in the CHESS program, that it be calculated using the U of S simulation program. This can be done by varying the air flow rates for a given coil design, and determining the actual heat transfer coefficient at each air flow rate then calculating the exponent, n, using Equation 26.

Table 6: *Variation in Effective Air Side Heat Transfer Coefficient for the Conventionally Designed Heating Coil with Varying Air Flow Rates.*

Air Heat Transfer Rate m^3/s (cfm)	Effective Air Heat Transfer Coefficient (W/m^2K)	Exponent, n, corresponding to Actual Heat Rate	Effective Heat Transfer Coefficient Using n = 0.8 (W/m^2K)
1.0 (2119)	62.32 (nominal)	not applicable	62.32 (nominal)
0.944 (2000)	61.0	0.38	59.5
0.826 (1750)	58.1	0.37	53.5
0.708 (1500)	54.8	0.37	47.3

Using the geometric parameters, fluid properties, and heat transfer coefficients found by our simulation program, a three cell model solved Equations 27, 28, and 29. A comparison of the results is shown in Table 7.

Table 7: *Comparison Between the CHESS Program Model and the U of S Simulation Model Using the Same Physical Parameters based on the Conventional Heating Coil Design.*

	Air Temp. In, °C	Air Temp. Out, °C	Liquid Temp. In, °C	Liquid Temp. Out, °C	Shell Temp., °C
CHESS Cell 1	**-35**	-10.2	73.3	**68.7**	3.6
CHESS Cell 2	-10.2	9.7	77.0	72.4	20.8
CHESS Cell 3	9.7	**25.8**	**80**	76.6	34.7
U of S Simulation	**-35**	**25.4**	**80**	**68.8**	

It is apparent that as long as the correct geometric parameters, fluid properties, and heat transfer coefficients are used, the heating coil model in the CHESS program will accurately simulate the performance of a heating coil.

4 .Conclusions

It is apparent that the model used in the CHESS program is valid only if the correct heating coil input values are used. A validated simulation program has been developed that produces the required input values needed for a simulation in the CHESS program.

The geometric parameters required in the CHESS program can be calculated knowing the dimensions of the heating coil being used in the HVAC system. The dimensions that are required are:

Height of the heating coil, H
Width of the heating coil, W
Longitudinal tube spacing, S_l

Transverse tube spacing, S_t
Tube outer diameter, D_o
Tube inner diamter, D_i
Fin spacing, s_f
Fin thickness, t_f
Fin wave depth, P_d, if wavy fins are used.

The other input values that are required are:

Air flow rate
Liquid flow rate
Glycol concentration for freeze or burst protection
Inlet air temperature, pressure and humidity
Inlet liquid temperature.

The CHESS program model appears to accurately simulate a heating coil at steady state conditions as long as the correct input values are used. The current assumptions used as input values in the CHESS model will lead to significant differences in the results in most cases. The calculations of air volume, and liquid and air side heat transfer coefficients are not accurate using the current assumptions and will lead to the largest errors.

5. References

1. Schmidt, T.E., 1949, "Heat Transfer Calculations for Extended Surfaces". Refrigeration Engineering. Vol. 57.

2. Gnielinski, V., 1976, " New Equation for Heat and Mass Transfer in Turbulent Pipe and Channel Flow". International Chemical Engineering, Vol. 16.

3. Gray, D.L., and R.L. Webb., 1986, "Heat Transfer and Friction Correlations for Plate Finned Tube Heat Exchangers Having Plain Fins". Proceedings of the 8th International Heat Transfer Conference, San Francisco, CA.

4. Webb, R.L., 1990, "Air-side Heat Transfer Correlations for Flat and Wavy Plate Fin and Tube Geometries". ASHRAE Transactions., Vol. 96., Pt.2., Atlanta, GA.

5. Beecher, D.T. and T.J. Fagan., 1987, "Effects of Fin Pattern on the Air-side Heat Transfer Coefficient in Plate Finned Tubed Heat Exchangers". ASHRAE Transactions., Vol. 93., Pt. 2., Atlanta, GA.

6. Bennett, I.J.D., R.W. Besant, G.J. Schoenau, and A.B. Johnson., 1994, "A Procedure for Optimizing Coils in a Run-Around Heat Recovery System", ASHRAE Transactions, Vol. 100, Pt. 2. Atlanta, GA.

7. Johnson, A.B., R.W. Besant, G.J. Schoenau 1995, "Design of Multi-Coil Run-Around Heat Exchanger Systems for Ventilation Air Heating and Cooling", accepted for publication in ASHRAE Transactions.

Acknowledgements to Steve Kuzma, Northern Saskatchewan Sales representative for Engineered Air Inc. for providing a heating coil design corresponding to the example given in the CHESS program manual.

APPENDIX

A brief introduction to the use of CHESS-ESI

The simulation program is adapted from a simulation tool named CYPROS SIM, which is a program for solving nonlinear dynamic problems. The SIM program is mainly a shell containing numerical solvers for solving differential equations. The program is also supported by flexible input/output routines for data presentation.

The adaption of the program primarily consists of developing a modular system solution. A modular structure is essential to enable a smooth and flexible change of system solutions, and each component is described by its own subroutine. The modular simulation technique greatly reduces the complexity of system simulation because it essentially reduces a large problem into a number of smaller problems, each of which can be more easily solved independently. In addition many components are common to different systems as well as being repeated within one system. Provided the performance of these components is described in a general form, they can be used in many different systems and often within the same system with little or no modification.

As previously explained CHESS-ESI consist of the following three system configuration which have to be simulated separately:

1) **ESI-REF** The reference system (See Figure 4.1, part 1).

2) **ESI-II** The system with serial connection of heating coil and radiator system on the secondary side (See Figure 4.2, part 1).

3) **ESI-III** The system with serial connection of heating coil and radiator system on the primary side (See Figure 4.3,part 1).

Each system has its data-file given the same name with the extension ***.dat**.

Manual for operating "CHESS-ESI"

The manual will give a brief description of:

- Requested hardware
- Installation
- The user interface
- Special features on system simulation
- Error messages

Requested hardware

- An IBM 486 or Pentium is recommended.
- Math-coprocessor for 386 and 486SX

Installation

The complete content of the floppy must be copied to your PC on the same directory. You are free to create any suitable directory for the simulation tool.

In order to run the graphic result presentation you must choose a proper graphic driver.

> Place the following in your AUTOEXEC.BAT SET CYPROSDISP = 562.
> This is the driver for VGA-screen 640*480 16 colours that can be used for most PCs.

The user interface

The simulation program is started by typing: **ESI-REF <CR>**, **ESI-II <CR>** or **ESI-III <CR>** depending on simulating system.

The program now asks for the data file which has corresponding names.

The following text will now appear on the screen: **Main Command set (simulation):......**
By typing **?** the available options will be listed:

Data-file:	Operations on the data-file, change parameters and initial values etc.
Simulation:	Choose solution method, specify time interval, step size etc and simulate.
Output:	Show the results on screen, as numeric tables or on TS-files.
Parameter:	Change system parameters, eg. the dimension of the solution table.
Call-User:	Execute subroutine USER once.
Help:	Not supported for CHESS-ESI.
Exit:	Leave the program.

By typing the first letter in the available options and <CR> the command will be executed.

Figure A.1 shows the simulation tool command structure. The most frequently used commands are written in bold text.

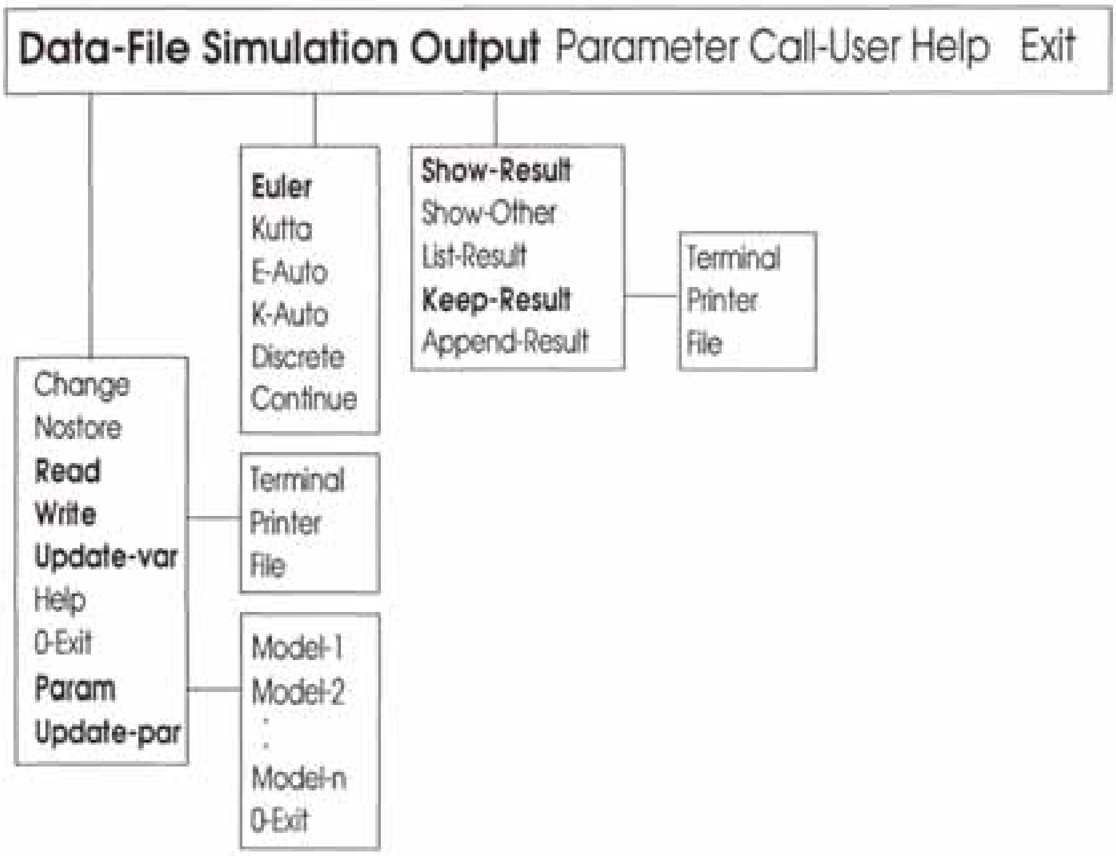


Figure A.1 Simulation tool command structure

Running "Simulation" mode:
In this mode your simulation problem will be solved. You will here interactively choose which method to use (Euler) and decide time interval and step size. The simulation tool also provide alternative solution methods to Euler which will not be described in this text.

Note that you can halt a run by pressing any key during simulation. If you want to restart the simulation you enter the "Parameter" option and turns SUSPEND ON (no. 4).

When running "Simulation" mode the following text will occur:

Solution method (Euler):............	: Euler	Euler is the default value which should be used.
Time interval (Ti,Tf) (0.0,0.0).....	: 0,10	Initial simulation to create parameter structure.
Step size (0.0)..................	: 0.1	"Normal" step size for this system.
Simulating.....		If you just want to run the system with default
I======================>		parameters you should set Tf > 400.
# of samples 100		
# stored 100		
Main command set (output)....	:	When the simulations are completed the system returns to the "Main command set".

After this initial simulation sequence it is possible to change parameters of the various system components by entering "Data-file" mode.

Running "Data-file" mode

In this mode you can change the initial parameter values of the system components. You can also write to new data files, or read from old files. You can also specify the variables you want to store during a simulation. A default set of parameters are normally stored.

Typing ? the available options will be listed:

Change:	not in use
Nostore:	With this command you can specify the variables which you wish not to be stored during simulation
Store:	With this command you can make the variables restored again
Read:	Read from specified *.Dat file
Write:	Write to specified *.Dat file
Update:	Updates the initial process and control states with the actual states
Param:	Enables change of initial parameters
Oppdat:	Updates the parameter states, eg. flow variables.
0-Exit:	Return to "Main Command" level

Running "Output" mode:

When the simulation is completed, you can enter "Output" mode. Here you select how to present the results. By typing ? the following options will appear:

Show-Result:	Display the result on screen
Keep-Result:	Copy all variables from the solution table to a specified TS-file
List-Result:	Show the result as numeric tables on terminal, printer or ASCII-file. Up to 3 variables separated by commas can be shown simultaneously
Append-Result:	Append the result to an existing TS-file.
0-Exit:	Return to Main Command level

By using List-Result and ASCII-file the data can be prepared and presented in a spredsheet.

Stabilizing the system

System evaluation request an initial stable system before starting the simulation sequence. The following instructions will indicate how to stabilize the system and how to perform simulations:

1. If you want to change model parameters you have to do an initial short simulation to establish the parameter structure.
2. Enter the "Data-File" mode and choose the option PARAM.
3. Change the requested model parameters
4. Enter "Simulation" mode and do a long simulation (2-3000 sec.) to stabilize the system.
5. Check if the system is stable by controlling strategic temperature states in "Output" mode (show-result).
6. When unstable system, enter "Data-File" mode and use the options UPDATE-PAR and UPDATE-VAR and repeat from point 4.
7. When stable system, enter "Data-File" mode and use the options UPDATE-PAR and UPDATE-VAR and PARAM.
8. Now the system can be exposed to variation in eg. inlet temperatures, volume flows, controller setpoints etc.

Error messages

Unfortunate choice of model parameters can lead to errors during simulation which usually occur as rolling error messages on the screen. The reason for the error is usually that high flow speed in a part of the system causes numerical instability. The following procedure should be used when error messages occur:

1. Press **<CR>** to stop the simulation
2. Write **P <CR>** to enter "system parameter" option
3. Write **4 <CR>** to turn suspend off which enables a restart of the simulation
4. There are two alternatives when restarting the simulations:
   a. reduce the time step
   b. change critical parameters eg. flow capacities and pipe dimensions
5. A third possibility is to do a complete restart of the program which will start with default values.

Simulation example

The objective of the simulation example is to illustrate the use of the simulation tool. The system model used for this purpose is the Reference system. The example is made as a demonstration of the possibilities provided by the simulation models both for static and dynamic analysis, and focuses on the service hot water preparation system.

The system is simulated with increasing hot water flow from zero to design flow in six steps. This variation serves two purposes:

1) The controller settings can be evaluated over the total range of operation

2) The primary temperature difference

The following text takes the user step-by-step through the example.

Example

Start the simulation by typing **ESI-REF<CR>** at the DOS prompt.

Screen picture:

```
          ****************************************
          **                                    **
          **   IEA DISTRICT HEATING AND COOLING   **
          ** EFFICIENT SUBSTATIONS & INSTALLATIONS **
          **                                    **
          ****************************************


          SYSTEM I, REFERENCE CONSUMER HEATING SYSTEM

             TWOSTEP CONNECTION WITH RADIATOR
              AND HEAT COIL SYSTEM COUPLED
             IN PARALLEL ON THE SECONDARY SIDE


           Rune Volla, Rolf Ulseth, Jacob Stang
                      SINTEF Energy
                        17.01.96

File name (.DAT)......................................:
```

Write the name of your datafile **IEAREFX1<CR>**

Screen picture:

```
Main command set    (Simulation)....................:
```

We want to run an initial simulation which is necessary to enable parameter change.
Type **<CR>** which make use of the default command (Simulation).

Screen picture:

```
Main command set    (Simulation)....................:
Solution method     (Euler).........................:
```

We want to use the Euler solution method. Type **<CR>** which make use of the default command (Euler).

Screen picture:
```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf) (0.0,0.0)........................:▮
```

Type **0,10<CR>** which initiates a simulation of 10 seconds.

Screen picture:
```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
Step size       (0.0)............................:▮
```

Type **0.05<CR>** which is a convenient time step size.

Screen picture:
```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
step size       (0.0)............................:0.05
Simulating....
|=======>=======================|
```

When the simulation is finished the following screen picture will appear:
```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
step size       (0.0)............................:0.05
Simulating....
|===============================>
# of sample 200
# of stored 200
Main command set    (Output)........................:▮
```

The initial simulation is finished and we want to verify that the solution is stabilized. Use the default parameter (Output); press **<CR>**.

Screen picture:
```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
step size       (0.0)............................:0.05
Simulating....
|===============================>
# of sample 200
# of stored 200
Main command set    (Output)........................:
Output command     (Show-Result)....................:▮
```

Use the default parameter (Show-Result);press **<CR>**.

Screen picture:

```
Main command set    (Simulation)......................:
Solution method     (Euler)...........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
step size        (0.0)............................:0.05
Simulating....
|===============================>
# of sample 200
# of stored 200
Main command set    (Output)..........................:
Output command     (Show-Result)......................:
Result command     (Y-Variable(s))....................:█
```

Use the default parameter (Y-Variable(s));press **<CR>**.

Screen picture:

```
Main command set    (Simulation)......................:
Solution method     (Euler)...........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
step size        (0.0)............................:0.05
Simulating....
|===============================>
# of sample 200
# of stored 200
Main command set    (Output)..........................:
Output command     (Show-Result)......................:
Result command     (Y-Variable(s))....................:
Specify variable(s)....................................:█
```

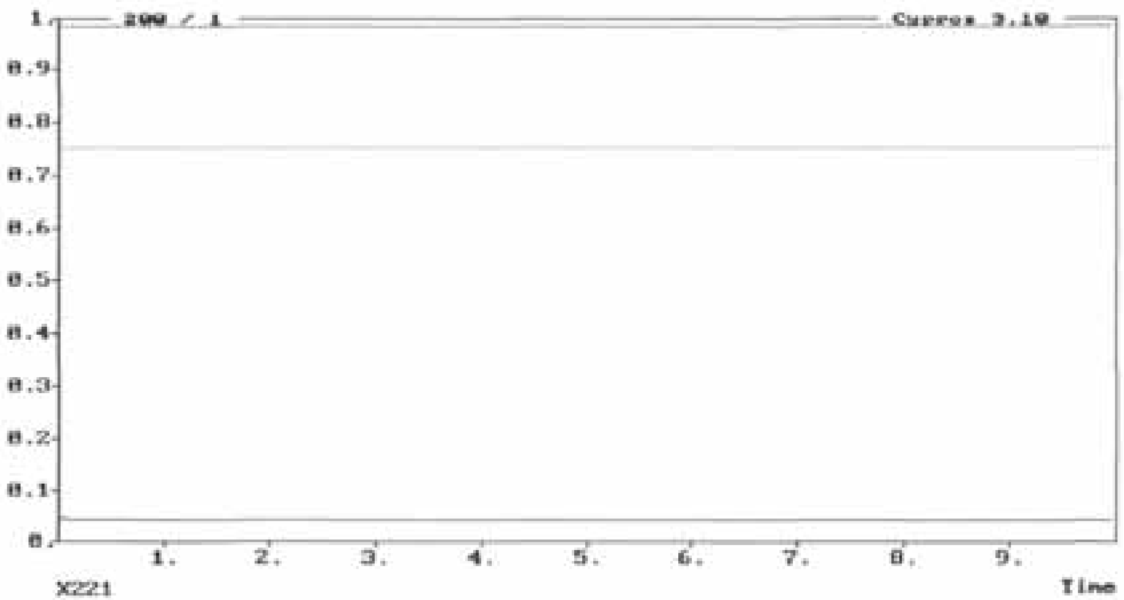
From Figure A.2 we choose to see the controller signals X221,X222,X223 and X224. Type **X221,X222,X223,X224<CR>**

Screen picture:

```
Main command set    (Simulation)......................:
Solution method     (Euler)...........................:
Time interval (Ti,Tf) (0.0,0.0)........................:0,10
step size        (0.0)............................:0.05
Simulating....
|===============================>
# of sample 200
# of stored 200
Main command set    (Output)..........................:
Output command     (Show-Result)......................:
Result command     (Y-Variable(s))....................:
Specify variable(s)....................................:X221,x222,x223,x224
Result command     (Separate-Axis)....................:
```

Type **P<CR>** to plot with a common axis.

Screen picture:




```
Result command        (Plot)..........................:_
```

We can see that all controller output signals have stabilized. We can then continue with the simulation of the steps in the hot water flow.

Exit from the graphics module by pressing **0<CR>**.

Screen picture:

```
Result command     (Plot)............................:0
Output command     (Show-Result)....................:▮
```

Exit from the output module by pressing **0<CR>**.

Screen picture:

```
Main command set    (Simulation)....................:▮
```

We are now back at the Main command set. We want to simulate the effect of steps in the hot water flow. To achieve this, we have to set the excitation in the hot water flow model.

Type **D<CR>** to enter the data module.

Screen picture:

```
Main command set    (Simulation)......................:d
Data file operations (Change)..........................:▮
```

Press **P<CR>** to enter the menu for parameter change.

Screen picture:

```
         1 OUTDOORTEMP                   14 HEAT COIL 1
         2 PRIMARY INLET TEMP.           15 PIPE 3
         3 PLATE HEAT EXCHANGER          16 VENTILATION DUCT
         4 TWO WAY VALVE 1               17 PIPE 4
         5 CONTROLLER 1                  18 PLATE HEAT EXCHANGER 2
         6 PIPE 1                        19 PIPE 5
         7 TWO WAY VALVE 2               20 TWO WAY VALVE 4
         8 CONTROLLER 2                  21 CONTROLLER 4
         9 RADIATOR 1                    22 PIPE 6
         10 TWO WAY VALVE 3              23 STEP FLOW EXCITATION
         11 CONTROLLER 3                 24 PLATE HEAT EXCHANGER 3
         12 EXCITATION VENT.AIR INLET    25 EXCITATION 1
         13 PIPE 2                       26 PIPE 7
                                         27 PIPE 8
28 GLOBAL VARIABLES
WRITE MODEL NO.:    ( 0 )............................:█
```

Type **23<CR>** to set the flow step size and time interval between the steps.

Screen picture:

```
█    5     COLD WATER TEMPERATURE [°C]
    1     FLOW MODE [1-Step, 2-Measurements]
    0     INITIAL FLOW [m3/s]
    0     FLOW STEP [m3/s]
  2000     TIME INTERVAL BETWEEN STEP [s]
 0.0006     FLOW LIMIT [m3/s]
```

Move the cursor with the down arrow to the line with flow step and type: **0.0001<SPACE>.** Move the cursor to the line with time interval between step and type: **200<SPACE>.**
Exit from the flow step excitation module by pressing: **<SPACE>**.
Exit from the parameter module by pressing **0<CR>**.
Exit from the data file operation module by pressing **0<CR>**.
Back at the Main command set type **<CR>** to select the default; Simulation.
Type **<CR>** to select the default simulation method; Euler.
Type **0,1200<CR>** as the time interval to simulate.
Type **<CR>** to select the default time step; 0.05 s.

Screen picture:

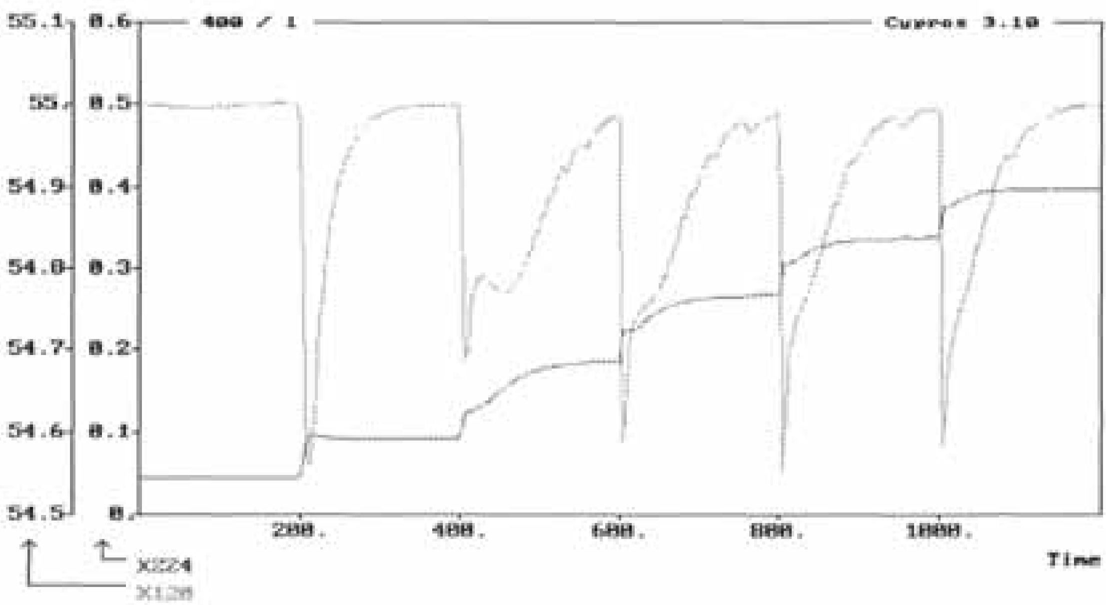
```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf) (0.0,10.0)......................:0,1200
step size        (0.05)...........................:
Simulating....
|>=============================|
```

When the simulation is finished type **<CR>** to select the default; Output.
Type **<CR>** to select the default; Show-Results.
Type **<CR>** to select the default; Y-variable(s).
Type **X120,X224<CR>** to select the hot water temperature and controller output.
Type **S-A<CR>** to plot the two variables with separate axis.

Screen picture:

```
Main command set    (Simulation).....................:
Solution method     (Euler)..........................:
Time interval (Ti,Tf)  (0.0,10.0)......................:0,1200
step size         (0.05)............................:
Simulating....
|==============================>
# of sample 24000
# of stored  400
Main command set    (Output).........................:
Output command     (Show-Result)....................:
Result command     (Y-Variable(s))..................:
Specify variable(s)   (X221,x222,x223,x224)............:X120,X224
Result command     (Plot)..........................:S-A
```

The effect of the steps on the hot water temperature and the controller output is shown in the following screen picture:

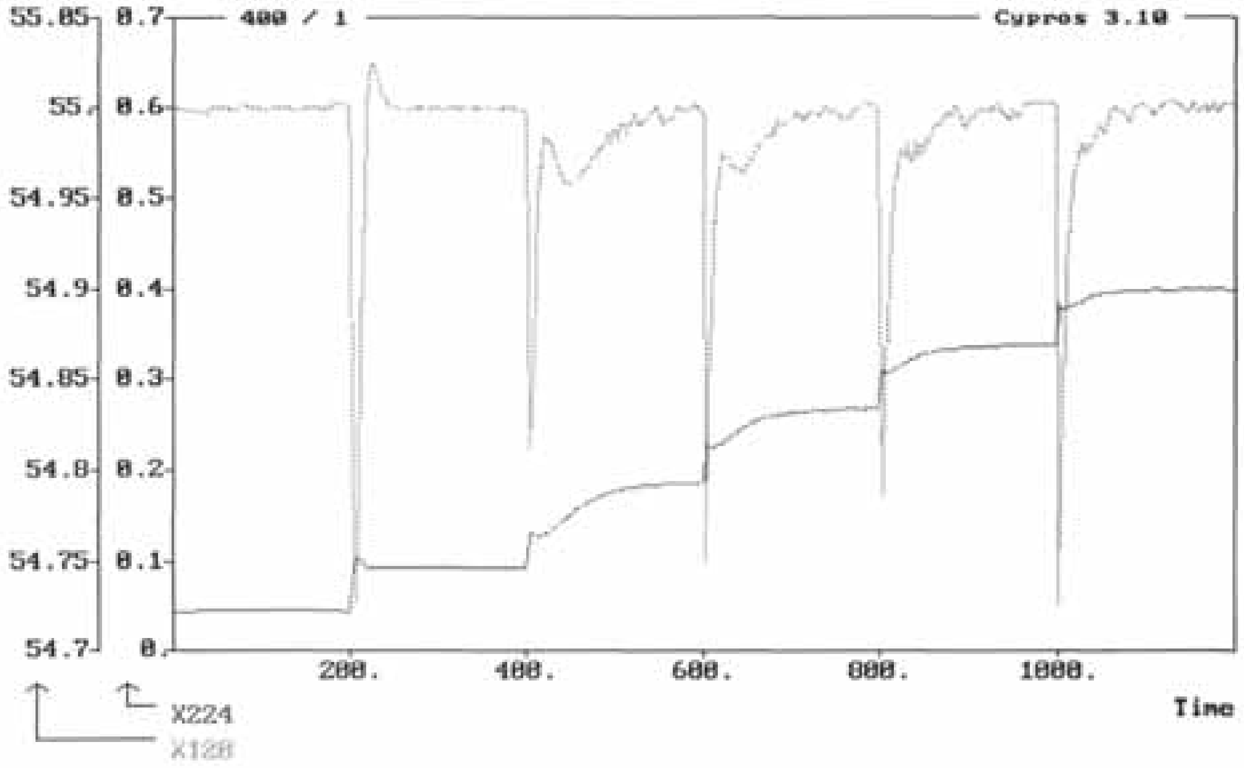



```
Result command          (Separate-Axis)................:_
```

From the screen picture we can see that the controller and thereby the hot water temperature is responding somewhat slowly to the steps in hot water flow. We can also notice that the controlling ability is better in the low and high range than in the middle range. We will try to speed up the response on the steps by increasing the controller gain and decreasing the integration time. The derivation time is leaved unchanged.

Exit from the graphics module by pressing **0<CR>**.
Exit from the output module by pressing **0<CR>**.
From the main command set type **D<CR>** to enter the Data file operations.
From the Data file operations type **P<CR>** to enter the Parameter module.
Type **21<CR>** to change the controller parameter in CONTROLLER 4 which controls the hot water temperature according to Figure A.2.
Move the cursor by the down arrow to the line with controller gain and type **0.2<SPACE>**.
Move the cursor by the down arrow to the line with controller integration time and type **10<SPACE>**.
Press **<SPACE>** to exit from the controller module.
Press **0<CR>** to exit from the parameter module.
Press **0<CR>** to exit from the Data file operations module.
Back at the Main command set type **<CR>** to select the default; Simulation.
Type **<CR>** to select the default simulation method; Euler.
Type **<CR>** to select the default time interval; 0,1200.
Type **<CR>** to select the default time step; 0.05 s.
When the simulation is finished type **<CR>** to select the default; Output.
Type **<CR>** to select the default; Show-Results.
Type **<CR>** to select the default; Y-variable(s).
Type **<CR>** to select the default variables X120,X224 (hot water temperature and controller output).
Type **<CR>** to plot the two variables with separate axis.

Screen picture:



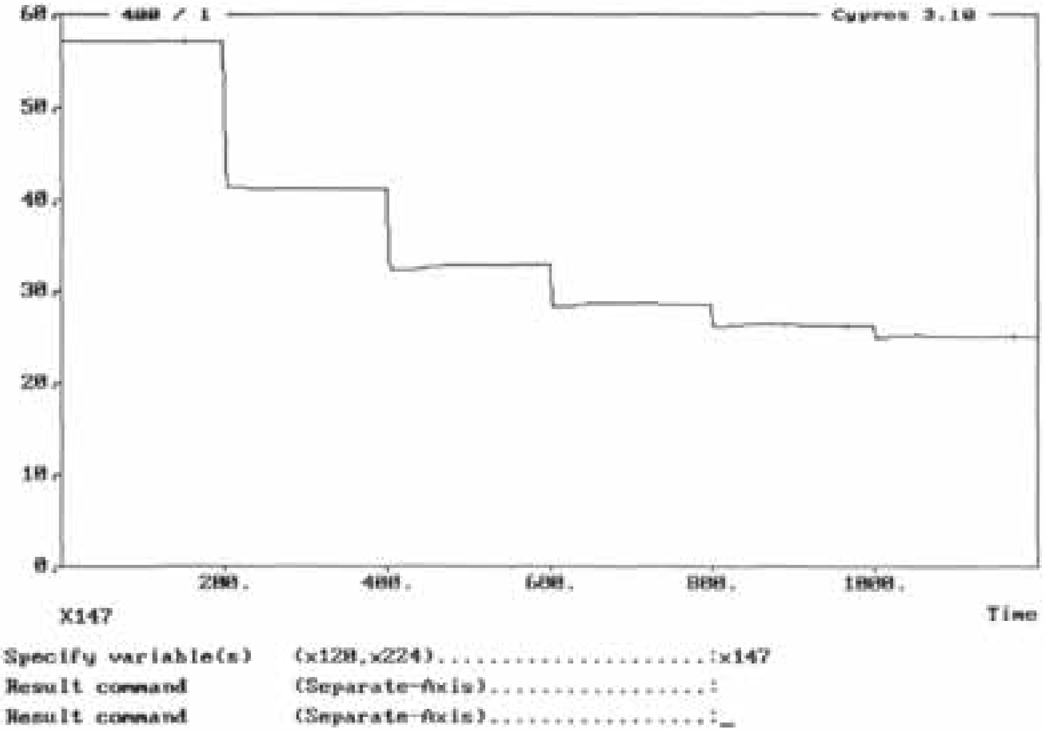

```
Result command         (Separate-Axis)................:_
```

Observe that the hot water temperature control has improved noticeably due to the changes in controller parameters.

We can also study the variation in primary return temperature which is caused by varying hot water flow.

Type **Y-V<CR>** to select Y-variable(s).
Type **X147<CR>** to select the primary return temperature.
Press **<CR>** to plot with the default; Separate-Axis.

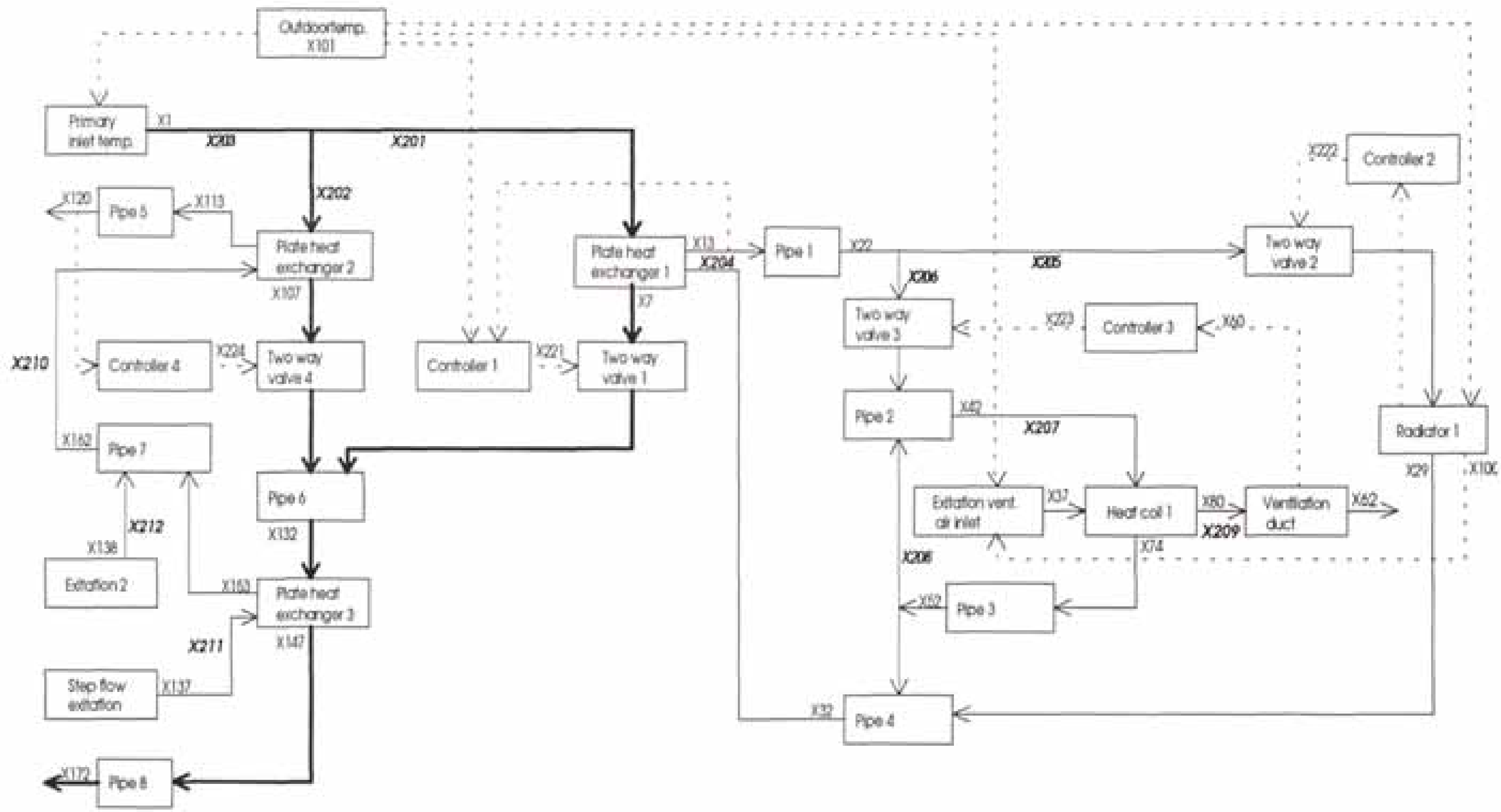
Screen Picture:



The latter part demonstrates the utilization of the tool to analyse the static system performance.

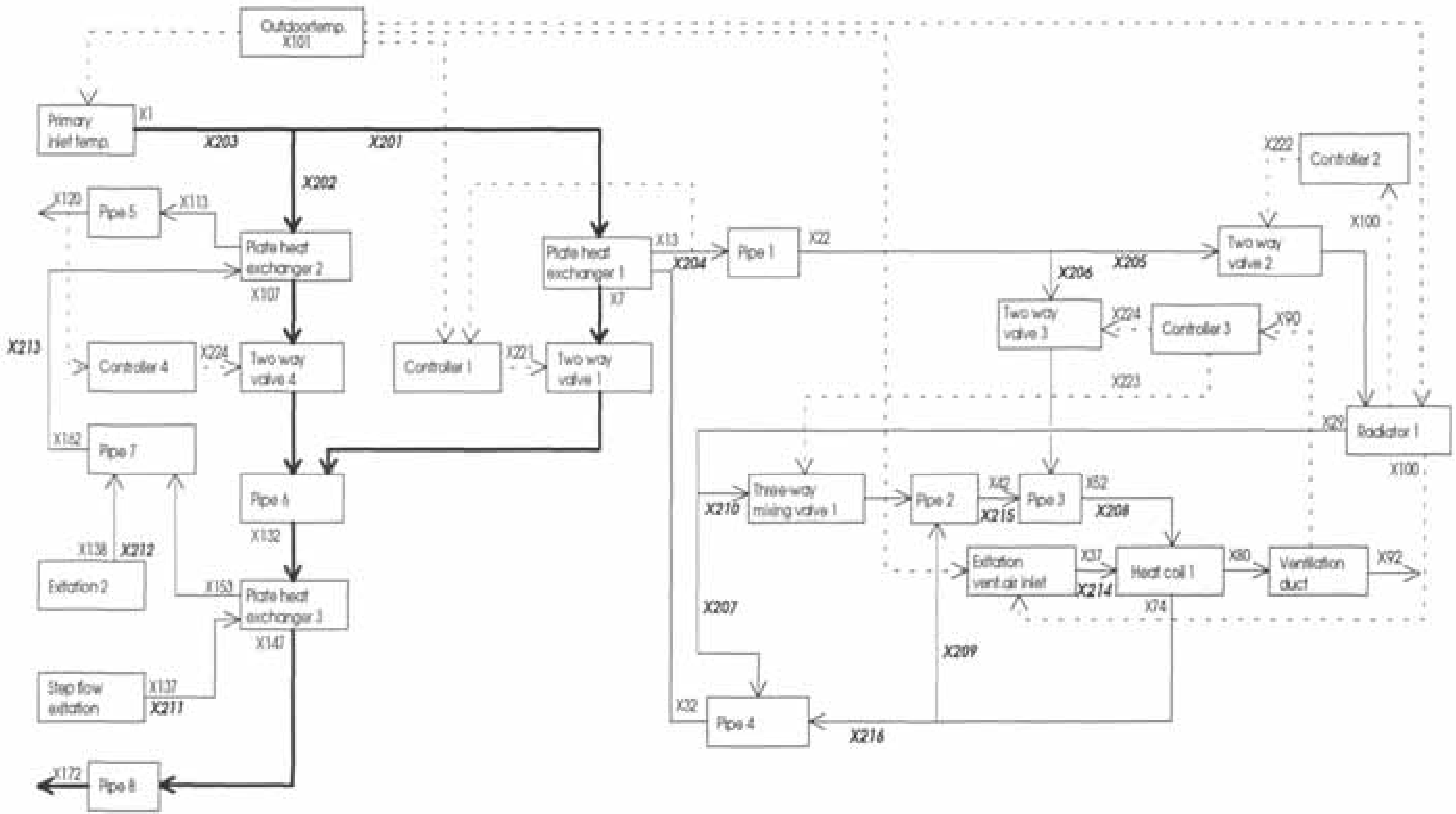
System Configuration

Figures A.2 to A.4 show the configuration of the three simulation systems. The numbers in the figures correspond to the numbers in the system simulation models. The numbers written in ***bold italics***, are the volume flows of the system.



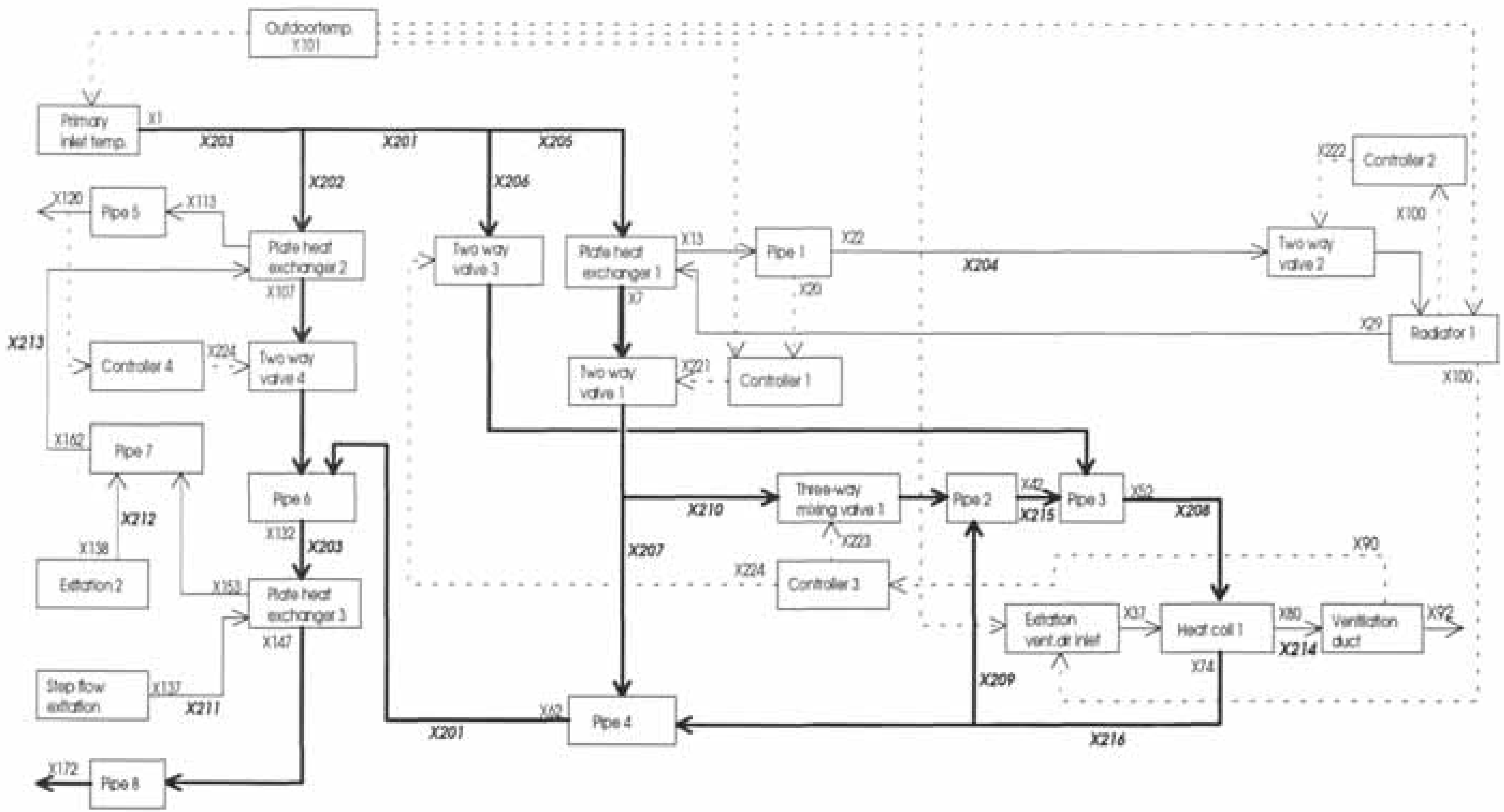
ESI-REF

Figure A.2 The reference system, ESI-REF



ESI-II

Figure A.3 System 2, ESI-II



ESI-III

Figure A.4 System 3, ESI-III

IEA District Heating and Cooling

EFFICIENT SUBSTATIONS AND INSTALLATIONS

Published by
Netherlands Agency for Energy and the Environment

Mailing address: P.O.Box 17, 6130 AA Sittard, the Netherlands
Street address: Swentiboldstraat 21, Sittard
Telephone: +31 46 4595295
Telefax: +31 46 4528260

1996: N5
ISBN 90-72130-88-X