Forskning och Utveckling

FOU 2001:59

ANALYS AV TRERÖRSYSTEM FÖR KOMBINERAD DISTRIBUTION AV FJÄRRVÄRME OCH FJÄRRKYLA

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FJÄRRVÄRME

Analysis of Three-pipe-Systems for Combined Heating and Cooling Distribution

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Master Thesis for Mechanical Engineering at EKOS, Dalarna University, Borlänge/Falun

June 2001 / ISSN 1402-5191

ZW
Energiteknik
Report ZW-01/07

Högskolan Dalarna

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Sammanfattning

Denna rapport analyserar förutsättningarna för kombinerade fjärrvärmedistributionssystem för värme och kyla, d v s *system med en gemensam returledning*, t ex trerörsystem (3P) i stället för konventionella fyrarörsystem (4P). I ett 3P-system varierar returtemperaturerna avsevärt under säsongen, från att vara nära värmesystemets returtemperatur vintertid till att närma sig kylsystemets returtemperatur på sommaren, beroende på kombinationen mellan kravet på uppvärmnings- och kyleffekt. I det här valda exemplet (maximal kyleffekt ca 50 % av värmeeffekten) är de gemensamma returtemperaturerna ca 48°C (vinter) respektive 25°C (sommar). Genom effektivare användning av rökgaskondensorer bör dessa lägre returtemperaturer kunna användas för effektivare drift av biobränslepannor. Å andra sidan betyder detta också att mer energi måste tillföras värmekretsen och på samma gång krävs högre kylkapacitet för att kyla ner returflödet till kylkretsens framledningstemperatur. I praktiken innebär detta att i 3P-systemet överförs energi från värmedistributionskretsen till kylkretsen som antingen måste kylas bort genom kylkretsens kyltorn eller/och delvis återvinnas med en värmepump. I det senare fallet är värmepumpens kondensor inkopplad i värmedistributionskretsen.

De allmänna förutsättningarna för de 3P-system som undersökts i rapporten är att 3P-system använder samma ledningsbädd som 4P-system och att både värme- och kylsystem, 4P såväl som 3P, installeras vid samma tillfälle. Det antogs emellertid att i genomsnitt endast var tionde fjärrvärmekund var ansluten till fjärrkylsystemet. Dessutom antogs att nätet ut från värme-/kylproduktionen delas upp på två huvudstammar. I grundanalysen inkluderades endast kostnaderna för kylutrustning och rör i den ekonomiska analysen, medan biobränslepannan antogs vara densamma för alla modeller. Beträffande driftkostnaderna valdes priset för elektricitet till 350 SEK/MWh och för biobränsle till 80 SEK/MWh. Denna prisskillnad speglar en typisk situation på den svenska energimarknaden. Investeringskostnaderna räknades om till årliga kostnader med en annuitet på 8 %.

I rapporten jämförs tre olika modeller för 4P- respektive 3P-system. *Modell 1* är basfallet för helt åtskilda fjärrvärme- och fjärrkylasystem när det gäller 4P. För 3P används värmeväxlare för återvinning av en del energi, vilket håller fjärrvärmereturen för uppvärmning på 30°C, när den gemensamma returtemperaturen T_R är lägre, och fjärrkylareturen på 30°C, när T_R ligger över detta värde.

I *Modell 2* samkopplas värme- respektive kylkretsarnas framledningar via en värmepump, vars förångare tar upp energi från kylkretsen och vars kondensor leverar energi för att förvärma värmemediet vid inloppet till biobränslepannan. Maximal förvärmningstemperatur till returledningen (sommartid) är 80°C (tvåstegs värmepump). På så sätt kan en stor del av värmen som kyls bort i Modell 1 återvinnas och den totala energin i ett sådant system ligger mycket nära energin i ett helseparerat 4P-system av Modell 1. En

nackdel är att en betydande del av den avgivna värmen härrör från kompressorarbete, dvs elektrisk energi, och detta är normalt ett dyrt sätt att producera värme.

För att undvika denna nackdel med Modell 2 genomförs en del av nedkylningen i *Modell 3* med ett första kyltorn (endast i 3P-system), som sänker den gemensamma returtemperaturen under sommaren till fjärrkylasystemets framledningstemperatur och endast genererar så mycket värme som krävs för att leverera sommarlasten till värmepumpen. Detta system använder totalt mer energi än Modell 2, men mycket mindre elkraft och kan därför under vissa omständigheter vara mer ekonomisk än Modell 2.

I alla 3P-system används fler eller större komponenter (värmeväxlare, kyltorn, värmepumpar) än i 4P-systemen. Men de högre *investeringskostnaderna* för denna utrustning kompenseras av lägre rörkostnader eftersom det endast krävs *en* returledning. Om man, generellt sett, jämför investeringskostnaderna för produktion av kyla med investeringarna för rör (kostnaderna för värmeanläggningen ej inkluderade), framgår det att 3P-systemen i allmänhet har lägre investeringskostnader än 4P-systemen.

Motsatsen gäller för *driftkostnaderna* (biomassabränsle och elektricitet), vilka för 3P-systemen ligger mycket över (Modell 1 och 2) eller något över (Modell 3) driftkostnaderna för 4P-systemen. Generellt sett är 3P-systemen mer energiintensiva än 4P, och bör normalt *inte* komma ifråga av energiekonomiska skäl.

I vissa fall, vid låga kyllaster (maximal kyleffekt 12,5 % av den maximala värmeeffekten), och när priset på elektricitet är lågt, ger jämförelsen resultat som påvisar att de totala årliga kostnaderna för 4P-systemen och 3P-systemen kan vara ganska lika. Det gäller särskilt när den totala energiförbrukningen för 3P och 4P inte skiljer sig åt för mycket. Kostnadsskillnaderna är emellertid små och dessa system kräver mer detaljerade analyser inför varje tillämpningsfall.

Sammanfattningsvis kan konstateras att 3P-system energitekniskt inte kan anses som ett lämpligt alternativ för fjärrkyladistribution. Detta har sin grund i *exergiförlusten* som åstadkoms genom temperaturblandningen: Returtemperaturen sänks i det varma mediet och höjs i det kalla genom blandningen. Den potentiella energivinsten i en rökgaskondensor som arbetar vid lägre temperaturer och som var motivet för introduktion av 3P-systemet kan inte till fullo kompensera dessa exergiförluster (vilka måste kompenseras genom extra tillförd energi). De eventuellt lägre investeringskostnaderna för 3P-distributionssystemet upphävs genom en mera komplicerad kylprocess som behövs för att effektivisera 3P-systemet. Totalt sett har det inte framkommit några fördelar mer än i undantagsfall (låga elpriser) som kan motivera ett trerörsystem.

Summary

This study analyses the possibility of constructing combined district heating and cooling systems with a joint return pipe, i.e. three-pipe systems (3P) instead of conventional four-pipe (4P) systems. In a 3P system the return temperatures varies widely under the season, from being closest to the heating system return temperature under winter time to being close to the cooling system return temperature summer time, depending on the combination between heating demand and cooling demand. In the chosen example in this paper (maximum cooling load 50 % of maximum heating load) the temperatures are about 48°C (winter) and 25°C (summer), respectively. These lower return temperatures are thought to be used for more effective operation of the biomass cooling plant by using stack gas condensers more effectively. On the other hand, this means also that more energy has to be supplied to the heating circuit and at the same time, a higher cooling capacity is necessary to cool down the return flow to the cooling supply temperature. In practice it means that in the 3P system energy is transferred from the heating circuit to the cooling circuit and either must be rejected by means of the cooling tower of the cooling circuit, or/and partially recovered by means of a heat pump. In the latter case the condenser of the heat pump is connected to preheat the water of the heating circuit.

The general presumptions for the 3P systems investigated in the study are that the 3P system uses the same pipe trench as the 4 P system and that both the heating and cooling systems, 4P as well as 3P, are installed at the same occasion. However, it was assumed that in average only each 10th customer was connected to the district-cooling network. Furthermore, it was assumed that the network out of the heating/cooling plant is divided into two main stems. In the basic analysis, only the cost of the cooling equipment and pipes were included in the economic analysis, whereas the biomass heating plant was taken to be the same in all models. As to the operating costs, the price of electricity was chosen to 350 SEK/MWh and that of biomass fuel to 80 SEK/MWh. This cost difference reflects a typical situation in the Swedish energy market. The investment costs were converted to annual costs by means of an annuity rate of 8 %.

In the study, three different models are compared for 4P and 3P system, respectively. *Model 1* is the base case for completely separated heating and cooling systems in the 4P case; in 3P heat exchangers are used for recovering part of the energies, keeping the heating return temperature at 30°C whenever the common return temperature T_R is below this value, and keeping the cooling return temperature at 30°C, whenever T_R is above that temperature.

In *Model 2,* a heat pump is interconnecting the supply pipes of the cooling and heating circuits, recovering energy by means of its evaporator cooling the cooling supply pipe and preheating the heating medium at the entrance of the biomass plant. The maximum preheating temperature (summer time) is 80°C (two-stage heat pump). In that way a large amount of the heat rejected in Model 1 can be recovered and the total energy of such a system is quite close to that of the totally separated 4P Model 1 system. However, an important part of the heating energy is electrical (compressor) work, and hence normally this is an expensive way of producing heat.

To avoid this disadvantage of Model 2, in *Model 3* part of the cooling was achieved by a pre-cooling tower (3P system only), bringing down a large amount of the common return temperature summer time to the cooling supply temperature and leaving only as much energy as necessary for supplying the summer load to the heat pump. This system uses totally more energy than Model 2, but much less electricity and can therefore under some circumstances be more economic than Model 2.

In all 3P cases, extra or larger equipment (heat exchangers, cooling towers, heat pumps) are used compared to the 4P systems. But the *investment costs* for this equipment can be compensated by lower costs for omitting one return pipe. So generally speaking, if cooling investment and pipe investment costs (heating plant costs not included) are compared, it can be seen that 3P systems generally have lower investment costs compared to 4P systems.

The opposite holds for the *operating costs* (fuel and electricity). The operating costs of the 3P systems are far above (Model 1 and 2) or slightly above (Model 3) those of 4P systems. That also means that 3 P systems are more energy intensive than 4P and therefore should in general *not* be applied for the reason of good energy economy.

In some occasions, where electricity prices are low**,** i.e. cases were the energy consumption of 3P and 4P systems does not differ too much, the comparison gives results indicating that the total annual costs of the 4P systems and 3P systems are quite similar at low cooling loads (maximum cooling load 12,5 % of maximum heating load). For these cases with low electricity prices, some favourable conditions for 3P systems could emerge. However, the differences are quite small and such systems need more detailed analysis in the case of application.

To sum up it can be stated that technically the 3P system cannot be considered a suitable alternative for distribution of district cooling. This is due to the loss of exergy caused by temperature mixing: the return temperature is reduced in the warm medium and increased in the cold medium. The potential gain of energy in a stack gas condenser working at low temperatures, which was the reason for introduction of the 3P system cannot completely compensate these losses of exergy (which have to be compensated by extra supply of energy). The possible lower investment costs for the 3P distribution system is counteracted by a more complicated cooling process needed to make the 3P system more effective. After all, advantages motivating a three-pipe system were only found in exceptional cases (low prices for electricity).

Acknowledgement

This work has been performed as a master thesis for Thermal Engineering at the University of Dalarna, Solar Energy Research Center, Borlänge, Sweden. The study was carried out at the consultant company ZW Energiteknik AB in Nyköping, Sweden. The work was primarily ordered and financed by the Swedish District Heating Association, Stockholm.

It is a great pleasure for me to acknowledge gratefully the guidance and assistance of my supervisors, Dr. Heimo Zinko, ZW Energiteknik, and Prof. Lars Broman, SERC. Lars Broman offered me the possibility to complete my education in Sweden and gave me a lot of friendship, knowledge and practical help to reach my study goal. I thank Heimo Zinko for the competent and daily guidance in the project and for the final correction of it. His illuminating instructions and helpful suggestions let me smoothly complete all of the model designs and modelling and optimisation work.

I would also like to express my gratitude to Civ. Ing. Peter Margen, who contributed with valuable ideas and hints regarding possible designs and function of district cooling systems and equipment. His profound knowledge was very helpful in completing this work.

Table of Content

Nomenclature

C	Investment costs	[SEK]
C_p	Heat capacity	[kJ/kg C]
E_c	Compressor work	[kJ]
E_{cr}	Energy rejected by cooling tower	[kJ]
E_{EL}	Electrical auxiliary energy	[kJ]
E_{fuel}	Energy provided by the biomass plant	[kJ]
I	Specific costs	[SEK/MWh]
i_{ck}, i_{ek}, i_s	Enthalpy	[kJ/kg]
k	Internal efficiency of heat pump	
$\dot{m}$	Mass flow	[kg/sec]
$\dot{m}_c$	Mass flow of cooling system	[kg/sec]
$\dot{m}_{c-1}$	Mass flow to evaporator of the heat pump	[kg/sec]
$\dot{m}_{c-2}$	Mass flow to evaporator of the chiller	[kg/sec]
$\dot{m}_h$	Mass flow of heating system	[kg/sec]
O_e	Operating costs (electricity)	[SEK/MWh]
O_f	Operating costs (fuel)	[SEK/MWh]
P_c	Power of district cooling system	[W]
P_c	Pressure of condensing medium(chiller/heat pump)	[kPa]
P_{c-load}	Design cooling load	[W]
P_{cond}	Condenser power	[W]
P_{cool}	Cooling power	[W]
P_e	Pressure of evaporating medium(chiller/heat pump)	[kPa]
P_e	Cooling power of heat pump evaporator	[W]
P_{EL}	Compressor work	[W]
P_h	Power of the district heating system	[W]
P_{h-load}	Design heating load	[W]
Q_{aux}	Auxiliary energy to heating system	[kJ]
Q_c	Heat taken up by evaporator or chiller	[kJ]
Q_{c-ex}	Energy provided by cooling heat exchanger the cooling mass flow	[kJ]
Q_{h-ex}	Energy provided by heating heat exchanger for pre-heating the heating mass flow	[kJ]
Q_{c-load}	Cooling load	[kJ]
Q_{con}	Energy delivered by heat pump (chiller) condenser	[kJ]
$Q_{cool-chiller}$	Cooling capacity of chiller	[kJ]
Q_{ct}	Cooling capacity of cooling tower	[kJ]
$Q_{el-chiller}$	Electrical work of the chiller	[kJ]
Q_{EL-HP}	Electrical work of heat pump compressor	[kJ]
Q_{eva}, $Q_{eva-chiller}$	Cooling capacity of chiller	[kJ]
Q_{fuel}	Supplied biomass fuel	[kJ]
Q_h	Energy delivered by heat pump to heating circuit	[kJ]
T_a	Ambient temperature	[°C]
T_c	Temperature of low temperature heat source of a heat pump	[°C]

T_{ci}	Evaporator inlet temperature or cooling tower ourlet temp.	[°C]
T_{co}	Condenser outlet temperature	[°C]
T_{con}	Condenser temperature of working medium of chiller	[C]
$T_{co\text{-}try}$, $T_{co\text{-}try\text{-}gu}$	Intermediate condenser temperatures	[°C]
T_{cr}	Cooling water return temperature	[°C]
T_{cs}	Cooling water supply temperature	[°C]
T_e, T_{eva}	Evaporator temperature of working medium of chiller	[°C]
T_h	Temperature of high temperature heat sink of a heat pump	[°C]
T_{hr}	District heating return temperature	[°C]
T_{hs}	District heating supply temperature	[°C]
T_r	Mixed return temperature	[°C]
T_{rc}	Temperature of cooling return flow after heat exchanger	[°C]
T_{rh}	Temperature of heating return flow after heat exchanger	[°C]
T_{ref}	Reference temperature for cooling optimisation	[°C]
W	Electrical compressor work	[W or kJ]

ε	Cooling factor of heat pump or chiller
ε_c	Carnot cooling factor of heat pump or chiller
Ö	Heating factor of heat pump or chiller
$Ö_c$	Carnot heating factor of heat pump or chiller
η	Stack gas condenser recovering efficiency

1 Introduction

1.1 The aim of this study

This paper is to focus on the system function of district heating and cooling systems. Our objective is to systematically analyse the possibility of applying a three-pipe-system instead of conventionally four distribution pipes. ***The working hypothesis is to investigate if it is possible with well-matched heating and cooling devices to substitute the two return pipes in separate four-pipe heating and cooling distribution systems by one common return pipe, returning the heating medium at a mixing temperature as a result of mixing the return flows from the heating and cooling circuits.***

The study is based on analysing different modellised combinations of heating and cooling devices connected to three-pipe and four-pipe distribution systems. Model calculations were performed by means of Excel for a complete year based on hourly load values. On the results of the energetic analysis, cost functions are applied in order to determine the relative price ranking among the systems and to compare cost relations between three-pipe and four-pipe systems.

The basic idea to the work originates from a study about the influence of low return temperature in district heating networks made by Walletun [18]. Combined district heating and cooling systems with one return pipe at mixed temperature could result in very low return temperatures, around 20 C summer time. A literature survey resulted in one article taking up this idea of three-pipe systems [1]. In this study the authors found that three-pipe systems could be cost effective under some conditions, especially where the costs of delivered heat and electricity were quite close. In most other papers different technical solutions for district cooling systems have been analysed [2], [8], [20]. In our paper we concentrated on the best way to apply such alternative system solutions to the technical criteria of 3P systems and to be compared with similar solutions for 4P systems.

2 District heating and cooling systems in Sweden

District heating and cooling (DHC) can be described as a method by which thermal energy from a central source is distributed to residential, commercial and industrial consumers for use in space heating, water heating and process heating as well as cooling. The central source may be a heating plant fueled by fossil fuel or biomass, municipality wastes, a geothermal source, solar energy or one which utilizes heat produced in combined heating and power plants. The latter approach, generally known as “cogeneration”, has a high level of energy utilization efficiency.

2.1 Example from district heating (DH) in Sweden

Sweden has more than 200 district heating systems, distributing annually about 40 TWh heat to the customers. For mentioning an example, in Stockholm, about 80 % of the inhabitants are served by district heating connections. The connected power is about 2200 MW and the annual heat load is about 6,2 TWh. The district heating network length of Stockholm is about 700 km. The heat is produced with different heating sources, as shown in Figure 2.1. As can be seen, only about 23 % of the energy is supplied by fossil fuels in Stockholm.

Another city worth to mention in the context of this paper is Västerås with one of the oldest district heating nets in Sweden. In Västerås, about 1,6 TWh heat are delivered to the inhabitants of the city with a pipe length of about 500 km. Västerås is the city with a very high degree of district heating penetration close to 90 %. A large source for district heating is coal and only about 30 % of the heat production is based on renewable sources.

For Sweden as a whole, about 80 % of all people living in multifamily houses and 7 % of all people living in detached houses are connected to district heating, the amount of fossil fuel produced heat being not more than roughly 20 %. A scheme on a conventional heat distribution system is shown in Figure 2.2. The supply temperatures vary - depending on load type and season - between 80°C and 100°C and the return temperature is normally around 50°C.

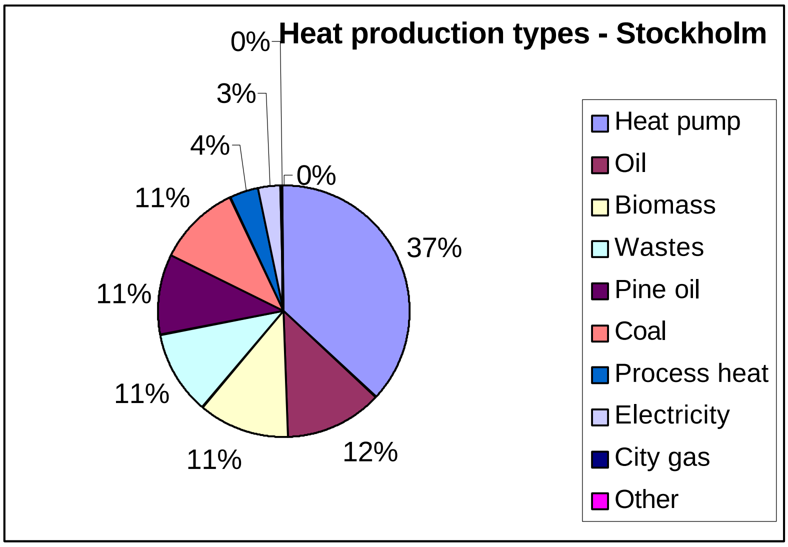
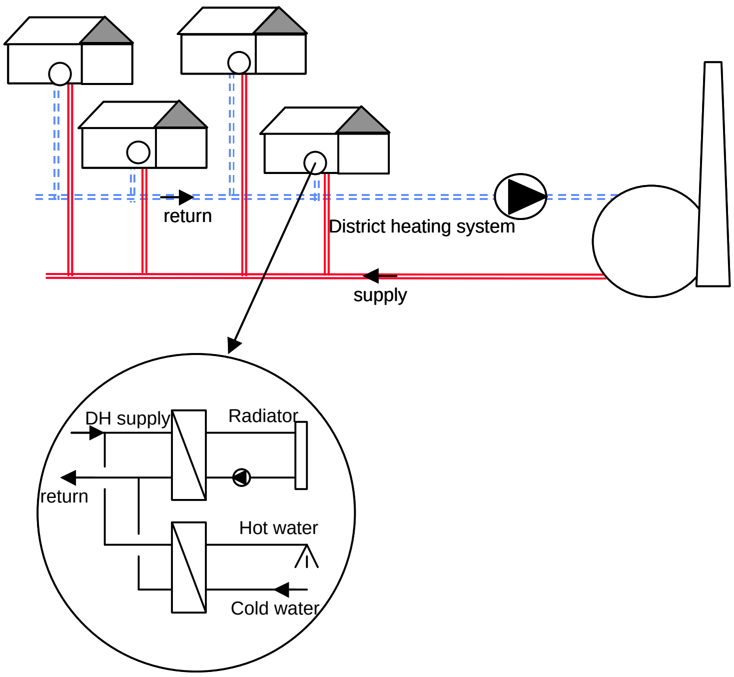


Figure 2.1: Types of heat production from different sources in Stockholm (delivered energy 6,2 TWh/yr).



Figur 2.2: Schematic district heating network based on two-pipe distribution system.

2.2 Example from district cooling (DC) in Sweden

District heating is a quite a recent technology. In Sweden the, first district cooling system was built in Västerås 1992. In principle this technology works similarly to the district heating system shown in Figure 2.2. In this case however, a central cooling device produces cooled water at temperatures between 4 and 7°C and distributes it to the local cooling loads were the fluid is heated up to temperatures around 15°C.

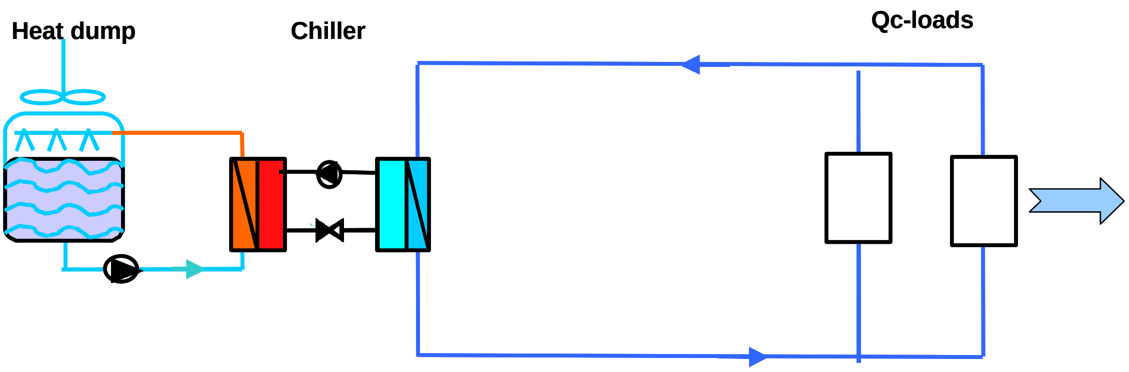


Figure 2.3: Schematic district cooling system based on two-pipe distribution system.

In Västerås, a combined district heating heat pump system is used for producing both heating and cooling by means of a heat pump working on a city sewage treatment system. About 20 MW cool are distributed in a network 7 km long, corresponding to 20 GWh annual cooling.

In Stockholm a large part of the 150 MW cooling load is supplied by cold sea water. The cooling distribution system is at present 40 km long and supplies 170 GWh per year.

Thus from these examples it is getting evident that so far the Swedish cooling load is at the best only about a tenth of that of the heating load in typical district applications. Normally, the cooling load consists of highly frequented office and service buildings, whereas the principal heating load is space heating in wintertime but only domestic hot water preparation in summertime.

2.3 Separated networks for district heating and cooling – four-pipe (4P)

In the normal case, the both networks for district heating and cooling are separated. Also, quite usually, the systems have been built at different times with different technologies. The district heating system might be older, built in the 1970- or 1980-ies, whereas the district cooling system has been built quite recently or is going to be built in the next future. In such a situation, two extra pipes are to be laid down in the ground for the cooling system. In larger cities, many distribution systems (heating, cooling, sewage, rain water, electricity, gas, TV, Internet, etc.) are competing for the sparse available ground space. The digging in city space is also expensive and the risk for unintentional interference among the distribution system increases.

Therefore for new systems, it could be useful to investigate if it is possible to combine both heating and cooling networks by a common return pipe, hence saving one pipe and the space occupied by it. The following analysis will try to answer the question.

3 Three-pipe distribution system for combined heating and cooling

3.1 The idea of choosing a three-pipe (3P) distribution system

In biomass fuel heating plants, the heat recovery in the stack gas condensing system is a strong function of the district heating return water temperature T_r (see diagram 5.2). The low mixed return temperature of three-pipe systems can eventually improve the fuel efficiency of such a system. On the other hand, a three-pipe distribution system means in general also a heat transfer from the heating circuit to the cooling circuit and hence the total system economy depends on the relation of heating to cooling load during the year and the system design.

Another important reason for choosing a three-pipe distribution circuit for combined heating and cooling systems is the saving of one distribution pipe as mentioned above. Saving one distribution pipe could reduce the cost of a combined district heating and cooling network by about 10 - 15 %.

However, it should be noted that the combined DHC system is more difficult to design and to adapt to changing load situations such as a growing cooling market compared with the two separated heating and cooling systems. The main reason is that the resulting return temperature may have an impact on the choice especially of the type of chillers.

It should be mentioned already in the beginning that the heat transfer from the heating circuit to the cooling circuit, which occurs in three-pipe systems, means cooling away heat which otherwise would contribute to keeping the heating return temperature high and therefore more heating energy is needed than in 4P systems. Hence the most important question is therefore how large this heat transfer contribution is and if an economic gain by saving one return pipe can justify the 3P system.

3.2 The basic layouts of 3P and 4P distribution systems for combined heating and cooling

The basic schemes for both systems 4P and 3P are shown in Figure 3.1 and Figure 3.2.

In Figure 3.1, the basic design of separate district heating and cooling systems (4P) is shown. In the heating net, the boiler (heating plant) is connected to the load in a closed cycle. A pump provides the mass flow $\dot{m}_h$ and the delivered power P_h is:

$$P_h = \dot{m}_h \cdot C_p \cdot (T_{hs} - T_{hr}) \qquad \text{[W]} \qquad (3.1)$$

The mass flow and the temperature difference determine the transported heating power and the integral over the time determines the delivered energy. Both the mass flow and the supply temperature are normally controlled in order to enable the system to deliver the demanded heating power.

The cooling circuit works in principle in a similar way: A cooling device, in this case an electrically driven compressor chiller, cools the return flow of the cooling circuit providing heat at the condensing site of the heating device. This heat must be rejected by an air cooler or as in this case a wet cooling tower. The cooling power can be expressed similar to equ. 3.1:

$$P_c = \dot{m}_c \cdot C_p \cdot (T_{cs} - T_{cr}) \qquad [W] \qquad (3.2)$$

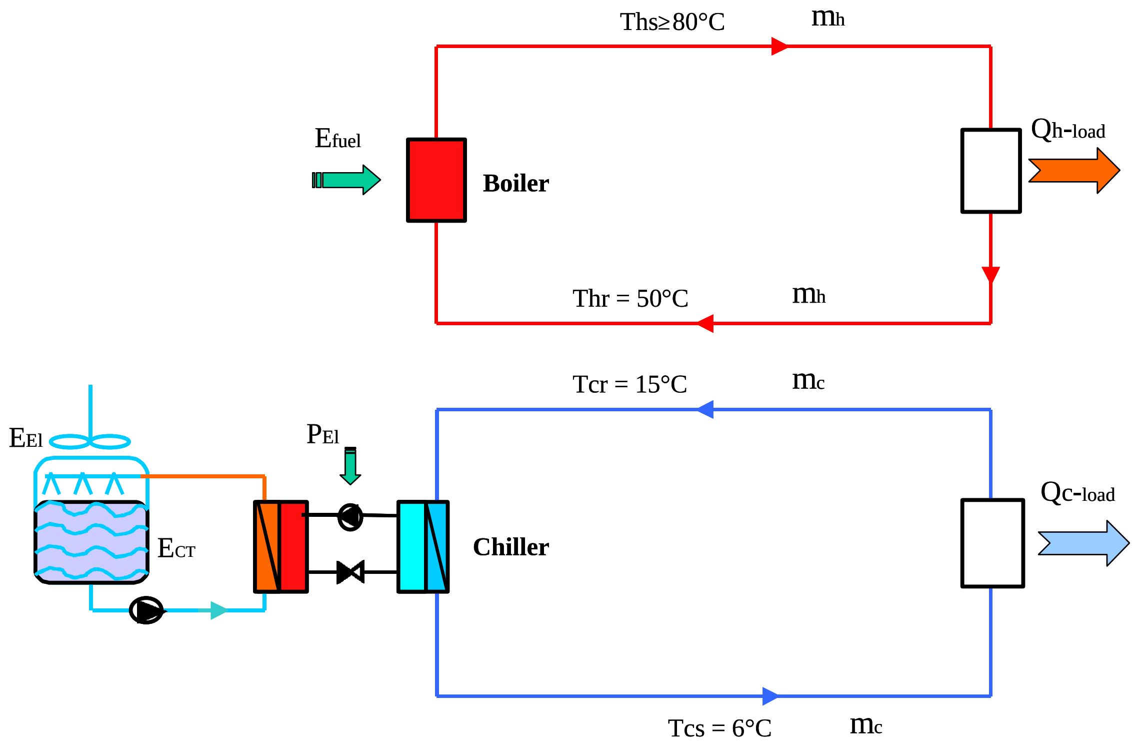


Figure 3.1: Schematic of separated heating and cooling system – four-pipe system.

In the scheme of Figure 3.1, both the heating and cooling systems are completely separated systems. However, it is important to mention that part of the heat, which is rejected in the cooling tower, could be used for preheating the district heating medium, if heat can be rejected at temperatures higher than T_{hr}. This can be principally arranged by letting the condenser of the chiller working at higher temperatures (but lower effiencies). Such an arrangement would result in an interacting DHC scheme according to Figure 4.2.

The three-pipe system is by definition a system that interacts between cooling and heating circuit, the basic idea is getting evident from Figure 3.2.

In order to make the system as efficient as possible it is anticipated that part of the heat produced in the condenser of the chiller can be used for preheating the return flow from the common return pipe if the common return temperature T_r is low enough. Likewise, we assume that the fluid returning from the cooling tower can be used to cool the incoming return flow to the evaporator of the chiller if the common return temperature T_r is high enough.

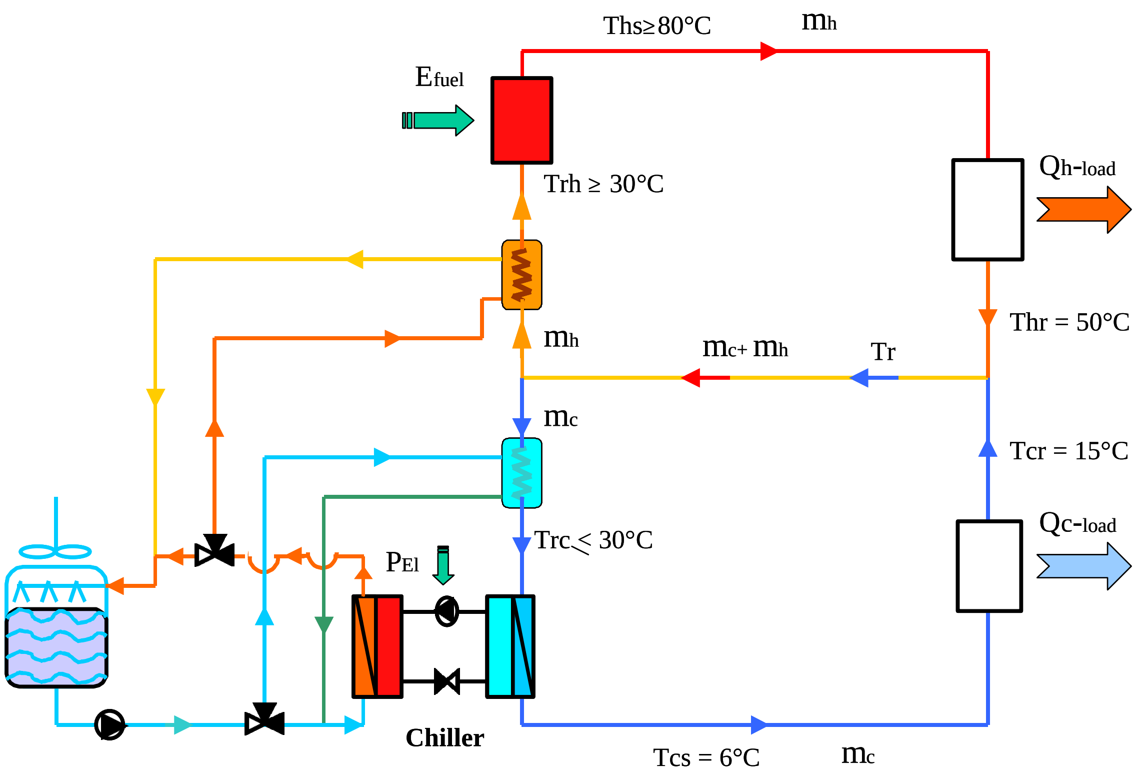


Figure 3.2: Schematic of combined heating and cooling system - three pipe system.

3.3 The basic criteria for a suitably combined three-pipe system

Hence the basic system description above elucidates that the following basic criteria are to be met for the choice of the three pipe system:

- Meeting the requirements from heating load and cooling load all over the year
- Saving substantial amount of distribution pipes
- Consuming fuels for heating and cooling in a three-pipe distribution system should be comparable to those in a four-pipe distribution system
- The heating plants are well matched with the cooling equipments in the cooling circuit
- The overall economy must be favourable.

4 Basic design of combined district heating and cooling systems

In the following, some features for basic heating and cooling production plants are described. As to ***heating plants***, only two types are considered here, i.e.: ***Biomass fired boilers*** in combination with flue gas condensers and ***electrically driven heat pumps***, respectively. In both types of systems, the low return temperature T_r can have some advantage to the system efficiency.

For ***cooling systems***, again ***heat pumps*** as well as ***compressor driven chillers*** in combination with ***dry or wet air coolers*** have been chosen. The heat pumps work directly on the district cooling return fluid.

4.1 Heat production systems

4.1.1 Heating plant using boilers fueled with biomass

In a biomass heating plant, the stack gases leave the boiler at temperatures well about 100°C. Usually, biomass is a fuel with high amount of humidity and hence the hot stack gases can contain a high amount of water vapour. Additionally, water is also produced in the combustion process (burning hydrogen molecules to water). For this reason it is common to recover part of the exhaust heat by a condensing heat exchanger (stack gas condenser). The heating enthalpy of the fuel is usually based on the stack gas temperature of 100°C (upper heating value) and hence does not include the energy which can be recovered at condensation temperatures below 100 C°.

The recovering efficiency of condensing stack gas energy of biomass is a strong function of the temperature of the recovering medium, which f. i. can be the district heating return water. Table 4.1 shows the increase of boiler efficiency η for a biomass-fueled boiler as a function of the temperature of the heat recovering medium, [18].

Table 4.1: Energy recovery factors h for typical wood chips plants versus hot water return temperature T_r :

Tr (°C)	40	45	50	55	60
η (%)	23,50	21,70	18,80	14,80	9,12

The values for η of Table 4.1 can be expressed as a function of T_r :

$$\eta = -0{,}0004\ T_r^{\ 3} + 0{,}0327\ T_r^{\ 2} - 1{,}0489\ T_r + 37{,}914 \qquad [4.1]$$

Hence the biomass energy recovery factors η is a strong function of the return temperature T_r . For instance, when the hot water return temperature T_r can be lowered from

50 °C to 30 °C, the biomass energy recovery factor η increases from 18,80 % to 25,08 %, which increase the energy balance of the total process with 5,3 %. Es an example, Figure 4.1 shows the total energy supply from the biomass plant and the supplied biomass fuel for the 3P system in Model 3. The difference of the both curves is the stack gas energy recovered by the stack gas condenser

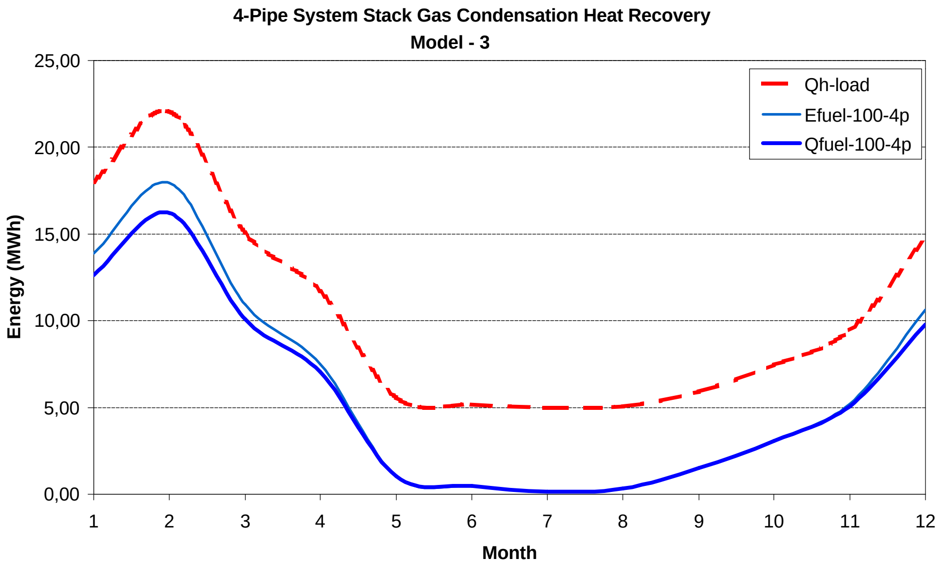


Figure 4.1: Heating load, heating energy and supplied fuel in the system of 3P Model 3.

A scheme of a biomass heating plant shown in Figure 4.2.

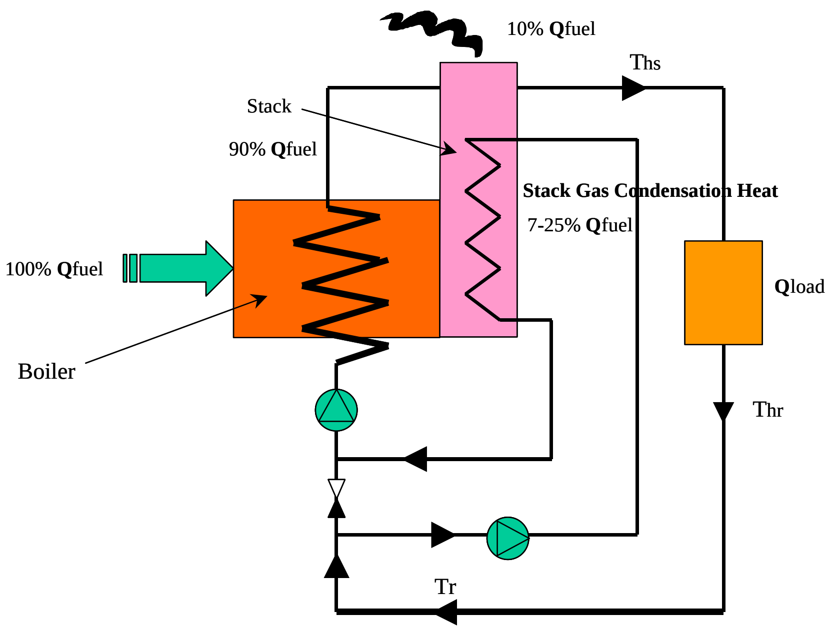


Figure 4.2: Schematic about biomass heating plant with stack gas condenser.

4.1.2 Compressor driven heat pumps

Heat pumps are variations of compressor chillers, used to heat hot water by extracting heat from low temperature heat sources such as outside air, municipal waste water sewage, ground or sea water, etc., see Figure 4.3.

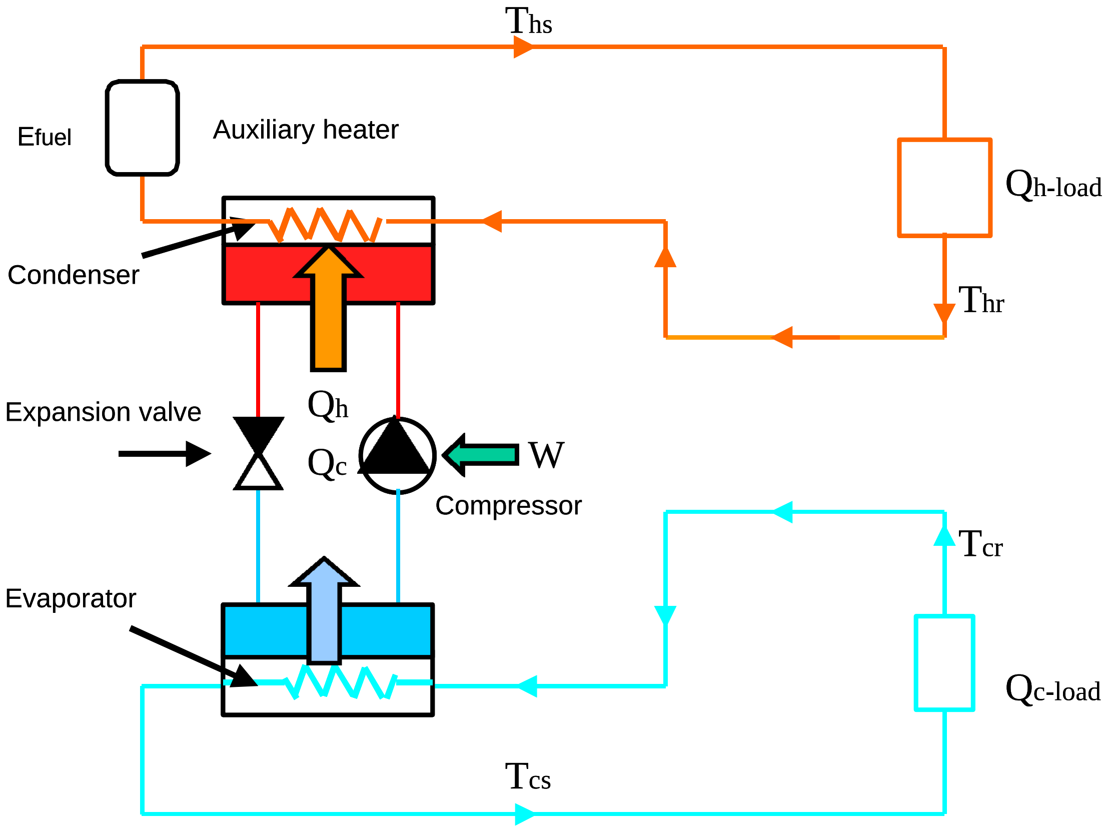


Figure 4.3: Compressor driven heat pump connecting a district heating and district cooling system.

The *evaporator* takes up energy from a low temperature heat source, in this case the district cooling return medium, and the *condenser* delivers heat to a high temperature fluid medium such as the district heating water to be preheated. The heat pump efficiency – or in this case coefficient of performance COP – is definied as follows:

$$COP = Q_h/W = Q_h/(Q_h - Q_c) = k * T_h/(T_h - T_c) \qquad [4.2]$$

where Q_h = Heat given off to a high temperature fluid medium by the heat pump
Q_c = Heat taken from a low temperature heat source by the heat pump
W = Electrical work supplied to operate the compressor
k = Internal efficiency coefficient. In the case of process engines used in this paper, k= 0,6. In the case of an ideal Carnot process, k =1
T_h = Temperature of high temperature heat sink of a heat pump (K)
T_c = Temperature of low temperature heat source of a heat pump (K).

The lower the return temperature of the district heating system T_{hr}, and the higher the return temperature of the district cooling medium, the lower is the efficiency, which can be derived from equation 4.2. With increasing temperature lift (T_h-T_c) the pressure difference in the heat pump increases and therefore the compression work. In practice, in order to transport the energy through the heat exchangers in the heat pump, the evaporator temperature of heat pump fluid is lower than T_c and the condenser temperature is higher than T_h, contributing to the lower internal efficiency factor k. Other loss factors are internal electricity consumptions for pumps and non-ideal expansion and compressions processes. A closer description of the heat pump is given in chpt. 4.2.1 (compression chillers).

Also in heat pumps, the maximum temperature lift to be delivered by a heat pump might be limited depending on the working medium. If higher temperature lifts are to be delivered, the heat pump usually operates in two steps, to and from an intermediate temperature T_m, further reducing the k value and also increasing the price of the heat pump.

In Figure 4.4, an example for typical COPs of heat pumps with one and two stages is presented. R134a is asssumed to be the working medium.

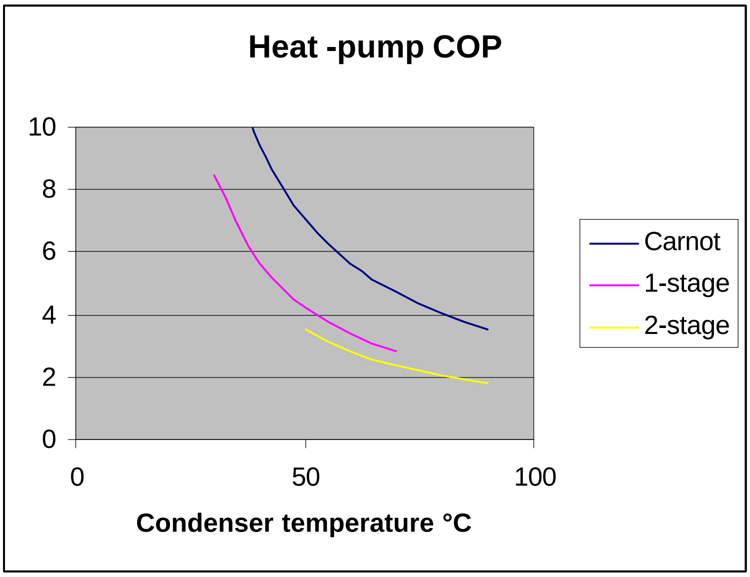


Figure 4.4: Theoretical COPs for 1-stage, 2-stage heat pumps and ideal Carnot processes.

4.2 Suitable cooling systems

4.2.1 Compression chiller (CC)

Function

The compression chiller is the most common type of cooling engines. It is used over a large range of sizes up to tenth of Megawatts. It is in principle the same device as shown in Figure 4.2 for heat pumps, however usually its temperature range is limited to about 60 °C condenser temperature. Eventually other cooling mediums are used in chillers than in heat pumps.

The following functions are known for such a system:

Evaporator

The cooling medium undergoes a phase change from liquid to gaseous phase in the evaporator. Evaporation happens at a low temperature T_e and low pressure P_e. The necessary latent evaporation heat is taken up from the medium to be cooled. The delivered energy per second is $Q_{eva.}$ The evaporation temperature T_e must be lower than the temperature T_c of the fluid to be cooled. The low pressure is sustained by the compressor, which sucks the gas away from the evaporator.

Compressor

The compressor is in our case driven by an electric motor that delivers the mechanical power W to the axis of the compressor. It transports the saturated and dry vapour from the evaporator to the condenser by compressing the vapour along a working curve that is as close as possible to an isentropic process (i. e. process with no change of entropy ($ç_{is}$ =1)). The pressure is increased to P_c and the temperature reached is T_c. In reality, this process is not completely isentropic but values of $ç_{is}$ = 0,8 –0,9 are normal.

Condenser

The condenser serves to reject the heat, which the evaporator has delivered to the fluid. This is done at the temperature T_c at constant pressure P_c. The heat is rejected to the ambient air, or to wet air in the heating tower or also to a medium such as the district heating water in which case the heat can be used for heating purposes. The amount of the rejected heat per second Q_{con} is equal to the mechanical power W from the compressors and the heat per second Q_{eva} delivered by the evaporator. At the exit of the condenser, the fluid should be completely liquefied.

Expansion valve

In order to close the fluid cycle, the fluid has to be expanded through the expansion valve. The expansion process is in principle adiabatic, i.e. there is no energy exchange with the environment. The pressure and the temperature after expansion are again that of the evaporator. The expansion valve also acts as a flow regulator, very often controlled by the evaporator temperature.

Theory of cooling processes

The cooling power can be calculated by means of the tabulated enthalpies ‘i’ for cooling working fluids. In general these enthalpies are depending of the state of the fluid, i.e. they vary with pressure and temperatures (see the phase diagram in Appendix 1 as an example of the common fluid R134a). The following theoretical relations are explained by means of Figure 4.5.

The cooling power can be calculated according to equ. 4.3:

$Q_{eva} = \dot{m}\ (i_{ek}\ {}_{-}\ i_s)$ [4.3]
(m is the mass flow in kg/sec, i denotes enthalpies in kJ/kg).

The heat rejected by the condenser can be calculated by a corresponding equation 4.4:

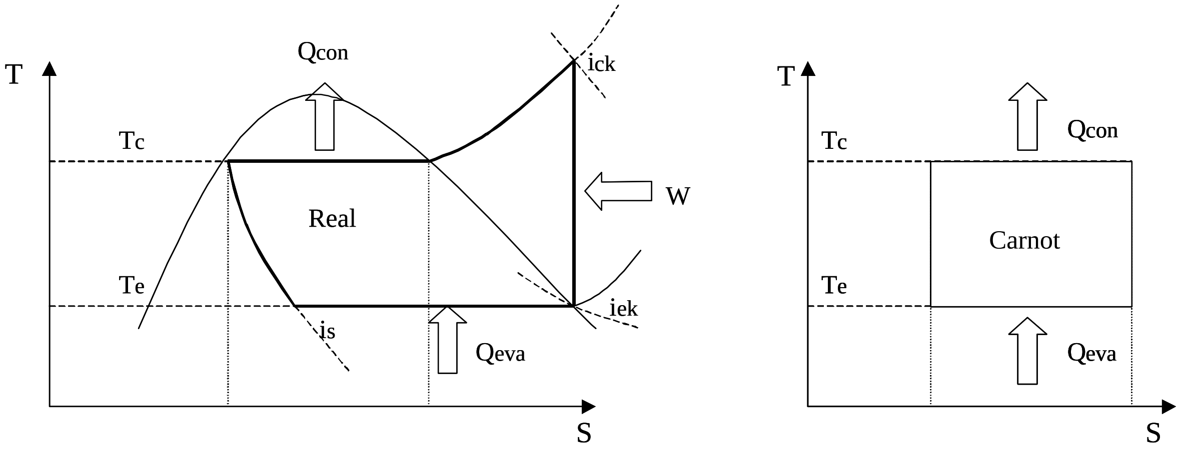


Figure 4.5: Temperature/Entropy diagram for a real and a Carnot compression process.

$Q_{con} = \dot{m}\ (i_{ck} - i_s)$ [4.4]

As described above, the following relationship holds between Q_{con} and Q_{eva}:

$Q_{con} = Q_{eva} + W$

The compression work W is defined by:

$W = \dot{m}\ (i_{ck} - i_{ek})$ [4.5]

The ***cooling factor å*** is then defined by the relation between the delivered cooling power and the work done by the compressor:

å = $Q_{eva}/W = (i_{ek} - i_s) / (i_{ck} - i_{ek})$ [4.6]

If the heat Q_c is used for heating purposes, we also define a ***heat factor Ö*** or ***coefficient of performance COP*** defined as the relation between heat output and supplied compressor work:

Ö = $Q_e/E_c = (i_{ck} - i_s)/ (i_{ck} - i_{ek})$ = å +1 [4.7]

Of interest is also the ***Carnot cooling factor å$_c$.*** The Carnot process is a reversible thermodynamic process consisting of two isentropic and two isothermal lines. Such a process is an idealisation of a thermodynamic process and therefore represents a maximum efficiency. The Carnot cooling factor determines the highest possible process efficiency according to the following definition:

å$_c$ = $T_e/(T_c - T_e)$ [4.8]

Accordingly, the ***Carnot heating factor*** can be defined by:

Ö$_c$ = $T_c/(T_c - T_e)$ [4.9]

In reality, in all thermodynamic cycles we have losses in each of the process curves of the Figure 4.5. That means that the real cooling factor å is much smaller than the Carnot cooling factor. Depending of the size of the system, å might be 50 – 70 % of the ideal Carnot values.

Practice

In the practice, compressor coolers have different designs depending on the application and system size. The compressor can be made as a piston engine, a screw or a spiral (scroll compressor). Also turbo compressors are used. In principle there are very often several compressors connected in parallel because this connection is easier to adapt to the lower partial load range. The load variation can easily vary a factor 20 in certain applications.

Scroll compressors:

This modern spiral-type compressor shows very efficient properties at small powers in the range from 30 to 110 kW. Scroll compressors can reach condenser temperatures of 57C .

Piston compressor:

Piston engines are used for relatively small units, up to ca 300 kW and with a number of parallel compressors up to totally 1 MW. Piston engines can reach condenser temperatures of 60 C (and 80 C in 2-stage versions).

Screw compressors

These are most common compressors for the power range between 100 kW to 4000 kW. For higher powers, usually several compressor units are used in parallel. The maximum allowable condenser temperature is 63C [8]. We assume in our work that the allowable temperature when used for heating purposes is 60C . An example of cooling factors and relative cooling power for screw compressors is shown in Figure 4.6.

Turbo compressors

These ones are normally used for very high powers, between 2 and 50 MW per unit.

Efficieny of compressor systems

From industrial product information, the cooling factors and other technical information can be received. For a screw compressor [11] performance parameters are presented in Figure 4.6. This figure shows the cooling factor and the relative cooling power as a function of the outlet temperature T_{co} of the condenser cooling water (the curves hold for a special type of Carrier Global chiller option 150 in the 1 MW range).

As theoretically expected, (see equ. 4.6) the cooling factor decreases with increasing condenser temperature, here presented by the condenser cooling water. Hence, for a given size of the compressor, also the cooling power decreases with increasing outlet temperatures.

If the compressors are to be run at partial load instead of design load, normally the cooling factor can decrease. At partial loads below 0.2 the factors are getting remarkably lower than the design values. Therefore, very often 2 or 3 parallel compressor units are installed, one of those well adapted to operate in low load conditions.

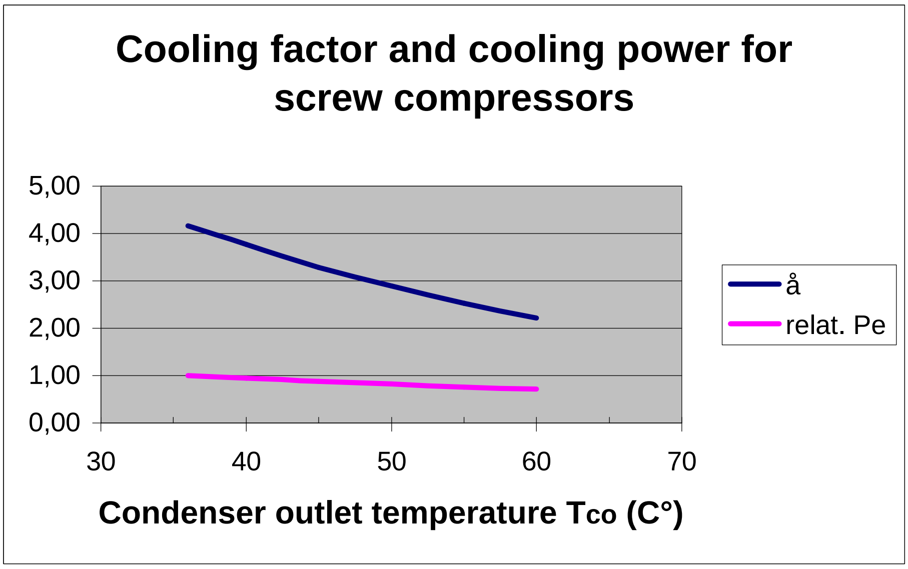


Figure 4.6: *Cooling factor å and cooling power P_e as a function of condenser outlet temperature. P_e is the cooling power relative to the power at T_{co} = 36 C° (from Ref [8]).*

4.2.2 Cooling towers

All kinds of chillers that produce chilled water require a heat sink for their heat rejection and the temperature of the heat sink is crucial in determining the performance of the chillers. In general, the heat sink can be any fluid of which the temperature is lower than that of the cooling water leaving the chiller. Cooling tower is a common term for a device that cools water by rejecting heat to ambient air, which in this case is the ultimate heat sink.

Types and characteristics of cooling towers

Cooling towers can be either of *direct* (open circuit) or *indirect* (closed circuit) type. They can also be of the *wet* or the *dry type*. In the open circuit-cooling tower, the circulating water is in direct contact with air and some of the water is vaporized, the evaporation heat is taken from the medium to be cooled. The water must be filtered to remove the dust and other impurities it collects from the air before recycling. Addition of fresh water, ca. 10%, is required to reduce the concentration of non-evaporating, enriching compounds.

In the closed circuit cooling tower the cooling water is indirectly cooled in a fluid/air heat exchanger, which is cooled by ambient air with enhanced heat transfer through spraying water into the airflow. The efficiency is somewhat lower compared to the open circuit cooling tower. The closed circuit-cooling tower can also be driven in *dry mode*, without spraying any water. In this way, the water consumption and also pumping power will be reduced. In the *wet mode,* the wet bulb temperature is the reference temperature of the heat sink. Because this temperature is some degrees below the ambient air temperature (depending on humidity and ambient temperature), heat pumps and compressor chillers can work more efficiently with wet cooling towers. The air is taken in from the side or the bottom of the tower and exits the tower from the top. The water is directed to fall from top to bottom or to circulate in tubes. Hence the heat exchanges takes place as cross-flow or counter-flow.

Figure 4.7 shows a schematic view of a cooling tower.

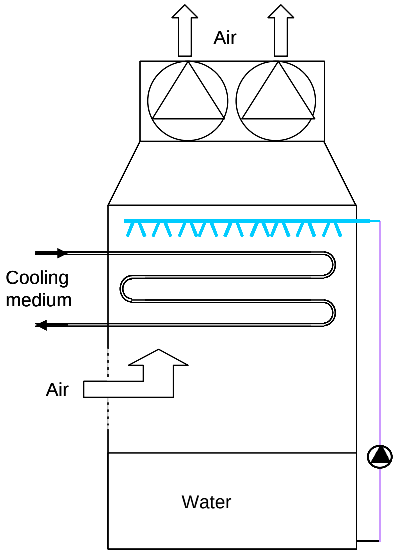


Figure 4.7: Scheme of a cross flow cooling tower with indirect circuit.

The cooling process of a cooling tower in combination with an air cooler

If the temperature of the cooling medium is high, a dry cooling tower and a wet cooling tower can be connected in series. That way the dry cooler with the ambient air as heat sink can bring down the temperature of the cooling medium, let say to about 15°C above the air temperature. If further cooling is necessary, the wet cooling tower will be used. In this study it is assumed that the final temperature of cooling medium leaving the wet cooler is 5°C above the wet bulb temperature.

Cooling towers consume energy for operating fans and pumps. For sake of simplicity, we assume in our study that the electricity consumption is 8 % of the heat to be rejected, which is a typical value for wet cooling towers.

5 The load models

The calculations in this study are performed in *Excel based on hourly values*. As basis for the calculations we have chosen hourly ambient temperature values from Stockholm for the year 1986. The mean values of his year were rather normal, but a rather rainy August was followed by a warm September.

For both the heating and cooling system we assumed an appreciable load that is rather big. The top load powers are in the 10-th of Megawatts (Figure 5.2). Both loads are assumed to depend strongly on the ambient air temperature. Also the district heating supply temperature T_{hs} is assumed to be dependent on the ambient temperature T_a (Figure 5.1).

The following load models have been used for the determination of *the heating and cooling loads,* respectively:

Heating load:

District heating supply temperature:			T_{hs}: Max [80;(80 – T_a)] °C
District heating return temperature:			T_{hr} = 50°C
Heating load:			$P_{h:}$ Max [5;(16 – T_a)] MW

Cooling load:

District cooling supply temperature:			T_{cs}: = 6°C
District cooling return temperature:			T_{hr} = 15°C
Cooling load:	100 %	⇨	$P_{c\text{-}100:}$ Max [3;(T_a -12)] MW
Cooling load:	50 %	⇨	$P_{c\text{-}50:}$ Max [3;(T_a -12)] x 0,5 MW
Cooling load:	25 %	⇨	$P_{c\text{-}25:}$ Max [3;(T_a -12)] x 0,25 MW

As can be seen from the definition above, both the heating load and the cooling loads are depending on the ambient temperature T_a. So is the supply temperature for district heating, whereas the other temperatures are kept constant.

In the base case of the cooling load model (called 100 for 100 % of the base case), the peak load power of the district heating is about half of that of the top load power of the cooling (P_h=36 MW at T_a = -20°C and $P_{c\text{-}100}$ = 18 MW at T_a = +30°C). The summer heating load is then 5 MW and the winter cooling load is 3 MW. In order study the situation with smaller cooling loads, the cooling load curve is s also multiplied by reduction factors 0,5 and 0,25, respectively, keeping the heating load the same and hence letting the load situation approach more closely to the market realities in Sweden.

The load curves can be seen from Figure 5.2.

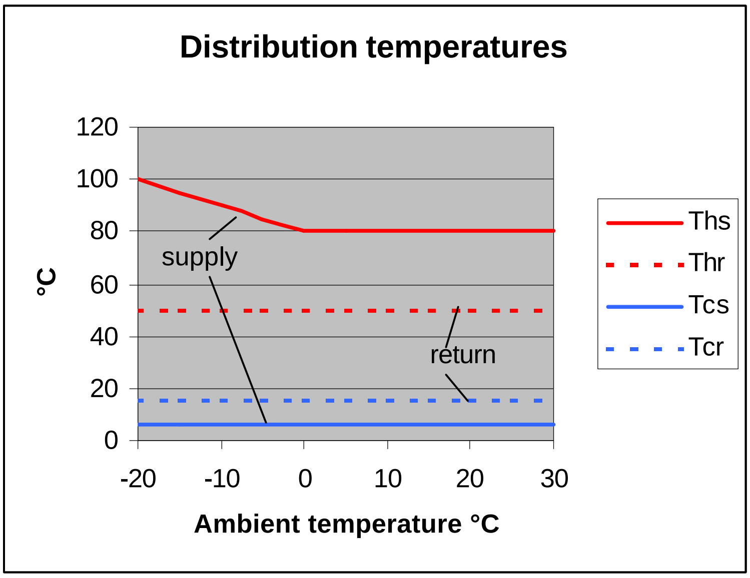


Figure 5.1: *District heating and cooling – supply and return temperatures.*

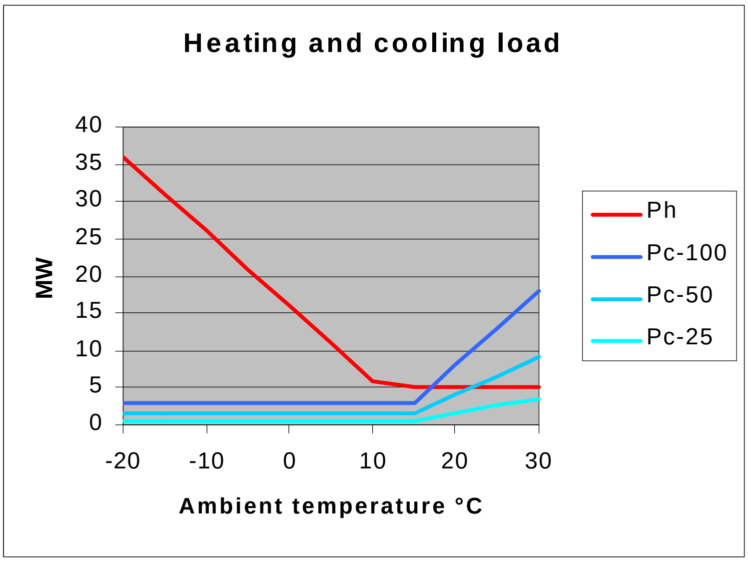


Figure 5.2: *District heating and cooling loads: Pc-50 and Pc-25 denotes 50 and 25 %, respectively, of the base load case Pc-100.*

From Figure 5.2 can be derived that extreme load situation can be expected during the summer time, when the cooling loads are larger than the heating loads. In this cases, a heat pump cannot much the both types of loads and hence an extra cooling device must be operated (The corresponding situation holds for the winter conditions were the heating plant supplies most of the district heat).

In a three-pipe system, the mixed return temperature T_r is a function of the load relation between heating and cooling circuit. Figure 5.3 shows T_r as a function of the ambient temperature for the three load cases. The plateau occurring between T_a = 12 – 17 °C is an artefact depending on our load models.

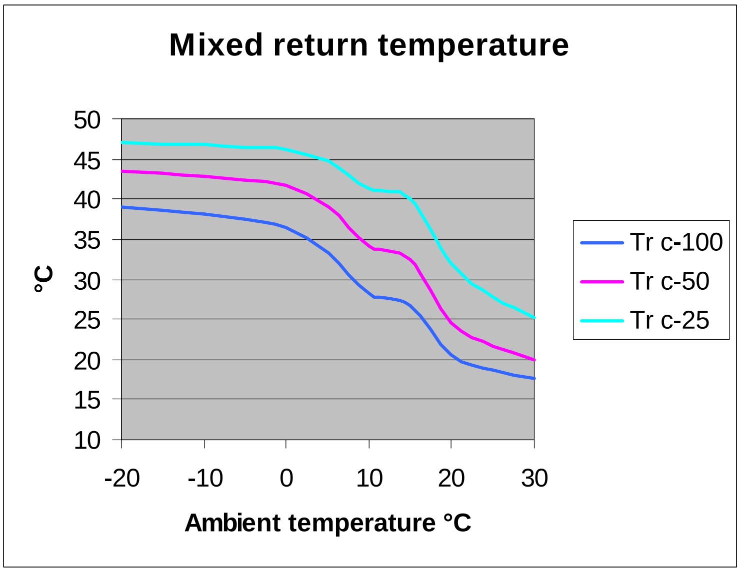


Figure 5.3: Mixed return temperature as a function of ambient temperature.

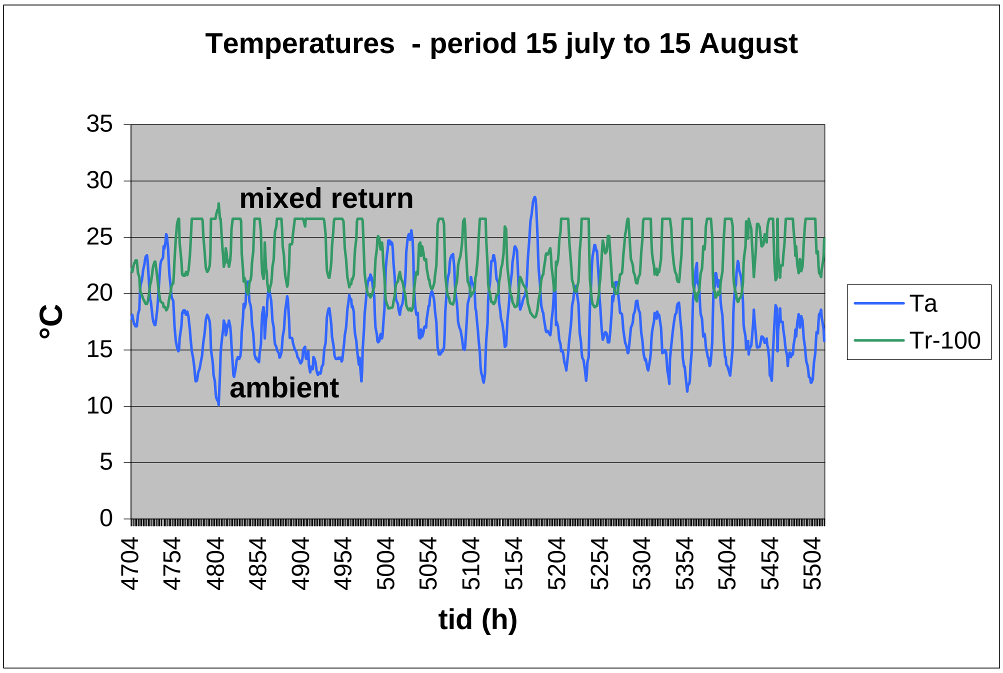


Figure 5.4: Mixed return temperature P_{c-100} and ambient temperature under a summer period

According to the model, in which the load is a strong function of the ambient temperature, the mixed return temperature can vary quite a lot over the day. This is shown in Figure 5.4 for a summer period. Ambient temperature T_a and return temperature T_{r-100} are presented.

6 Description of three district heating and cooling models

In order to evaluate the feasibility of a three-pipe system in comparison with four-pipe systems, we investigated three basic solutions for such arrangements. The task was to supply *all the heating and cooling demands* according to the load definition shown in chpt. 5. In the following chapters, these models are described in detail, for each model presenting a 3P and a 4P solution. It should be mentioned here that we in the Models 2 and 3 for all solutions have chosen systems were interaction between the district heating and district cooling circuits occurs, i.e. that heat rejected by the cooling system, whenever possible, is transferred to the district heating system. The investigated systems are:

Model 1 4P: Two separated systems
3P: Systems interacting through heat exchangers

Model 2: 4P: Systems interacting through heat pump
3P: Systems interacting through heat pump

Model 3: 4P: Systems interacting through heat pump
3P: Systems interacting through heat pump and cooling tower

6.1 Model 1

6.1.1 Separated four-pipe system 4P-Model 1

The schematic diagram of four-pipe of Model 1 is shown in Figure 6.1. The systems are simplified showing only the main features of interest for the system analysis.

The system is made up of two independent distribution circuits. The *heating circuit* consists of a *heating plant* (biomass boiler) and a *heating load*. The *cooling circuit* consists of three components, *compression chille*r with cooling supplied by a *cooling tower* and the *cooling load.*

The *hot water for district heating* with a supply temperature of $T_{hs} \geq 80°C$, which is a function of the ambient temperatures and variable over the time, is prepared by the heating plant, then flowing along the distribution network to the consumer stations and back to the heating plant. In the consumer stations (functioning as a heat exchanger with the fluid being cooled by delivering energy), the return temperature T_{hr} is assumed to be fixed at a temperature of 50°C.

The *cold water* for district cooling is supplied from chiller at a temperature of 6 °C and then flowing along the distribution pipes to the substations connecting the cooling load, where it will discharge its cooling power and therefore be heated up to a return temperature of T_{cr} = 15°C, based on the demand of the cooling load. After that, it will flow back to the chiller. The condensation heat from the condenser of the chiller is rejected by an indirect cooling tower, which finally releases the heat to the ambient air. The condenser power is equal to the cooling power of the evaporator plus the electrical power of the compressor.

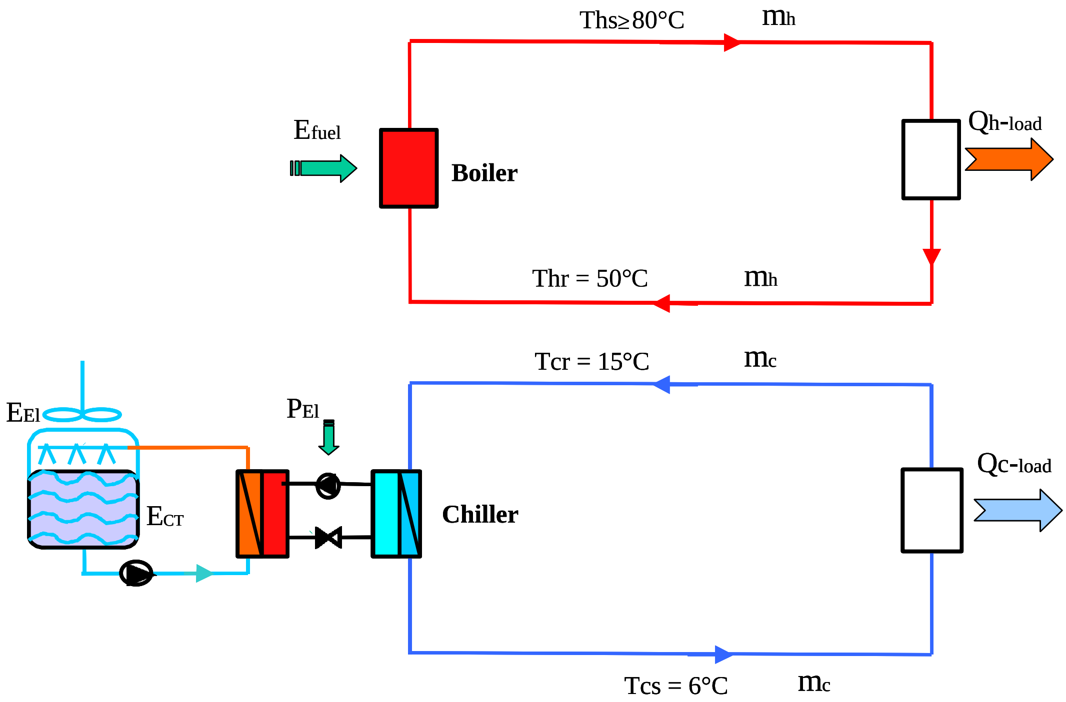


Figure 6.1: Basic scheme for four-pipe system, 4P-Model 1.

6.1.2 Combined three-pipe system 3P-Model 1

The schematic diagram of the basic three-pipe system is shown in Figure 6.2. The *heating circuit* again basically consists of the *boiler* and the *heat load*. However, a *heat exchange*r for pre-heating enables to transfer condenser heat from the *cooling tower* circuit. That means, that if the condenser outlet temperature is higher than the common return temperature T_r, reject heat can be used for preheating the heating circuit.

The *cooling circuit includes* a *chiller*, a *water-heat exchanger* for pre-cooling, and the *cooling tower*. It is assumed that the compression chiller in the most simple design delivers a condenser outlet temperature T_{co} of 40°C.

The heating mass flow will be mixed with the cooling mass flow in the common return pipe to form a common mass flow with a temperature between 15°C and 50°C, depending on the load conditions see chpt.5. In networks where the top loads for heating and cooling are comparable, T_r will be close to the heating return in the winter and close to the cooling return in the summer as shown in Figure 5.3. At the site of the heating and cooling plant, the flow will again be split up according to the individual heating and cooling loads.

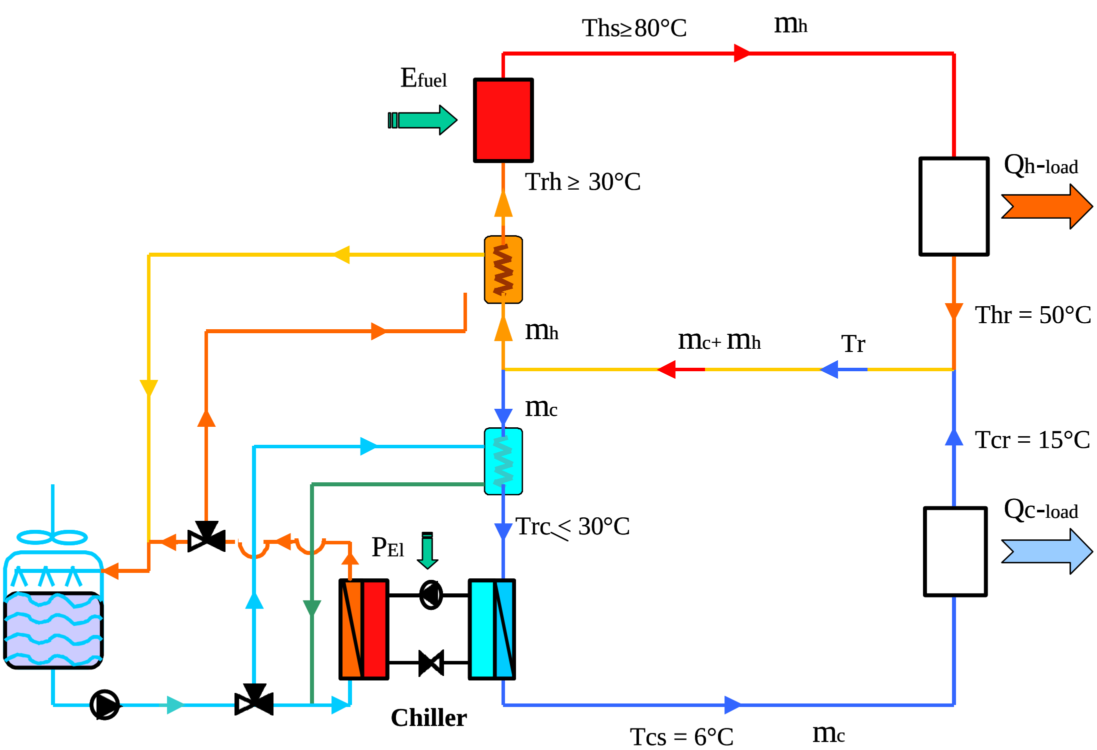


Figure 6.2: Basic scheme for three-pipe system, 3P-Model 1.

In order to make the system energetically more interesting (energy supplied by heating is to be rejected by a chiller), some heat recovery is incorporated in this scheme.

If the common return temperature T_r is less than 30°C, the heating mass flow will be *pre-heated* to 30°C by means of the heat produced in the condenser of the chiller at T_{co} = 40°C. At T_r>30°C, preheating is not applicable and all condenser heat is rejected. Instead another function is applied, namely *precooling* of the mass flow returning to the chiller. Part of the mass flow circulating through the cooling tower is used to bring down the return temperature of the cooling to at least 30°C.

.

The cooling mass flow is finally returned to the chiller for producing chilled water at a supply temperature of T_{cs} = 6°C.

The cooling circuit between condenser and cooling tower therefore has three functions:

- Rejecting the condensation heat from the condenser of the compressor chiller
- Pre-cooling the cooling mass flow if necessary
- Supplying preheating to the district heating circuit.

6.2 Model 2

In this Model 2, the district heating circuit and district cooling circuit are interconnected by means of a heat pump, generating as much heat as possible with the cooling circuit as the low temperature heat source in order to preheat the district heating system. A two-stage heat pump with a heat delivery temperature of 80°C is assumed. When enough heat is available in the cooling circuit, the preheating temperature is assumed to be 80°C.

6.2.1 Combined four-pipe distribution system 4P-Model 2

The schematic diagram of the four-pipe system of Model 2 is shown in Figure 6.3.

In this Model 2, the district heating circuit and district cooling circuit are interconnected by means of a heat pump - similar to the three-pipe system shown in the next chapter. The heat pump will generate as much heat as available with the cooling circuit as low temperature heat source in order to preheat the district heating system. However, the amount of energy to be transferred is limited: *In winter time by the amount of cooling load from which the low temperature energy can be taken, using the wood chip boiler as the supplementary heater. In the summer time the limitation is due to the limited heating requirement in the heating circuit.* In this case extra cooling must be supplied by a chiller.

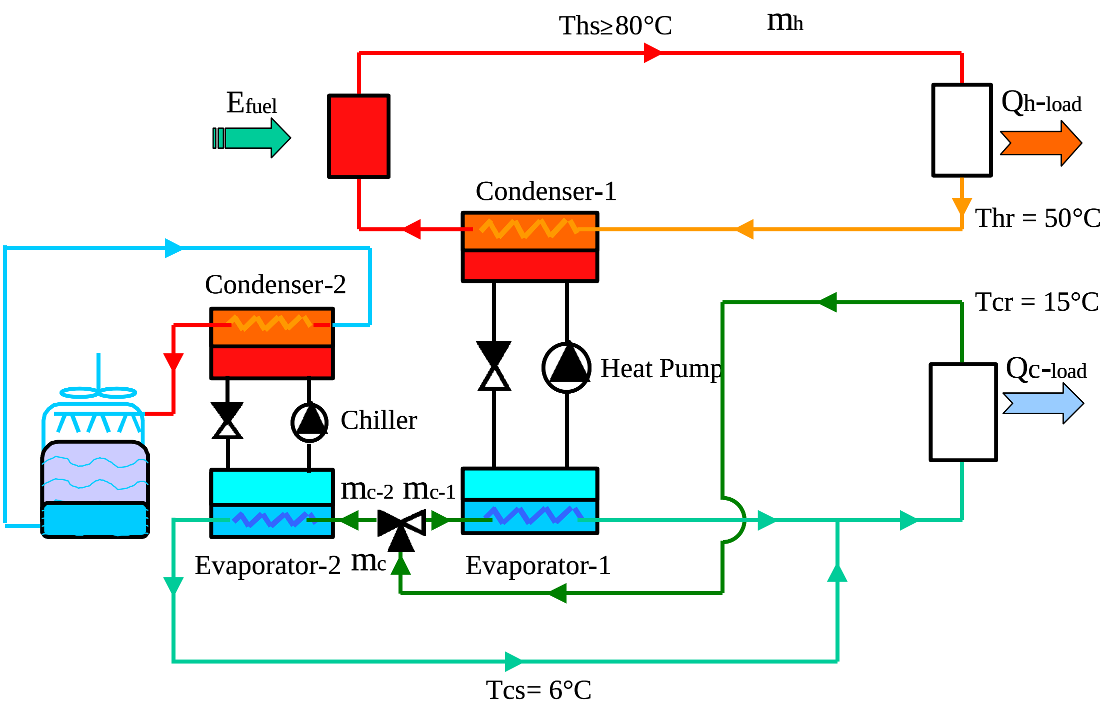


Figure 6.3: Scheme of combined four-pipe system, 4P-Model 2.

The heating circuit is similar that of Model 1. However, a heat pump condenser is used for preheating the return flow with energy taken from the cooling circuit. The preheating temperature is 80°C whenever the energy output of the heat pump is larger than the preheating load. Otherwise, the condenser outlet temperature T_{co} is optimised to deliver lower temperatures corresponding to the energy available from the heat pump in relation to the necessary preheating load. This temperature adjustment is performed in an iterative calculation. The maximum condensation outlet temperatures in the winter are 60°C.

In the cooling circuit, the cold water is prepared by both the heat source of the heat pump evaporator and an additional chiller working in parallel with the heat pump. The excess heat from the chiller is rejected in a cooling tower.

6.2.2 Combined three-pipe distribution system 3P-Model 2

The schematic diagram of the three-pipe of Model 2 is shown Figure 6.4.

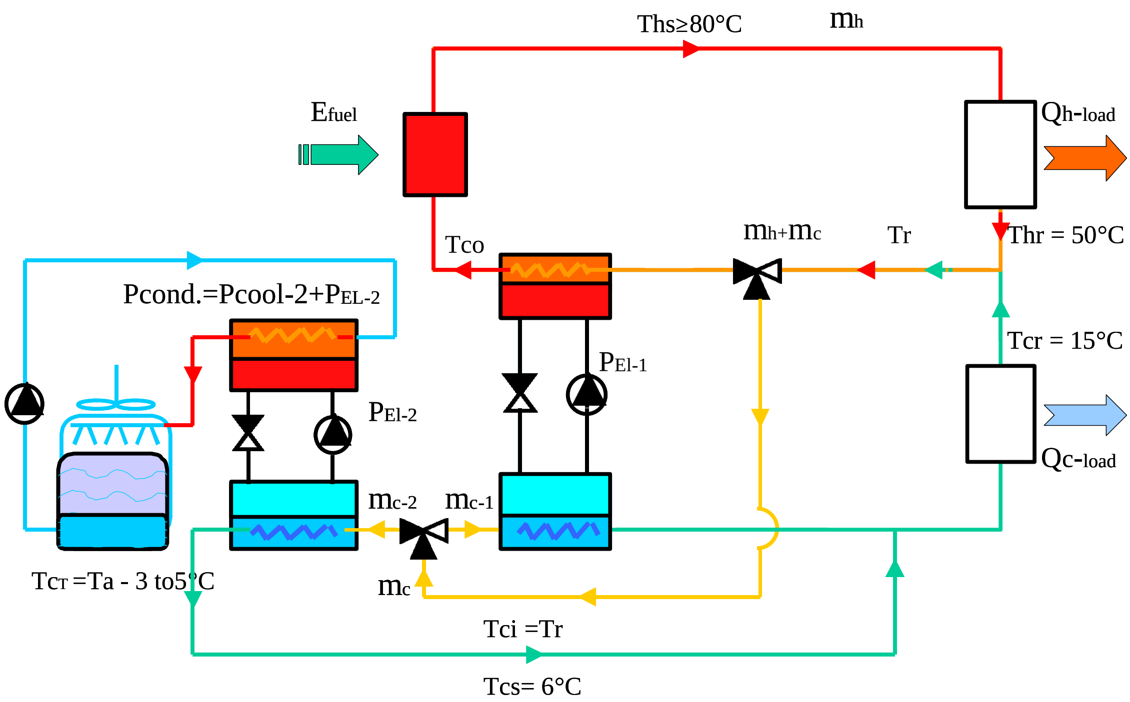


Figure 6.4: Scheme of combined three-pipe system, 3P-Model 2.

The three-pipe system is basically similar to the four-pipe system described above.The heat pump works on the cooling circuit and delivers heat to the heating return pipe with a continuously varying temperature depending on the load situation (which in turn is depending on the ambient air temperature).

The common return flow splits up into two parts, one flow, m_h, corresponding to the heating load, and another flow, m_c, corresponding to the cooling load. The heat pump is continuously adjusted to deliver the maximum possible power depending on the load situation. In *winter time* the heat load is large enough to consume the necessary cooling power. In *summer time*, only part of the cooling load can be transferred to the condenser pre-heater, and therefore, an extra chiller has to be used for supplying the remaining cooling load. For this reason the cooling mass flow is split up into two parts, $m_{c\text{-}1}$ and $m_{c\text{-}2}$, respectively.

Whereas the heat pump condenser works at variable temperatures between 60 and 80°C, the outlet temperature of the chiller condenser is assumed to be 40°C, a temperature quite common in commercial chiller systems.

Compared to the system of Model 1, the system of Model 2 will save substantial amount of biomass fuel at the expenses of consuming more electrical energy, because the heat pump condenser can deliver heating energy at temperatures of 80°C during quite a long summer period over the year. However, this means practically that electrical work is put into the heating circuit.

6.3 Model 3

Model 3 is quite similar to Model 2, but an extra cooling tower is added for cooling the cooling mass flow before it enters the evaporators of the heat pump and the chiller. The main idea for this design is to save heat pump electricity at the cost of biomass fuel, the former being assumed to be less expensive than the latter.

6.3.1 Combined four-pipe distribution system 4P-Model 3

The scheme of the four-pipe system for Model 3 is the same as for Model 2, Figure 6.3.

6.3.2 Combined three-pipe distribution system 3P-Model 3

The schematic diagram of three-pipe of Model 3 is shown Figure 6.5.

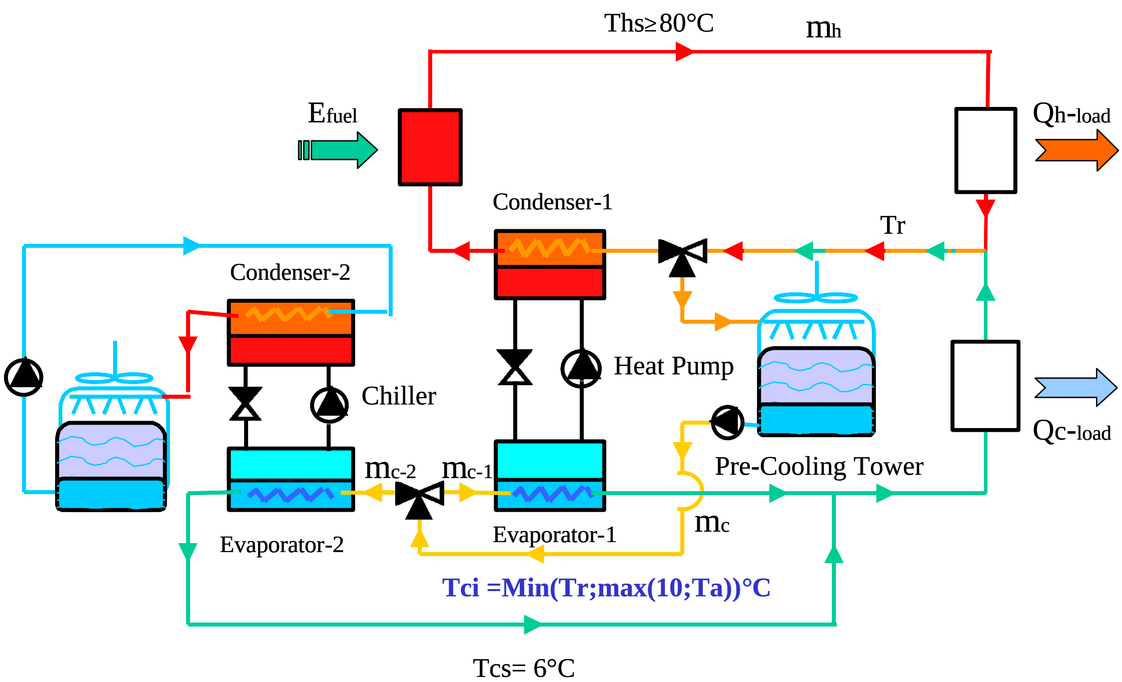


Figure 6.5: Scheme of combined three-pipe system, 3P - Model 3.

From Figure 6.5 the location of the extra cooling tower can be seen. This is an effective component to reduce the mixed return temperature T_r as much as possible, depending on the ambient temperature, with a low electricity consumption. The lower limit of this precooling temperature is chosen to T_{ref}.

Some thoughts must be given to the pre-cooler outlet temperature (= evaporator inlet temperature T_{ci}). This temperature is a delicate choice of being too low (and therefore making a larger heat pump power necessary) or being too high and infering a larger chiller power. The right temperature T_{ci} was chosen in an iterative calculation process and could be approximated by:

$$T_{ci} = \mathrm{Min}\ [T_{r;}\ \max\ (T_{ref};T_a)] \qquad [6.1]$$

T_{ref}, i.e. the optimal minimum evaporator inlet temperature was found to be 10°C.

The system of 3P-Model 3 was found to use much more fuel, but less electricity than the corresponding system 3P-Model 2. Under some circumstances it turned out to be the most cost effective 3P system.

7 Theoretical calculation of the different system models

In this chapter, some fundamental theoretical relations are presented in order to facilitate the discussion of the system analysis in the following chapters.

7.1 Model 1

7.1.1 Four-pipe system 4P-Model 1

The four-pipe system diagram of Model1 is shown in figure 6.1.

Heating circuit

The mass flow rate $\dot{m}_{h:}$*is derived:*

$$\dot{m}_h = \frac{P_{h-load}}{C_p * (T_{hs} - T_{hr})} * 1000 \quad \text{(kg/sec)} \qquad [7.1]$$

Where P_{h-load} in MW, C_p in kJ/(kg °C), T in °C and $\dot{m}_h$ in kg/s.

The heating boiler uses biomass fuel and stack gas condensation is applied as shown in chapter 4. The Heat recovery efficiency η is presented in Table 4.1 and equation 4.1.

In the four- pipe system T_r =50°C (constant) and hence the

consumption of biomass Q_{fuel} is:

$$Q_{fuel} = \frac{E_{fuel}}{(1+h)} = \frac{E_{fuel}}{1{,}17} \qquad [7.2]$$

Where $E_{fuel} = Q_{load}$ is the amount of fuel needed at the upper heating value of the biomass.

Cooling circuit

mass flowrates of the chiller $\dot{m}_c$*:*

$$\dot{m}_c = \frac{P_{c-load}}{C_p * (T_{cs} - T_{cr})} * 1000 \qquad [7.3]$$

Where P_{c-load} in MW, C_p in kJ/(kg°C), T in °C and $\dot{m}_c$ in kg/s.

The cooling factor of the chiller:

The cooling medium temperature at the condenser outlet is assumed to be 40°C and the evaporator outlet temperature at 6°C, and therefore the temperatures of the working medium of the chiller are T_{con} = 45°C (condenser) and T_{eva} = 1°C (evaporator) respectively. With the internal chiller efficiency being 0,6 the following chiller cooling factor ε_c can be calculated:

$$e_c = k * \frac{T_{eva}}{T_{con} - T_{eva}} = 0{,}6 * \frac{(273+1)}{(273+45-(273+1))} = 3{,}74 \qquad [7.4]$$

Electrical work of the chiller:

$$Q_{el} = \frac{Q_{eva-chiller}}{e_c}$$

And the condenser energy cooled away by the cooling tower connected to the chiller:

$$Q_{con-chiller} = Q_{el-Chiller} + Q_{eva-chiller} \qquad [7.5]$$

Where $Q_{eva-chiller} = Q_{c-load}$ is the cooling load.

7.1.2 Three-Pipe system 3P-Model 1

A three-pipe scheme of Model 1 is shown in figure 6.2. Besides the items mentioned above, the following new features are typical for the system:

Heating circuit:

Mixed return temperature T_r:

The medium temperature in the combined return pipe is a temperature weighted mixture of the mass flows in the heating and the cooling circuits, respectively. This temperature T_r can be calculated according to equ. 7.6:

$$T_r = \frac{(\dot{m}_h * T_{hr} + \dot{m}_c * T_{cr})}{(\dot{m}_h + \dot{m}_c)} \qquad [7.6]$$

Heating energy provided by the preheating heat exchanger to the heating mass flow:

$$Q_{h\text{-}ex} = \dot{m}_h * C_p * (T_{hr} - T_r)\,\Delta t \qquad [7.7]$$

The energy provided by the biomass plant is:

$$E_{fuel} = \dot{m}_h * C_p * (T_{hs} - Th_r)\,\Delta t \qquad [7.8]$$

Where: $Th_r = T_r$, when $T_r \geq 30$°C, or 30°C, when $T_r < 30$°C.

Taking into consideration the heat recovery efficiency η of the stack gas condenser, the fuel consumption of biomass heating plant becomes (Δt = 1 h):

$$Q_{fuel} = \frac{E_{fuel}}{(1+h)} = \frac{\dot{m}_h * C_p * (T_{hs} - T_{rh}) * \Delta t}{(1 - 0{,}0004 * T_r^{\ 3} + 0{,}0327 * T_r^{\ 2} - 1{,}0489 * T_r + 37{,}914)}$$

[7.9].

Cooling circuit:

Energy provided by the cooling heat exchanger for pre-cooling the cooling mass flow:

$$Q_{c\text{-}ex} = \dot{m}_c * C_p * (T_r - T_{rc}) \qquad [7.10]$$

The cooling energy provided by the chiller is:

$$Q_{cool\text{-}chiller} = \dot{m}_c * C_p * (T_{rc} - T_{cr}) \qquad [7.11]$$

Where, Th_r = T_r, when $T_r \leq 30$°C, or 30°C, when $T_r > 30$ °C.

Electrical work of the chiller:

$$Q_{el-chiller} = \frac{Q_{cool-chiller}}{e_c} \qquad [7.12]$$

And the cooling energy cooled away by the cooling tower of the chiller:

$$Q_{con-chiller} = Q_{el-Chiller} + Q_{eva-chiller}$$

The cooling factor ε_c of the chiller is again calculated according to equ. 7.4 , and found to be ε_c = 3.74.

7.2 Model 2 and Model 3

In both models, the calculations of the heating and cooling mass flows, the common return water temperature T_r, the cooling factor of the chiller and the recovering efficiency of biomass stack gas condensation heat of the heating boiler are the same as those in Model 1.

The essential part of calculation is to find out the optimised hot water temperature T_{co} leaving the condenser outlet of the heat pump after being pre-heated. The control of the right condenser outlet temperature will improve the COP of the heat pump and at the same time make full use of the disponible from heat pump.

7.2.1 The four-pipe system

Model 2 and Model 3 use the same four-pipe scheme, see Figure 6.3.

The concept of optimisation of the condenser outlet temperature T_{co} is to precisely match it with the mass flow into the evaporator of the heat pump.

In winter, the cooling mass flow into the evaporator of the heat pump will only deliver a small amount of the heating load. Therefore we adjust the condenser temperature of the heat pump to an adequate lower level T_{co-try} according to the energy the heat pump condenser would deliver at T_{co-try}, thus improving the heat pump COP compared to the case that the heat pump would deliver the energy at a fixed T_{co}. This is done in an iteration process.

In summer, there is enough cooling load available for producing all heat at for providing the summer heating load, and hence the heat pump is operated to an condenser outlet temperature of 80°C. Therefore the following calculation steps are applied:

Step1:

Assume a guessed value for $T_{co-try-gu}$ and find out the relationship among it and other parameters

The energy delivered to the heating mass flow from the condenser of the heat pump is:

$$Q_{cond} = \dot{m}_h * C_p * (T_{co-try-gu} - T_{hr}) \quad [7.13]$$

The energy taken up by the evaporator of the heat pump from the cooling mass flow is:

$$Q_{eva} = \dot{m}_c * C_p * (T_{ci} - T_{cs}) \quad [7.14]$$

The instant COP of the heat pump is:

$$e_{HP} = k * \frac{T_{eva}}{(T_{con} - T_{eva})} = k * \frac{273 + T_{eva}}{(T_{co-try} + 5 - T_{eva})} \quad [7.15]$$

Hence the energy delivered to the condenser of the heat pump is:

$$Q_{cond} = Q_{el} + Q_{eva} = \dot{m}_c * C_p * (T_{ci} - T_{cs}) * (1 + \frac{1}{e_{HP}}) \quad [7.16]$$

By making use of the equation 7.13 – 7.14 we can find the new condenser outlet temperature T_{co-try} :

$$T_{co-try} = \frac{(273 + T_{eva}) * k * \dot{m}_h * T_{hr} + ((273 + T_{eva}) * k + 1) * \dot{m}_c * (T_{ci} - T_{cs})}{(273 + T_{eva}) * k * \dot{m}_h - \dot{m}_c * (T_{ci} - T_{cs})} \quad [7.17]$$

Substituting the parameters: $k = 0{,}6$, $T_{eva} = 1^\circ C$, $T_{hr} = 50^\circ C$, $T_{ci} = T_{cr} = 15^\circ C$ and $T_{cs} = 6^\circ C$, we receive the following equation for the adjusted outlet temperature:

$$T_{co-try} = \frac{8220 * \dot{m}_h + 1488{,}6 * \dot{m}_c}{164{,}4 * \dot{m}_h - 9 * \dot{m}_c} \qquad [7.18]$$

Because we know the hourly values of the heating mass flow m_h and cooling mass flow m_c, we can get the values of $T_{co\text{-}try}$ every hour. After that we will have to compare this $T_{co\text{-}try}$ with the highest desired temperature T_{co} 80°C. The final value will be selected according to the condition [7.19]:

$$T_{co} = \min[80; T_{co-try}]^\circ C \qquad [7.19]$$

Final preheating energy delivered by the heat pump is found to be:

$$Q_{cond} = \dot{m}_h * C_p * (T_{co} - T_{hr}) \qquad [7.20]$$

The *mass flow* $\dot{m}_{c-1}$, actually flowing into the evaporator of the heat pump is given by:

$$\dot{m}_{c-1} = \frac{\dot{m}_h * (T_{co} - T_{hr})}{(T_{ci} - T_{cs}) * (1 + \frac{1}{e_{HP}})} \qquad [7.21]$$

The *mass flowrates* $\dot{m}_{c-2}$ flowing into the evaporator of the chiller,
with $\dot{m}_{c-2} = \dot{m}_c - \dot{m}_{c-1}$.

Step 2: Heating circuit

$$E_{fuel} = \dot{m}_h * C_p * (T_{hs} - T_{co}) \qquad [7.22]$$

The supply temperature of the hot water is defined as following:

$$T_{hs} = \max[80; (80 - T_a)]^\circ C \qquad [7.23]$$

$$Q_{fuel} = \frac{E_{fuel}}{(1 + h)} = \frac{E_{fuel}}{1{,}17} \qquad [7.24]$$

Where η is the recovering efficiency of stack gas condensation heat of biomass.

Step 3: Cooling circuit.

Heat Pump

The evaporation energy and compressor's electricity consumption of the heat pump are respectively:

$$Q_{eva} = \dot{m}_c * C_p * (T_{cr} - T_{cs}) \qquad [7.25]$$

$$Q_{EL-HP} = \frac{Q_{eva}}{e_{HP}} \qquad [7.26]$$

Chiller

The energy input to the evaporator and the compressor work are respectively:

$$Q_{cooler} = \dot{m}_c * C_p * (T_{cr} - T_{cs}) \qquad [7.27]$$

$$Q_{EL-chiller} = \frac{Q_{cooler}}{e_{chiller}} = \frac{Q_{cooler}}{3{,}74} \quad . \qquad [7.28]$$

The chillers condenser output to be rejected by the cooling tower is the sum of Q_{cooler} and $Q_{EL\text{-}chiller}$.

7.2.2 The three-pipe system of Model 2 and Model 3

The three-pipe system of Model 3 is shown again in Fig. 7.1. The basic calculation method is similar to that of the four-pipe system. However, there is a difference in two parameters, the cooling flow temperature T_{ci} flowing into the evaporators of the heat pump and the chiller, which is not longer constant as it is in the four-pipe system. Instead, an extra cooling tower is used for precooling the cooling return flow before entering the evaporators of the heat pump and the chiller, respectively. There was no suitable way to analytically calculate the optimal outlet temperature T_{ci} of the cooling tower, therefore T_{ci} was varied at 7°C, 10°C, 15°C and 20°C, in order to observe the final results after finishing all of the calculations. The optimum value for T_{ci} was then found to be 10°C.

Cooling tower outlet temperature T_{ci}

After carefully considering industrial practice and optimisation, we define T_{ci} as following:

$$T_{ci} = \min[T_r; \max(T_{ref}; T_a)]^\circ C \qquad [7.29]$$

with T_{ref} chosen finally to be 10°C.

In Model 2, this temperature T_{ci} is instead equal T_r.

The energy calculation of the pre-cooling tower can then be calculated as:

$$Q_{ct} = \dot{m}_c * C_p * (T_{ci} - T_r) \quad [7.30]$$

The other procedures of calculation are the same as those for the four-pipe system.

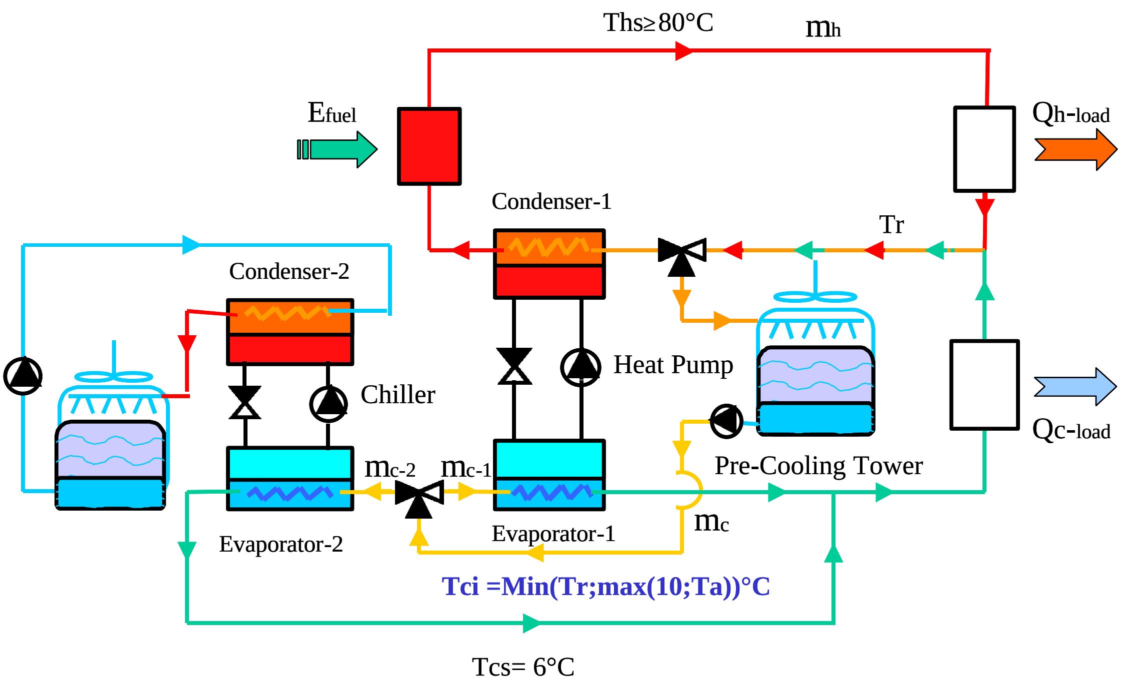


Figure 7.1: Scheme of combined three-pipe system, 3P-Model 3.

7.2.3 General calculation procedure

The six models mentioned above were simulated on an hourly basis. The calculus was built up completely on the models discussed above, i.e. the load model, the three basic system models, both for four-pipe and the three-pipe schemes. Also the cooling load was varied with 100, 50 and 25 % of the base case. All the calculations were performed for hourly values of the ambient temperature, which is the principal parameter on which the models are build up.

In Table 7.1, an example of the parameters is given for 5 consecutive hours 58 – 62 in July.

Table 7.1: Example of hourly calculation parameters for calculations of Model 3 based on hourly varying ambient temperature.

Common Parameters						
Hour nr. in July		58	59	60	61	62
T ambient	°C	24,5	25,4	26,3	27,1	28
$P_{h\text{-}load}$	MW	5	5	5	5	5
$P_{c\text{-}load\ 100\%}$	MW	12,5	13,4	14,3	15,1	16
$T_{h\ supply}$	°C	80	80	80	80	80
m_h	kg/sec	39,87	39,87	39,87	39,87	39,87
$m_{c\text{-}100}$	kg/sec	332,27	356,19	380,12	401,38	425,31
$T_{r\text{-}100}$	°C	18,75	18,52	18,32	18,16	18
$T_{ci\text{-}100}$	°C	18,75	18,52	18,32	18,16	18
3P, 100% cooling load						
T_{co}	°C	80	80	80	80	80
ε cool		1,96	1,96	1,96	1,96	1,96
$P_{cool\text{-}HP}$	MW	6,76	6,78	6,8	6,82	6,84
$P_{EL\text{-}HP}$	MW	3,45	3,46	3,48	3,49	3,49
$P_{cond\text{-}HP}$	MW	10,21	10,25	10,28	10,31	10,33
mc-1	kg/sec	126,77	129,54	132,08	134,17	136,34
E_{fuel}	MW	0	0	0	0	0
mc-2	kg/sec	205,5	226,65	248,04	267,22	288,96
$P_{cool\text{-}chiller}$	MW	10,95	11,86	12,78	13,59	14,49
$P_{EL\text{-}chiller}$	MW	2,93	3,18	3,42	3,64	3,88
$P_{cool\ tow\text{-}chiller}$	MW	13,88	15,04	16,2	17,22	18,37
$P_{cool\ tow\text{-}main}$	MW	0	0	0	0	0
$P_{EL\text{-}total}$	MW	6,38	6,64	6,9	7,12	7,37
4P, 100% cooling load						
$T_{co\text{-}100}$	°C	80	80	80	80	80
ε cool		1,96	1,96	1,96	1,96	1,96
$P_{cool\text{-}HP}$	MW	3,31	3,31	3,31	3,31	3,31
$P_{EL\text{-}HP}$	MW	1,69	1,69	1,69	1,69	1,69
$P_{cond\text{-}HP}$	MW	5	5	5	5	5
mc-1	kg/sec	87,96	87,96	87,96	87,96	87,96
E_{fuel}	MW	0	0	0	0	0
mc-2	kg/sec	244,31	268,23	292,15	313,42	337,34
$P_{cool\text{-}chiller}$	MW	9,19	10,09	10,99	11,79	12,69
$P_{EL\text{-}chiller}$	MW	2,43	2,66	2,9	3,11	3,35
$P_{ct\text{-}chiller}$	MW	11,62	12,76	13,89	14,9	16,04
$P_{EL\text{-}total}$	MW	4,12	4,36	4,59	4,8	5,04
Difference (4-pipe - 3-pipe)						
Q_{fuel}	MW	0	0	0	0	0
$P_{EL\text{-}total}$	MW	-2,27	-2,28	-2,3	-2,32	-2,33
$P_{ct\text{-}total}$	MW	-2,27	-2,28	-2,3	-2,32	-2,33

The presentation of the results of the system analysis, however, is for the sake of simplicity based on the monthly averages and the annual sums of energy values and other important parameters. The results of this analysis are presented and discussed in the next chapter.

8 Results from Energy Analysis

8.1 Cooling load 50 %

The ***cooling load 50 %*** is chosen as a reference case. In this load case, the maximum heating load (36 MW) is 4 times the maximum cooling load (9MW).

8.1.1 Model 1

The first series of Figures 8.1 – 8.3 belongs to ***4P-Model 1*** serving as a reference case. As described in Chapter 7, in this Model 1 the heating systems are completely separated in the 4P system and connected via a heat pump in the 3P system. The Figures show the monthly energy flows (based on monthly 1-hour average values).

Observe: ***The diagrams in the Figures are based on the monthly energy sums divided by the number of the hours per month, i.e. the diagrams present the monthly average power of each item.***

Figure 8.1 presents the *basic* ***reference system*** with which all other models should be compared. The Figure shows the heating load as well as the cooling load, supplied fuel and chiller electricity (compressor electricity only; not included is the electricity consumed by the cooling tower). Also the heat rejected by the cooling tower is shown. Absolute values for the annual sums of these energies are presented in Table 8.1

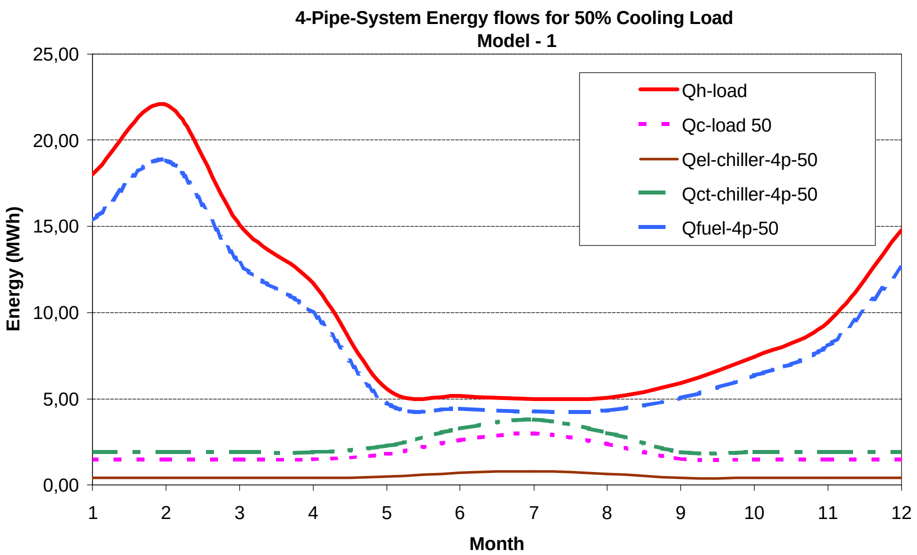


Figure 8.1: Energy flows in the 4P-Model 1 base case.

Here the supplied fuel follows the heat load all the year round with some reduction because of the recovered heat in the stack gas condenser, saving about 15 % energy. The heat rejected by the cooling tower is higher than the cooling load due to the fact that the compressor work also has to be rejected by the cooling tower. The annual mean COP of the chiller is 3,43. Hence 77,5 GWh biomass fuel and 5,9 GWh electricity has to be supplied annually to the heating and cooling systems (see Table 8.1).

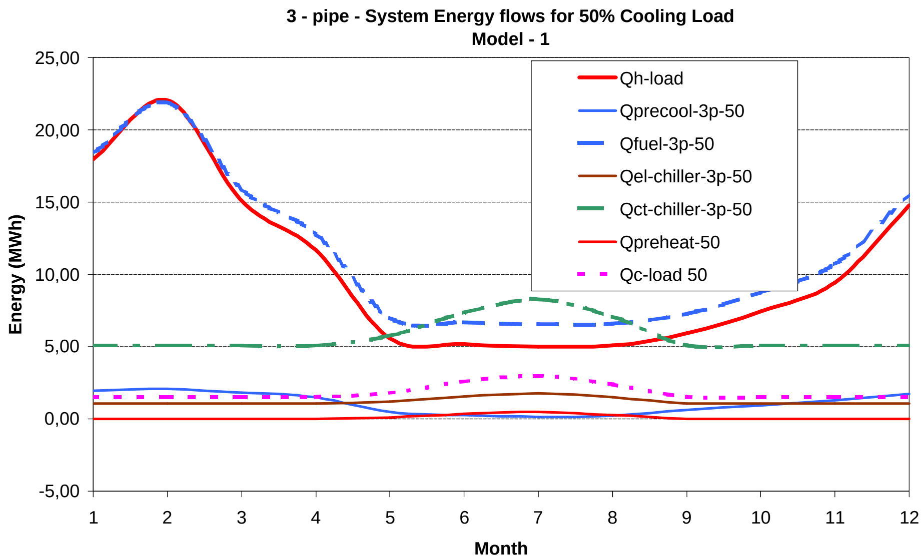


Figure 8.2: Energy flows for 3P-Model 1.

Figure 8.2 shows the energy flows for the 3P version of Model 1. In this model, the return pipe is common for both the heating and the cooling system. Preheating and precooling is supplied by heat exchangers connected to the cooling tower circuit, supplying maximum temperatures of 30 C to the cooling system and minimum temperatures of 30 C to the heating system, respectively. A compressor cooler working at 40 C condenser tempe - rature supplies the cooling.

This system P3-Model 1 is the simplest 3-pipe system, but also the most uneconomic one of the investigated models. The reason is that a large amount of energy is transferred from the heating circuit to the cooling circuit without being recovered. Hence the amount of supplied biomass fuel is 30 % higher than in the 4P system. This heat must be cooled away by a cooling system that requires about three times more capacity than in the 4P system. This is especially true in the winter, when the return temperature is relatively high due to high heating and low cooling load, as shown in Figure 5.3. In Figure 8.3, the energy differences 4P minus 3P for the supplied fuel and electricity are given as well as the energies transferred by the heat exchangers. Negative energy differences indicate that the 4P system consumes less energy than the 3P system.

From Figure 8.3 we can see especially the large differences for the heat rejected by the cooling tower, which in turn is a result from the increased fuel supply summer and winter. The total energy consumption is about 40 % higher for 3P compared to 4P, and the electricity consumption is about 2,6 times higher. About 110 GWh fuel and totally (including electricity for cooling tower) 15 GWh electricity are consumed in 3P-Model 1.

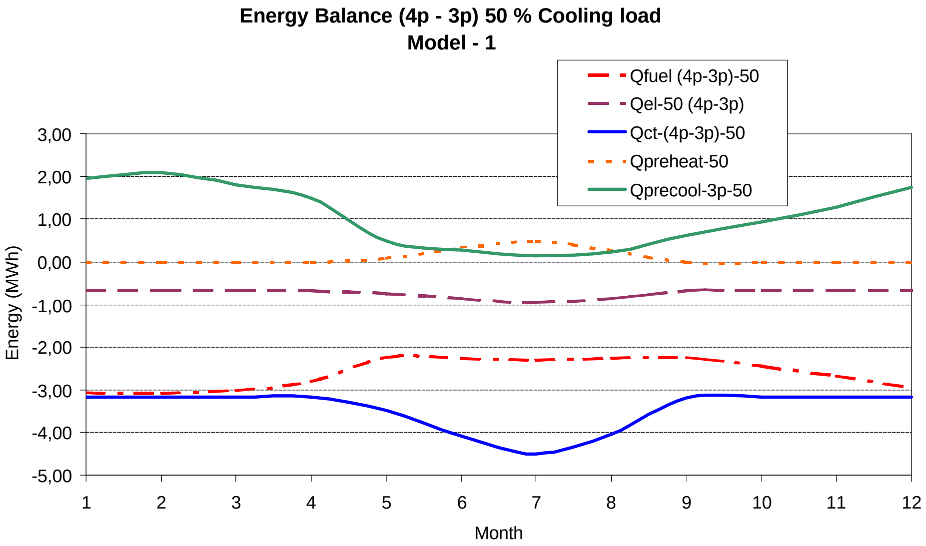


Figure 8.3: Energy differences 4P minus 3P for Model 1. Negative energy differences indicate that the 4P systems consume less energy than the 3P system.

8.1.2 Model 2

One way to make the combined systems more effective was to recover heat by means of a heat pump, i.e. to use the district cooling system as heat sink to the evaporator of a heat pump delivering its condenser heat to the district heating system. The difference between the two systems 4P and 3 P in Model 2 is the amount of heat that can be transferred from the cooling to the heating cycle due to different return temperatures and flow capacities in the both systems. Figures 8.4 – 8.6 are presenting results for the energy flows 4P, 3P and 4P-3P, respectively. Basically it was thought to improve only the 3P system by this heat pump, but on the other hand – and for the sake of correctness – we can apply the same improvement to the 4P system and it also positively affects the system efficiency of the 4P system.

The basic difference to Model 1 is that we - by means of the heat pump interconnection - replace biomass fuel by electrical heat pump work, which increases electrical consumption but saves energy in total, because cooling energy can be converted to heat. This is true for 4P as well as 3P systems. Hence the final decisions between these systems depend on the relation of electricity costs to biomass costs and the extra costs for the investment that depends on the size of the heat pump.

In Figures 8.4 and 8.5 energy flows for the respective systems are depicted.

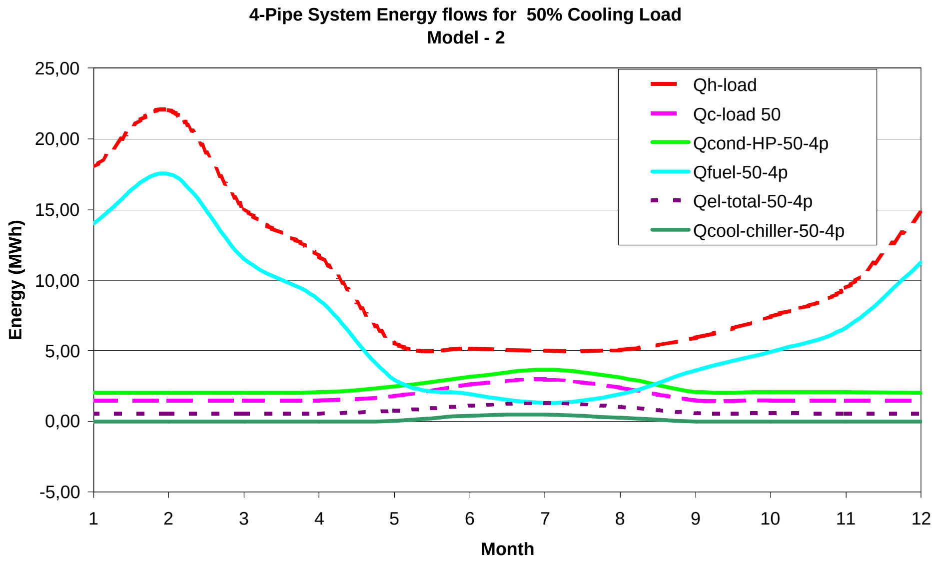


Figure 8.4: Energy flows for 4P-Model 2.

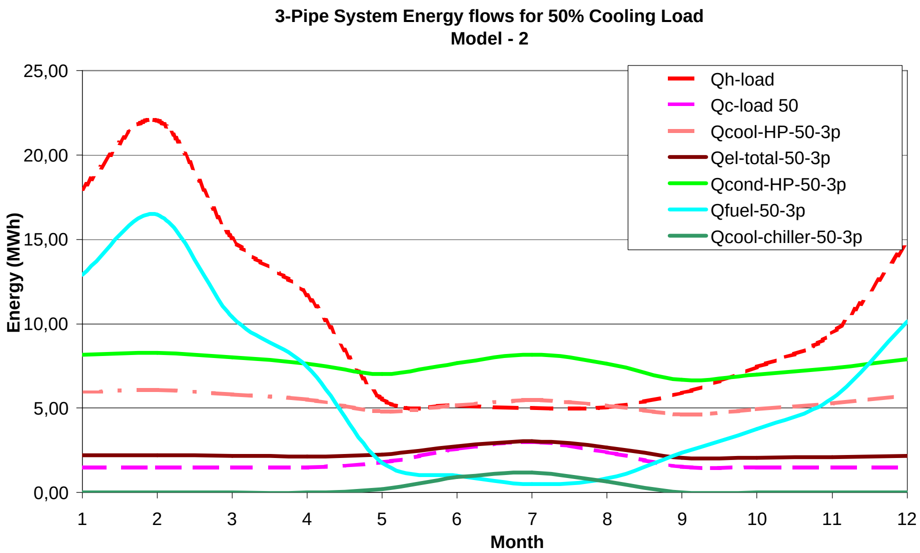


Figure 8.5: Energy flows for 3P-Model 2.

Compared to Model 1 the total annual electricity consumption is with 21 GWh 30 % larger in 3P-Model 2 compared to 3P-Model 1 and about three times larger for 3P than for 4P (Model 2). However, a large amount of fuel is saved due to the use of the heat pump. The HP-condenser can produce almost all heat in summer time and only a small amount of fuel is supplied during the summer. On the other hand, the heat pump is three times larger in the 3P system than in the 4P system, resulting in less fuel consumption but higher electricity consumption in 3P. The total energy consumption of 73 GWh is about

7 % higher in 3P than 4P. The interesting point is - compared with the reference case - that the energy consumption of the 4P system is only 83 % and that of the 3P system only 88 %. Hence Model 2 increases the energy efficiency compared to Model 1 for both types of systems.

From Figure 8.6 the positive value of Q_{fuel} indicates that the 3P system uses less biomass than the 4P system (at the expenses of increased electricity consumption). The negative energy difference of the energy rejected by the cooling tower indicates that still more energy – but much less than in Model 1 – is transferred in the 3P system from the heating circuit to the cooling circuit compared with the 4P system. The heat rejected in cooling towers in Model 2 is only about 5 % of the rejected heat in Model 1.

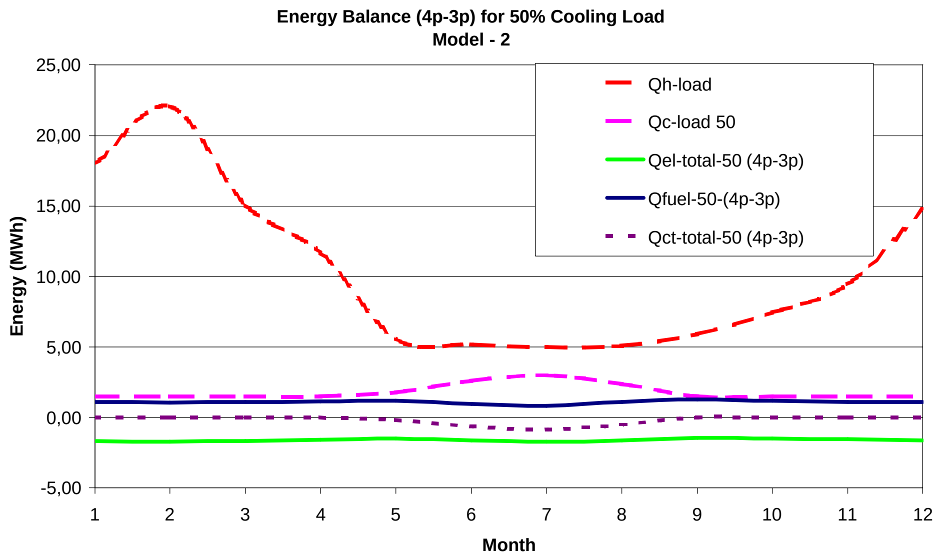


Figure 8.6: Energy differences 4P minus 3P-Model 2.

8.1.3 Model 3

In the Model 3 only a change for the 3P system was meaningful. The idea was to replace a part of the heat pump work by a cooling tower. Hence the precooling tower first chills the return flow before the heat pump achieves the final cooling. The main idea for this design is to save heat pump electricity at the cost of biomass fuel, the former being assumed to be less expensive than the latter. As explained in section 6.3, a subtle balance had to be worked out to optimise the flows between cooling tower and heat pump, i.e. minimising the energy consumption. For the 4P system, the optimisation resulted in the fact that the pre-cooling tower should be omitted due to the lower return temperatures of the cooling system; which means that Model 2 and Model 3 are the same for 4P systems.

In the 3P system, normally the pre-cooling of the return flow by the main cooling tower is not large enough to preheat the district heating flow completely and hence heat from the biomass plant also has to be added in summer time. On the other hand, the 4P system does not use the main cooling tower, the heat pump works at higher cooling power and the heating load can be matched during a longer period in summer time, on the expenses of a higher electricity consumption. In wintertime, as can be seen from Figure 8.7, most of the cooling is done by the main cooling-tower in the 3P system, saving compressor work, compared to the 4P system, where cooling is done by the heat pump. The large flows at the both cooling towers consume additional operation electricity. This is the reason why the total electricity consumption in the 3P-Model 3 system is 7,6 GWh/yr and therefore 20 % higher than in the 4P system and 30 % higher than in the reference case 4P-Model 1.

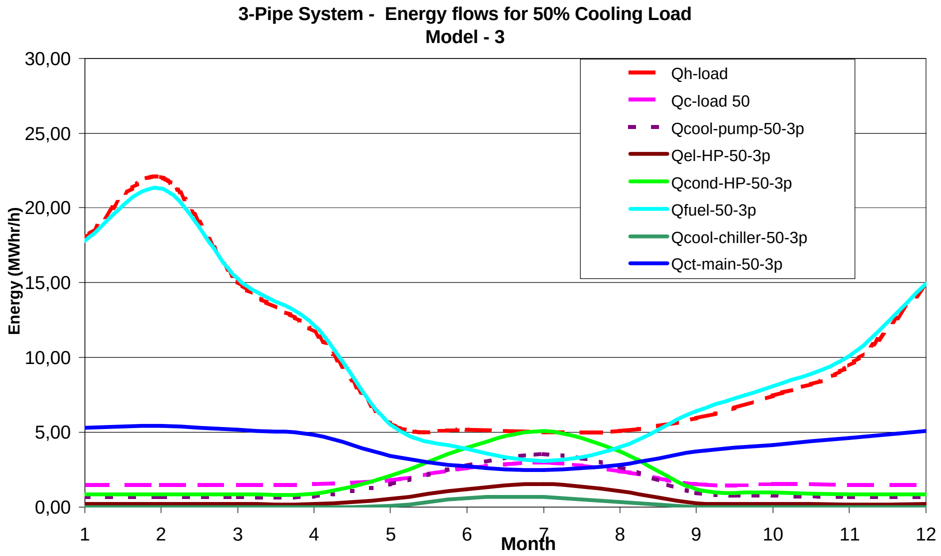


Figure 8.7: Energy flows for 3P-Model 3.

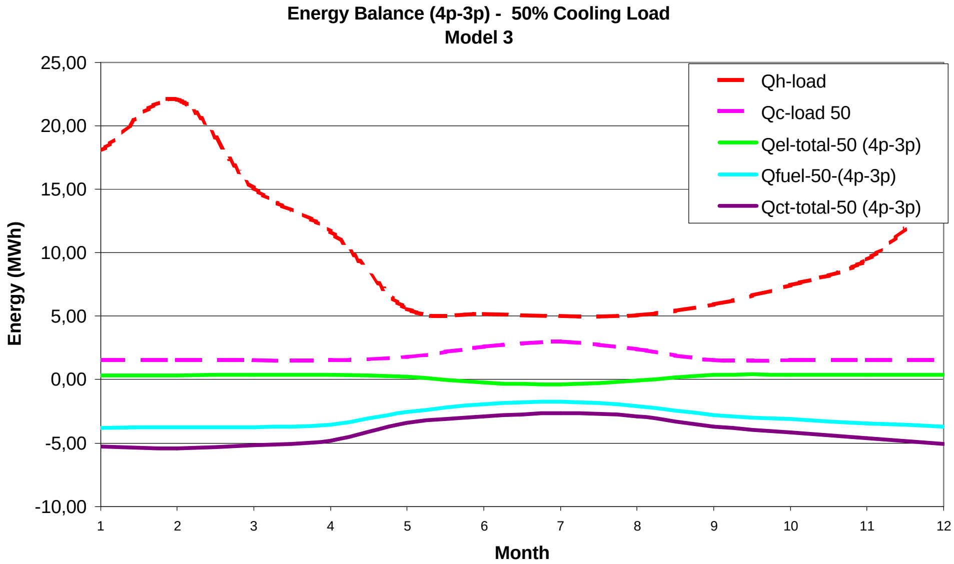


Figure 8.8: Heat and cooling load for Model 3 and energy differences of supplied electricity, supplied fuel and of rejected heat between 4P-Model 2 (=Model 3) and 3P-Model 3.

The total energy consumption of 3P-Model 3 is 13 % higher than for 3P-Model 2 respectively 12 % higher compared to reference case 4P-Model 1.

8.2 Cooling load 100 %

In the 100 % load case the top cooling load power is half of that of the heating power or 18 MW. That means that all the effects from the return flow mixing are stronger expressed. The mixed return temperatures are lower but also more heat is transferred in the 3P system from the heating to the cooling circuit compared to the 4P system.

The heating and cooling loads and differences in supplied fuel and electricity are seen from Figure 8.8 – 8.11 for the three models.

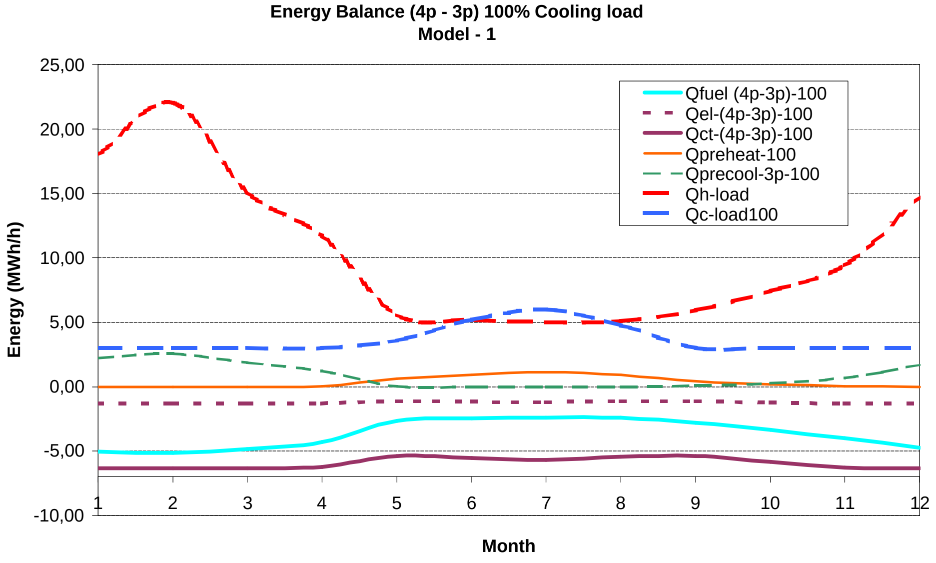


Figure 8.9: Heat and cooling load for Model 1 and 2and energy differences of supplied electricity, supplied fuel and of rejected heat

Figure 8.9 shows also the precooling and preheating energies, respectively, transferred by the heat exchangers. Both supplied fuel and supplied electricity are negative which means that the 4P system is more efficient. The Model 2 it is more energy efficient than Model 1 and the 3P system consumes less biomass fuel on the cost of larger electricity consumption. The total energy consumption is about 14 % larger in 3P than in 4P and compared to the reference case, the total energy consumption is ca 70 % (4P) and 80 % (3P). The 3P system of Model 3 uses about the same amount of electricity compared to 4P, but 90 % more heat is transferred from the heating circuit to the cooling circuit and has to be rejected in the two cooling systems. ***From an energy point of view, the 4P system of Model 2 is the most efficient system of all the systems investigated in this study.***

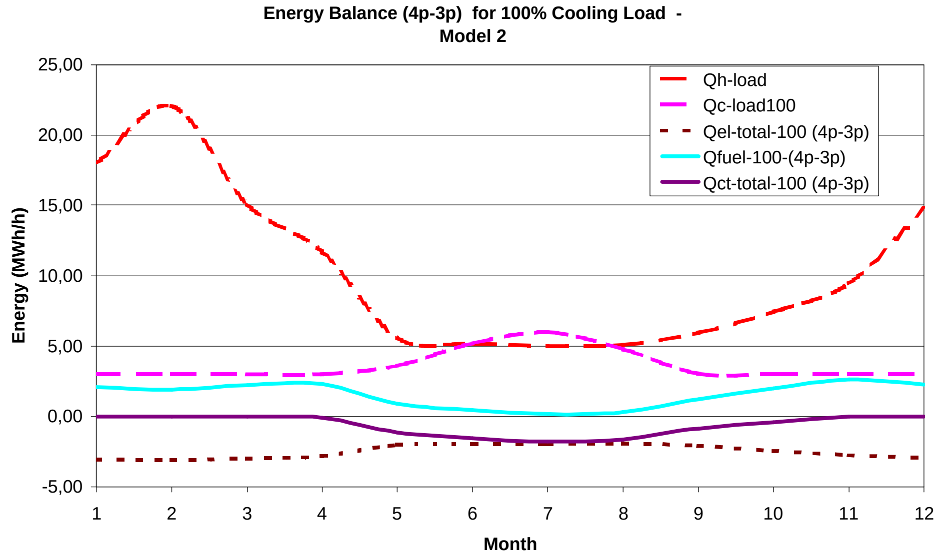


Figure 8.10: Heat and cooling load for Model 2 and energy differences of supplied electricity, supplied fuel and of rejected heat.

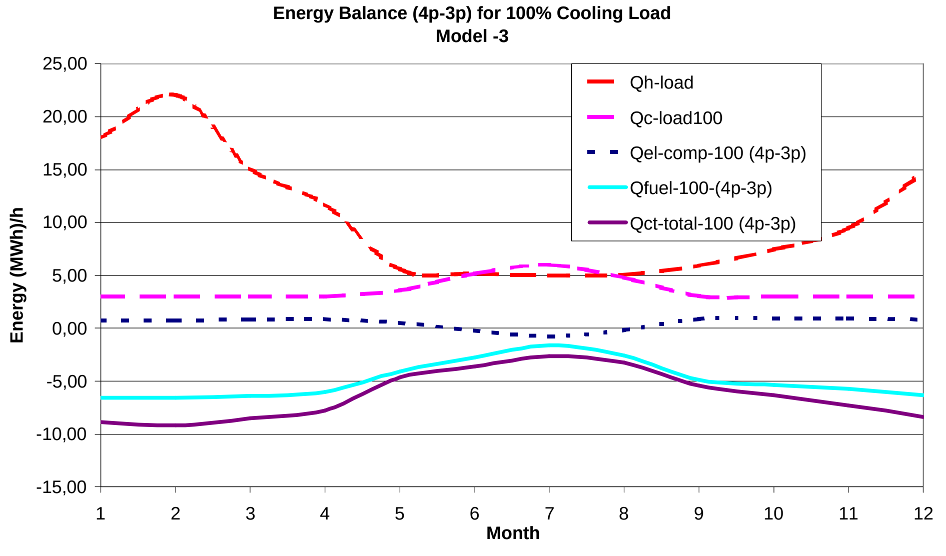


Figure 8.11: Heat and cooling load for Model 3 and energy differences of supplied electricity, supplied fuel and of rejected heat.

8.3 Cooling load 25 %

In the 25 % cooling load model, i.e. with only 4,5 MW top cooling load, the influence of the system type is getting of less importance.

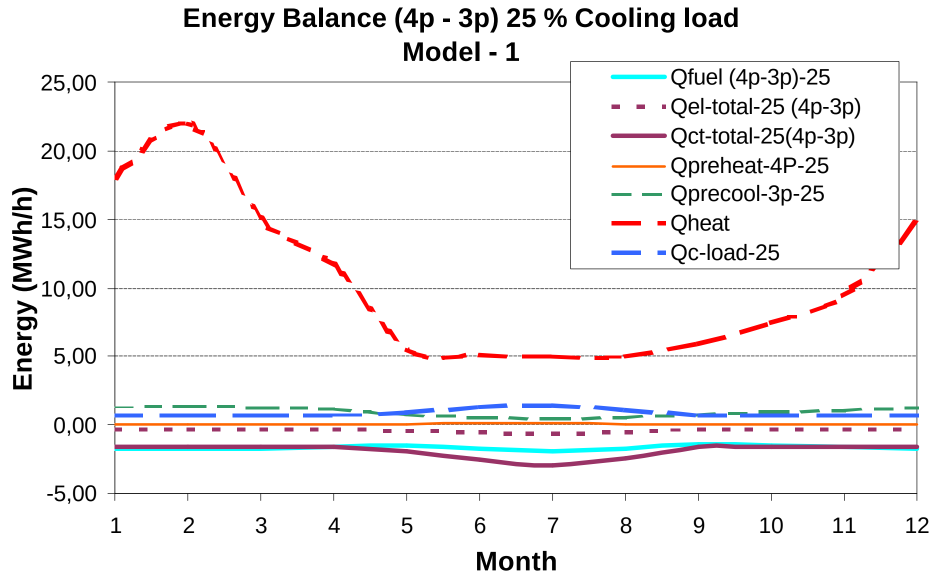


Figure 8.12: Heat and cooling load for Model 1 and energy differences of supplied electricity, supplied fuel and of rejected heat.

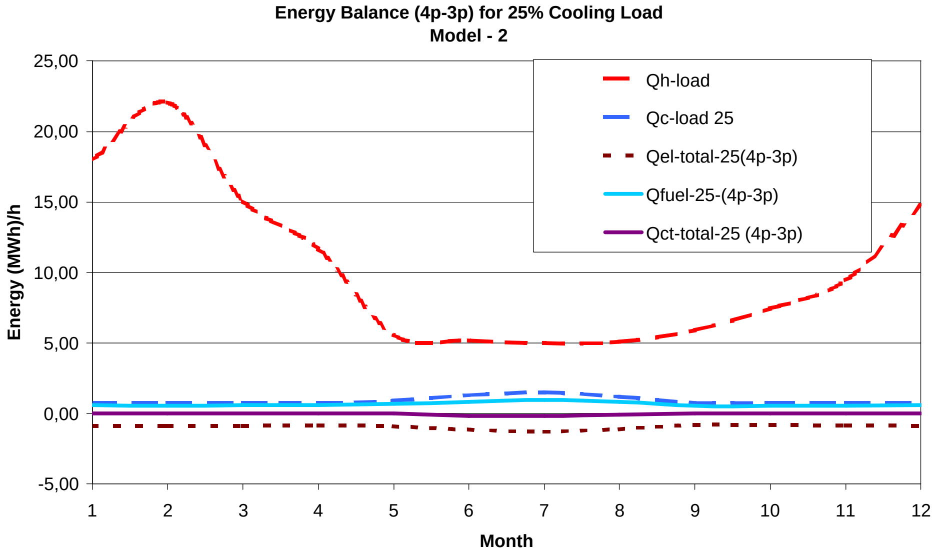


Figure 8.13: Heat and cooling load for Model 2 and energy differences of supplied electricity, supplied fuel and of rejected heat.

As discussed for the other loads before, 4P-Model 2 is the most energy efficient system, about 10 % less energy than the base case, with the 3P system using least biomass on the cost of relatively large electricity consumption. Totally, the 3P system uses only 4 % more energy than the 4P system.

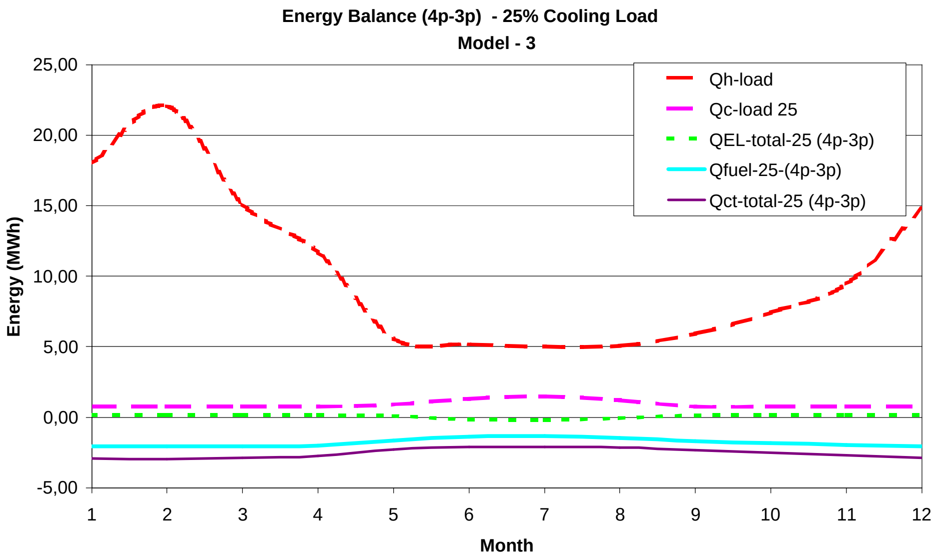


Figure 8.14: Heat and cooling load for Model 3 and energy differences of supplied electricity, supplied fuel and of rejected heat.

Hence the general conclusion can be that combined district heating and cooling systems interconnected by a heat pump according to Model 2 are energy efficient, so far the total energy consumption is concerned. (However, the final decision about optimal system is a question of how electricity is produced. If electricity is produced by condensation power plants, the electricity costs will be high and therefore the total costs will be high).

Table 8.1 summarises the resulting energy flows on an annual basis for all investigated systems. A further discussion about the economical differences between 3P and 4P systems must be based on an economical analysis.

Table-8.1 The summary of annual energy consumptions according to the three calculated models

100 %	**Model 1 3p-100**	**Model 2 3p-100**	**Model 3 3p-100**	**Model 1 4p-100**	**Model 2 4p-100**	**Model 3 4p-100**
Energy	**MWh/year**			**MWh/year**		
Heating Load	***90797***	***90797***	***90797***	***90797***	***90797***	***90797***
Supplied fuel (biomass)	109636	35244	91460	77460	48610	48610
Heat supply by Chiller	3161	0	0	0	0	0
Heat supply by HP	0	103426	27310	0	38424	38424
Cooling Load 100	***31839***	***31839***	***31839***	***31839***	***31839***	***31839***
Cooling by HP	0	70926	20025	0	26729	26729
Cooling by Chiller	72780	9480	5134	31839	5110	5111
Cooling by Pre-CT or HEX	7627	0	55248	0	0	0
Rejected heat	92219	12018	6508	40346	6460	6460
Compressor electricity	19470	35037	8660	8518	13044	13044
Total electricity	27711	35998	13600	11746	13561	13561
Fuel+electricity	137347	71243	105061	89206	62172	62172
50 %	**Model 1 3p-50**	**Model 2 3p-50**	**Model 3 3p-50**	**Model 1 4p-50**	**Model 2 4p-50**	**Model 3 4p-50**
Heating Load	***90797***	***90797***	***90797***	***90797***	***90797***	***90797***
Supplied fuel (biomass)	100253	52850	88839	77460	62397	62397
Heat supply by chiller	876	0	0	0	0	0
Heat supply by heat HP	0	66872	16261	0	21065	21065
Cooling Load 50	***15919***	***15919***	***15919***	***15919***	***15919***	***15919***
Cooling by HP	0	47090	11928	0	15006	15006
Cooling by Chiller	39830	2181	1187	15919	913	913
Cooling by Pre-CT or HEX	9442	0	36157	0	0	0
Rejected heat	50470	2765	1505	20173	1155	1155
Compressor electricity	10656	20365	4651	4259	6300	6300
Total electricity	15518	20586	7664	5873	6392	6392
Fuel+electricity	115771	73437	96504	83333	68790	68790
25%	**Model 1 3p-25**	**Model 2 3p-25**	**Model 3 3p-25**	**Model 1 4p-25**	**Model 2 4p-25**	**Model 3 4p-25**
Heating Load	***90797***	***90797***	***90797***	***90797***	***90797***	***90797***
Supplied fuel (biomass)	91947	64077	85617	77460	69854	69854
Heat supply by chiller	153	0	0	0	0	0
Heat supply by heat HP	0	39898	8901	0	10928	10928
Cooling Load 25	***7960***	***7960***	***7960***	***7960***	***7960***	***7960***
Cooling by HP	0	28697	6004	0	7943	7943
Cooling by Chiller	20923	253	106	7960	17	17
Cooling by Pre-CT or HEX	8027	0	22240	0	0	0
Rejected heat	26512	320	134	10087	22	22
Compressor electricity	5598	11201	2325	2130	2985	2985
Total electricity	7729	11295	4115	2936	2990	2990
Fuel+electricity	99676	75372	89723	80396	72844	72844

8.4 Dimensioning power of components

So far, the analysis where based on monthly mean energies and finally on annual energy sums. From these figures, the operational costs for the different system Models can be derived as well as the general energy efficiency and hence the technical meaning of the different systems can be evaluated. However, in order to evaluate the systems completely, also the investment costs for the different solutions must be compared. This cost must be based on dimensioning capacities, i.e. maximum powers in MW for each component. These dimensioning capacities are summarised in Table 8.2.

Table 8.2: The peak power capacities of the individual components of the different systems:

a) Cooling load 100 %;

COMPONENT	Unit	Model 1 3p-100	Model 2 3p-100	Model 3 3p-100	Model 1 4p-100	Model 2 4p-100	Model 3 4p-100
Heating Plant	MW	31,4	22,3	30,2	26,3	24,0	24,0
HP Evaporator	MW	0,0	10,9	6,9	0,0	3,3	3,3
CH Evaporator	MW	22,1	15,2	15,2	16,7	13,4	13,4
Pre-Cooling CT	MW	0,0	0,0	9,5	0,0	0,0	0,0
HP Compressor	MW	0,0	4,3	3,5	0,0	1,7	1,7
CP Compressor	MW	5,9	4,1	4,1	4,5	3,5	3,5
HP Condenser	MW	0,0	15,1	10,4	0,0	5,0	5,0
CH Condenser	MW	28,0	19,3	19,3	21,2	16,9	16,9
H-Exchanger	MW	2,0	0,0	0,0	0,0	0,0	0,0
C-Exchanger	MW	2,9	0,0	0,0	0,0	0,0	0,0

b) Cooling load 50%

COMPONENT	Unit	Model 1 3p-50	Model 2 3p-50	Model 3 3p-50	Model 1 4p-50	Model 2 4p-50	Model 3 4p-50
Heating Plant	MW	29,2	24,1	28,7	26,3	25,1	25,1
HP Evaporator	MW	0	6,6	6,6	0	3,3	3,3
CH Evaporator	MW	13,30	6,7	6,7	8,4	5,0	5,0
Pre-Cooling CT	MW	0	0	5,5	0	0	0
HP Compressor	MW	0	3,4	3,4	0	1,7	1,7
CP Compressor	MW	3,6	1,8	1,8	2,2	1,3	1,3
HP Condenser	MW	0,0	10,0	10,0	0	5	5
CH Condenser	MW	16,9	8,5	8,5	10,6	6,4	6,4
H-Exchanger	MW	1,6	0	0	0	0	0
C-Exchanger	MW	2,2	0	0	0	0	0

c) Cooling load 25 %

COMPONENT	Unit	Model 1 3p-25	Model 2 3p-25	Model 3 3p-25	Model 1 4p-25	Model 2 4p-25	Model 3 4p-25
Heating Plant	MW	27,9	25,2	27,6	26,3	25,7	25,7
HP Evaporator	MW	0,0	6,2	6,2	0,0	3,3	3,3
CH Evaporator	MW	8,5	2,3	2,3	4,2	0,9	0,9
Pre-Cooling CT	MW	0,0	0,0	3,0	0,0	0,0	0,0
HP Compressor	MW	0,0	3,1	3,1	0,0	1,7	1,7
CP Compressor	MW	2,3	0,6	0,6	1,1	0,3	0,3
HP Condenser	MW	0,0	9,3	9,3	0,0	5,0	5,0
CH Condenser	MW	10,7	2,9	2,9	5,3	1,1	1,1
H-Exchanger	MW	1,0	0	0	0	0	0
C-Exchanger	MW	1,4	0	0	0	0	0

9 Economy

9.1 Cost models

From the energy analysis it gets clear that three-pipe systems can save energy compared to the reference-case of two separate circulation systems for heating and cooling, but that similar efficiency increasing system components can also be installed in 4P systems, resulting in still better system efficiency. Because of the fact that always some net heat in the 3P system is transferred from the heating to the cooling circuit that must be rejected or recovered, the design capacities of cooling towers and heat pumps will be larger than in the 4P system and therefore will cause higher investment costs. Hence an economic analysis has to be carried out in order to see the value of the 3P systems.

The basic model is simple and straightforward: The investment costs C_{costs} (in SEK) of the main components in the heating and cooling circuits are summed up based on specific investment costs I_i defined below. Additionally the costs for the installed pipe system C_{pipes} (new pipes) are added. The heating and stack gas condensation systems are not included because it is assumed that he heating plant will be built at given design power independently of eventual savings due to the interconnected heat pump systems. (It should be noted that the maximum and minimum heating power capacities according to Table 8.2 are 31,2 and 22,4 MW respectively, a difference which also can be included in the economical analysis. This is done in form of a sensitivity analysis, chpt. 9.3.1). The specific investment costs I_i are multiplied with the design capacity in order to result in the investment costs C:

$$C_{investment} = \sum_i C_{comp,i} + \sum_j C_{pipes} \quad \text{SEK.} \qquad 9.1$$

The investment costs are thereafter annualised for an amortisation time of 20 years and an interest rate of 5 %. This results in an annuity of 8 %. To these annual investment costs the operating costs, i.e. the annual costs of biomass fuel and electricity and the value of pipe heat losses are added. By that way the costs of different systems are compared ([8], [11], [14], [15]).

The costs are based on manufacturer prices and literature references for cooling systems. All prices are taken for component sizes in the range of Megawatts. Although maximum cooling capacities can be as large as 18 MW, we decided to keep prices at maximum capacity ranges of 5 MW because it is assumed that large engines will be divided in subunits for the reason of efficient partial load operation.

Costs for the pipe system are published by the Swedish District Heating Association [16] and heat loss estimates for a given mixture of pipe systems are taken according to Lögstör design tables [19]. Pipe system costs are including installation. Pipe cost summaries are presented in Table 9.1.

9.1.1 Cooling systems

Compressor chillers

The costs of compression chillers based on evaporator power can be expresses as:

$I_{chiller}$ =2200 $(P/300)^{-0,3}$ SEK/kW (P in kW) 9.2

This relation corresponds to costs at 5000 kW of ca 950 SEK/kW (see Figure 9.1). The delivery temperature T_{co} of the condenser outlet of these chillers is 40 C.

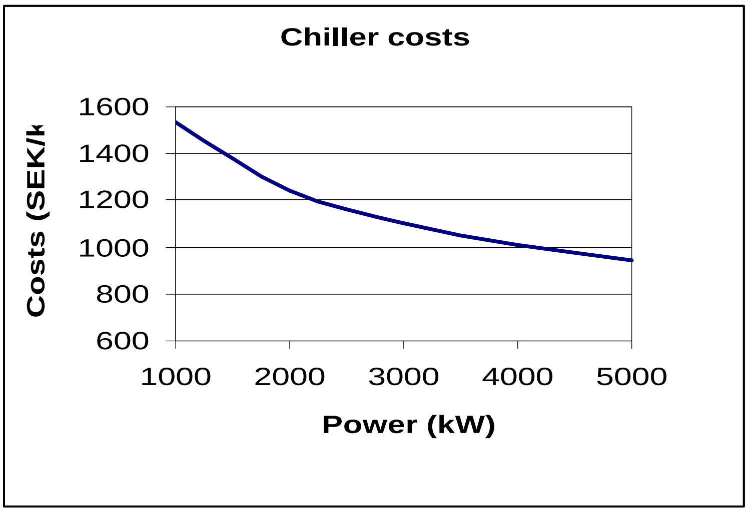


Figure 9.1: Costs of compressor chiller systems (1999).

Single stage heat pump

For higher condenser outlet temperatures up to T_{co} = 60 C, single stage heat pumps are used.

$$I^1_{HP} = I_{chiller} * (\frac{T_{co}}{40})^{0,66} \quad \text{SEK/kW} \qquad 9.3$$

Double stage heat pump

Such heat pumps are used to provide condenser outlet temperatures above 60 C. These heat pumps are assumed to be able to deliver condenser outlet temperatures up to 80 C.

$$I^2_{HP} = I_{chiller} * 1{,}6 * (\frac{T_{co}}{40})^{0,66} \quad \text{(SEK/kW)} \qquad 9.4$$

Cooling Tower

The investment cost for a wet cooling tower system can be found to I_{CT} = 250 SEK/kW. 9.5

Heat Exchanger
The investment cost of plate exchangers with low logarithmic mean temperatures (= 3 C) are estimated to be I_{HEX} = 80 SEK/kW for systems larger than 1 MW.

9.1.2 Pipe systems

The costs of pipes can be calculated according to the Swedish statistics of the Swedish District Heating Association [16] for district heating pipe systems. The costs for district cooling pipes are based on information supplied by manufacturers (material) in combination with the costs for ground work and installation costs for DH pipes from [16]. Because the aim of this paper is to compare different system models on a principal basis it is sufficient to base the cost calculations on a simple, but comparable cost structures.

In order to get the total costs we have to make an assumption about the type of built environment and heating density of the area. For this reason, a model city must be defined. The connected heating capacity is 36 MW (DUT = -20 C) and the connected cooling capacity (DUT = 30 C) is 18 MW, 9 MW and 4,5 MW respectively, depending on the model type. The maximum pipe dimension for heating will be in this case (T_{hs}=100 C, T_{hr} = 50 C) DN 400, whereas the maximum dimension for cooling (T_{cs}= 6 C, T_{cr}=15 C) will be DN 600 for the 100 % system. However, we assume that the heating and cooling net is divided into ***two main stems***, resulting in a maximum pipe dimension of DN 280 for heating. For cooling, the dimensions will then be DN 450 (100 %), DN 320 (50 %) and DN 270 (25 %), respectively. The length of the heating system is taken to be 25 km (which is a typical size for the heating capacity of 36 MW according the Swedish DH statistics), which per definition is also the length of the cooling system for all systems.

In order to simplify the calculation we define ***an average pipe dimension*** for all the systems. By assuming that the line density is equally distributed all over the net, and that the net has a hierarchical tree structure where the sums of the length of the smaller dimensions increase in inverse proportion to the distributed power, we can assume that the average pipe dimension is half the maximum pipe dimension, for both the heating and the cooling system.

Costs for service pipes should also be included. Heating in general is distributed in such an area to a large amount of customers, typically 200 or more, with relatively small service pipe dimensions, DN 40 in the average. The average service pipe length is assumed to be 50 m. Hence, we assume 10 km service pipes DN 40 for the heating connection. For the cooling, although also evenly distributed, we assume that it is distributed among 50 users, resulting in typical service pipe connections for the three different cooling load models of DN 110 (100 %), DN 80 (50 %) and DN 65 (25 %) respectively. The total length of the cooling service pipes is adequately assumed to be 2,5 km. A control calculation for the differences of the service pipes for 3P and 4P resulted in -2 MSEK for 100 % and less than 1 MSEK in the load cases 50 % and 25 % respectively, therefore the service pipe costs have been omitted in further cost analyses.

For the 3P system, the return pipe dimension must be dimensioned for the maximum possible return flow. This will be the flow of delivering the cooling power at the summer design capacities. This depends on the system model type (100, 50 or 25 %).

Costs for the pipes are taken as overall costs for cities outer region from [16]. A 30 % -price reduction for ground work is used for the main distribution network for the reason of combined installation work for both heating and cooling systems. For the 3P system, costs of the DH heating supply and return pipes are calculated on the bases of the dimension of each, also with a 30 % cost reduction for the combined ground work.

Also heat losses for the heating and return pipes can be quantified. The differences of heat losses 4P-3P are in all system models about 1600 –1800 MWh/yr (higher heat losses for the 4P system because of higher return temperatures), compensating part of the higher operating costs of the 3P system. Heat losses from service pipes are neglected.

In Table 9.1, the pipe costs for heating and cooling systems are summarised, 4P systems as well as 3P systems.

Table 9.1: Piping system cost comparison. Cost of heat losses based on 80 SEK/MWh biomass fuel costs.

Type	**Dimension**	**Length**	**Costs**	**Total Costs**	**Heat loss**	**Tot. heat loss**	**Value of heat loss**
	DN	(m)	SEK/m	MSEK	W/m	MWh/yr	SEK/yr
Heating pipe	150	25000	3800		0,309		
Heating Service pipe	25	10000	1400		0		
Cooling pipe 100%	450	25000	5000		0		
Cool. service pipe 100%	110	2500	2500		0		
Cooling pipe 50%	350	25000	4600		0		
Cool. service pipe 50%	80	2500	2100		0		
Cooling pipe 25%	265	25000	4200		0		
Cool. service pipe 25%	65	2500	1800		0		
Pipe-system costs							
Four-pipe 100%		25000	5470	136,75		8797,23	703778,4
Four-pipe 50%		25000	4580	114,5		8797,23	703778,4
Four-pipe 25%		25000	3990	99,75		8797,23	703778,4
Three-pipe 100%	150/450/500	25000	4560	114,0		7156,92	572553,6
Three-pipe 50%	150/310/350	25000	3670	91,75		7130,64	570451,2
Three-pipe 25%	150/210/265	25000	3120	78,0		6964,20	557136,0

9.1.3 Operation costs

The following operation costs are applied.
Biomass: O_f = 80 (SEK/MWh)
Electricity: O_e = 350 (SEK/MWh).

9.1.4 Total annual costs

The annualised investment costs are the sum of all investment costs multiplied by the annuity rate a = 8 %.

The total annual system costs: Annual investment costs + annual operating costs.

9.2 Example of calculation

Investment costs:
The investment costs of equipments are based on their maximum power capacities. For the following example of the three-pipe system of Model 3 for 100 % cooling load, the evaporation power of the heat pump reaches a peak power of 6,85 MW at the hottest day over a year. Its maximum operating temperature is 80 C and hence cost function equ. 9.4 applies:

I_{HP} (5 MW): 946 SEK/kW
C_{HP}: 2946 SEK/kW * 6850 kW * 1,6* $2^{0,66}$ = 16,5 MSEK.

Similarly the investments costs for the other components are:
$I_{Chiller}$ (5 MW): 946 SEK/kW
$C_{chiller}$: 946 SEK/kW * 15200 kW = 14,4 MSEK
$I_{cooling\ tower}$: 250 SEK/kW
$C_{cooling\ tower}$: 250 SEK/kW *(9,5 + 19,3 MW) *1000= 7,2 MSEK.

Hence the total investment cost (cooling equipment) for 3P is: 16,4 + 14,4 + 7,2 MSEK.
= 38,0 MSEK.

The investment cost in the pipe system is according to Table 9.1: 114 MSEK.

Operating costs:
According to Table 8.1, the annual supplied energies of the 3P Model 3 system are 91460 MWh fuel and 13600 MWh electricity. Hence the annual operating costs for this system are 91460*80 +13600*350 =12 070 800 SEK. Heat loss costs are 572 553 SEK.

Annual costs to be compared:
With an annuity of 8 %, the annual system costs can be calculated (heating plant not included):

(114 000 000 + 38 000 000) * 0,08 + 12 070 800 + 572 553 = 24,8 MSEK/yr.

A similar calculation for the 4P system Model 3 would result in:

(136 750 000 + 25 840 000) * 0,08 + 3 088 880 + 4 746 350 +703 760 = 22,3 MSEK.

Hence the 4P system is more economic than the 3P system.

9.3 Cost comparison

The summarising cost comparison for the three system models and for the three load cases, 100 %, 50 % and 25 % respectively, is shown in Table 9.2. The breakdown of costs for the different systems is shown in more detail in Tables 9.3, a - c.

Table 9.2: Comparison of annual costs – all systems. Costs in MSEK. Costs of biomass fuel 80 SEK/ MWh electricity 350 SEK/ MWh.

	Model 1	Model 2	Model 3
4P-100	23,64	22,35	22,35
3P-100	30,43	28,73	24,82
4P-50	18,96	18,31	18,31
3P-50	22,73	21,28	19,75
4P-25	16,35	16,17	16,17
3P-25	17,73	17,34	16,61

If we take the cooling load 50 % (i.e. maximum cooling load 9 MW, or a quarter of the maximum heating load) as the base case, we can see that the sum of the annual costs, including pipes and the cooling system as well as the operating costs, is lower for the 4P system than for the 3P system. But it also gets evident that the base case 4P-model 1 with completely separated heating and cooling systems is not the economically best system, but the system 4P- Model 2 with a heat pump recovering part of the cooling energy to the district heating system. The reason is the fuel saving of the Model 2 system compared to Model 1. In spite of the higher investment costs for the heat pump in Model 2 and the low fuel costs (80 SEK/MWh), the total annual costs are about 0,65 MSEK lower per year. However, the ranking is depending on fuel costs and an increase of the biomass fuel costs to 90 SEK/MWh would equalise the costs of both systems.

Among the three different 3P systems, Model 3 is the most economic one, the difference to Model 4P-Model 3 (= 4P-Model 2) being about 1, 4 MSEK per year, i. e. about 8 %. The difference to the 4P-Base Model 1 case is 4 %, but in both cases the 4P system is more economic than 3P. By comparing the 4P and 3P systems of Model 3 we can state that the three 3P systems have higher investment costs (26 MSEK (3P)) instead of 16 MSEK (4P)), but lower pipe system costs (91 MSEK (3P) instead of 115 MSEK (4P)). But the energy consumption in the 3P system is much higher than in the 4P system and therefore the total annual energy costs of the 3P system (9,8 MSEK/yr (3P), 7,6 MSEK/yr (4P)) do not balance its lower annual investment costs.

The system 3P-Model 2 has no pre-cooling tower, instead it uses the heat pump during a longer period for cooling and needs therefore more electricity than Model 3. Because of the chosen electricity price of 350 SEK/MWh this makes the annual costs of Model 2 higher than those of Model 3.

The relations between 3P and 4P systems are also valid for the other cooling loads, 100 % and 25 %, respectively. The main difference is that the gap between 4P and 3P is getting larger the higher the cooling load is, which is expected because the absolute systems costs are getting larger for higher cooling loads.

Hence from the cost comparison based on the given cost conditions it can be concluded that the 4P system is slightly more economic than the 3P system. In any case the 3P system uses much more energy than the 4P system and for that reason one should be reluctant when using it. However, there is one exception, namely 3P Model 2, for which the total energy used is only slightly higher than in the 4P case, but the electricity consumption of it is high. ***Under such conditions, where electricity has a low price, 3P Model 2 could be an alternative possibility for combined heating and cooling systems.*** This is shown in more detail in the sensitivity study of the next section.

Table 9.3: Cost comparison of 4P and 3P systems
a) Cooling load 100 %.

COMPONENT	Unit	Model 1	Model 2	Model 3	Model 1	Model 2	Model 3
		4p-100	**4p-100**	**4p-100**	**4p-100**	**4p-100**	**4p-100**
		Cost basis			Costs/SEK		
Investment							
H-Exchanger	MW	0	0	0	0,00	0,00	0,00
C-Exchanger	MW	0	0	0	0,00	0,00	0,00
Pre-Cooling CT	MW	0	0	0	0,00	0,00	0,00
Heat Pump	MW	0	3,3	3,3	0,00	8,94	8,94
Chiller	MW	16,7	13,4	13,4	15,80	12,68	12,68
Chiller Cooling tower	MW	21,2	16,9	16,9	5,30	4,23	4,23
Pipes					136,75	136,75	136,75
Sum investment	MSEK				*157,85*	*162,59*	*162,59*
Annual investment value (annuity = 8%)	MSEK/yr				***12,63***	***13,01***	***13,01***
Operating costs							
Fuel	MWh	77460	48610	48610	6,20	3,89	3,89
Electricity	MWh	11746	13561	13561	4,11	4,75	4,75
Heat losses	MWh	8797	8797	8797	0,70	0,70	0,70
Sum Operating costs	MSEK/yr				***11,01***	***9,34***	***9,34***
TOTAL ANNUAL COSTS	*MSEK/yr*				***23,64***	***22,35***	***22,35***

COMPONENT	Unit	Model 1	Model 2	Model 3	Model 1	Model 2	Model 3
		3p-100	**3p-100**	**3p-100**	**3p-100**	**3p-100**	**3p-100**
		Dimensioning basis			Costs		
Investment							
H-Exchanger	MW	2	0	0	0,16	0,00	0,00
C-Exchanger	MW	2,9	0	0	0,23	0,00	0,00
Pre-Cooling CT	MW	0	0	9,5	0,00	0,00	2,38
Heat Pump	MW	0	10,9	6,9	0,00	26,07	16,50
Chiller	MW	22,1	15,2	15,2	20,91	14,38	14,38
Chiller Cooling tower	MW	28	19,3	19,3	7,00	4,83	4,83
Pipes					114,00	114,00	114,00
Sum investment	MSEK				*142,30*	*159,27*	*152,08*
Annual investment value (annuity = 8%)	MSEK/yr				***11,38***	***12,74***	***12,17***
Operating costs							
Fuel	MWh	109636	35244	91460	8,77	2,82	7,32
Electricity	MWh	27711	35998	13600	9,70	12,60	4,76
Heat losses	MWh	7157	7157	7157	0,57	0,57	0,57
Sum Operating costs					***19,04***	***15,99***	***12,65***
TOTAL ANNUAL COSTS	*MSEK/yr*				***30,43***	***28,73***	***24,82***

Table 9.3: Cost comparison of 4P and 3P systems
b) Cooling load 50 %.

COMPONENT	**Unit**	**Model 1**	**Model 2**	**Model 3**	**Model 1**	**Model 2**	**Model 3**
		4p-50	**4p-50**	**4p-50**	**4p-50**	**4p-50**	**4p-50**
		Cost basis			Costs/SEK		
Investment							
H-Exchanger	MW	0	0	0	0,00	0,00	0,00
C-Exchanger	MW	0	0	0	0,00	0,00	0,00
Pre-Cooling CT	MW	0	0	0	0,00	0,00	0,00
Heat Pump	MW	0	3,3	3,3	0,00	8,94	8,94
Chiller	MW	8,4	5	5	7,95	4,73	4,73
Chiller Cooling tower	MW	10,6	6,4	6,4	2,65	1,60	1,60
Pipes					114,5	114,5	114,5
Sum investment	MSEK				*125,10*	*129,77*	*129,77*
Annual investment value (annuity = 8%)	MSEK/yr				***10,01***	***10,38***	***10,38***
Operating costs							
Fuel	MWh	77460	62397	62397	6,20	4,99	4,99
Electricity	MWh	5873	6392	6392	2,06	2,24	2,24
Heat losses	MWh	8797	8797	8797	0,70	0,70	0,70
Sum Operating costs					***8,96***	***7,93***	***7,93***
TOTAL ANNUAL COSTS	*MSEK/yr*				***18,96***	***18,31***	***18,31***

COMPONENT	**Unit**	**Model 1**	**Model 2**	**Model 3**	**Model 1**	**Model 2**	**Model 3**
		3p-50	**3p-50**	**3p-50**	**3p-50**	**3p-50**	**3p-50**
		Dimensioning basis			Costs		
Investment							
H-Exchanger	MW	1,6	0	0	0,13	0,00	0,00
C-Exchanger	MW	2,2	0	0	0,18	0,00	0,00
Pre-Cooling CT	MW	0	0	5,5	0,00	0,00	1,38
Heat Pump	MW	0	6,6	6,6	0,00	15,78	15,78
Chiller	MW	13,3	6,7	6,7	12,58	6,34	6,34
Chiller Cooling tower	MW	16,9	8,5	8,5	4,23	2,13	2,13
Pipes					91,75	91,75	91,75
Sum investment	MSEK				*108,86*	*116,00*	*117,37*
Annual investment value (annuity = 8%)	MSEK/yr				***8,71***	***9,28***	***9,39***
Operating costs							
Fuel	MWh	100253	52850	88839	8,02	4,23	7,11
Electricity	MWh	15518	20586	7664	5,43	7,21	2,68
Heat losses	MWh	7131	7131	7131	0,57	0,57	0,57
Sum Operating costs	MSEK/yr				***14,02***	***12,00***	***10,36***
TOTAL ANNUAL COSTS	MSEK/yr				**22,73**	**21,28**	**19,75**

Table 9.3: Cost comparison of 4P and 3P systems
c) Cooling load 25 %.

COMPONENT	Unit	Model 1	Model 2	Model 3	Model 1	Model 2	Model 3
		4p-25	**4p-25**	**4p-25**	**4p-25**	**4p-25**	**4p-25**
		Cost basis			Costs/SEK		
Investment							
H-Exchanger	MW	0	0	0	0,00	0,00	0,00
C-Exchanger	MW	0	0	0	0,00	0,00	0,00
Pre-Cooling CT	MW	0	0	0	0,00	0,00	0,00
Heat Pump	MW	0	3,3	3,3	0,00	8,94	8,94
Chiller	MW	4,2	0,9	0,9	4,19	1,42	1,42
Chiller Cooling tower	MW	5,3	1,1	1,1	1,33	0,28	0,28
Pipes					99,75	99,75	99,75
Sum investment	MSEK				***105,26***	***110,39***	***110,39***
Annual investment value (annuity = 8%)	MSEK/yr				***8,42***	***8,83***	***8,83***
Operating costs							
Fuel	MWh	77460	69854	69854	6,20	5,59	5,59
Electricity	MWh	2936	2990	2990	1,03	1,05	1,05
Heat losses	MWh	8797	8797	8797	0,70	0,70	0,70
Sum Operating costs					***7,93***	***7,34***	***7,34***
TOTAL ANNUAL COSTS	MSEK/yr				***16,35***	***16,17***	***16,17***

COMPONENT	Unit	Model 1	Model 2	Model 3	Model 1	Model 2	Model 3
		3p-25	**3p-25**	**3p-25**	**3p-25**	**3p-25**	**3p-25**
		Dimensioning basis			Costs		
Investment							
H-Exchanger	MW	1	0	0	0,08	0,00	0,00
C-Exchanger	MW	1,4	0	0	0,11	0,00	0,00
Pre-Cooling CT	MW	0	0	3	0,00	0,00	0,75
Heat Pump	MW	0	6,2	6,2	0,00	14,83	14,83
Chiller	MW	8,5	2,3	2,3	8,04	2,75	2,75
Chiller Cooling tower	MW	10,7	2,9	2,9	2,68	0,73	0,73
Pipes					78	78	78
Sum investment	MSEK				*88,91*	*96,30*	*97,05*
Annual investment value (annuity = 8%)	MSEK/yr				***7,11***	***7,70***	***7,76***
Operating costs							
Fuel	MWh	91947	64077	85617	7,36	5,13	6,85
Electricity	MWh	7729	11295	4115	2,71	3,95	1,44
Heat losses	MWh	6964	6964	6964	0,56	0,56	0,56
Sum Operating costs	MSEK/yr				***10,62***	***9,64***	***8,85***
TOTAL ANNUAL COSTS	MSEK/yr				***17,73***	***17,34***	***16,61***

9.3.1 Sensitivity study

Table 9.4: Comparison of annual costs – all systems. Costs in MSEK.
Costs of biomass fuel 200 SEK/ MWh, electricity 200 SEK/ MWh.

	Model-1	Model-2	Model-3
4P-100	27,33	23,65	23,65
3P-100	33,06	24,50	29,00
4P-50	23,83	22,02	22,02
3P-50	27,14	21,36	24,93
4P-25	21,80	21,08	21,08
3P-25	23,11	20,05	22,27

In the example of Table 9.4 above the price relation between fuel and electricity was drastically changed, making costs of electricity equal to that of fuel at 200 SEK/MWh. As expected, in this case the 3P system of Model 2 (50 and 25 %) is the most economic system of all the systems. Of course this situation might be exceptional, but in some countries, such as Sweden, Norway and Canada, electricity might be available at low costs especially during the season when the cooling demand is high.

Table 9.5: Comparison of annual costs – all systems. Costs in MSEK.
Costs of biomass fuel 150 SEK/ MWh, electricity 300 SEK/ MWh.

	Model 1	Model 2	Model 3
4P-100	29,09	25,69	25,69
3P-100	37,22	29,90	31,04
4P-50	24,71	22,98	22,98
3P-50	29,47	24,45	26,08
4P-25	22,24	21,53	21,53
3P-25	24,27	21,75	22,89

In the example of table 9.5 a more realistic price relation was chosen where electricity price is twice the biomass fuel price per MWh. In this case the 4P systems are dominating the 3P systems, except in the case of smaller cooling power (25 %), where the difference for 3P-Model 2 becomes very small and the costs are more or less equal.

From these examples the general conclusion can be drawn that there might exist conditions on the market on which a 3P system can at least be economically equivalent to 4P systems. However, these conditions must be analysed carefully and with more detailed cost approximations than those used in this study.

9.3.2 Influence of heating plant costs

In all the cost analysis so far, the costs of the heating plant has been omitted, assuming that this will be the same type (biomass boiler including stack gas condenser) and the same size (36 MW) for all systems. However, this might not be the case. As can be seen from table 8.2, the necessary biomass plant has different maximum heating powers in each system model. One can argue that the system will be built anyhow at design capacity (36 MW at T_{design} -20 C) and therefore it would be the same for all types of models with a given cooling load. On the other hand, for temperatures used in our model, the really design capacities of the heating plant vary. In the case of new construction of the whole system the heating plant can be designed with lower power because the heat pumps supply additional heating, see Table 9.6. That means that the annual investment costs can be reduced by different amounts of money presented in Table 9.6.

Table 9.6: Maximum design loads and cost saving for the systems in the different models. Annual cost saving for reduced heating plant power compared to design case 32 MW with Stockholm climate. The values in the last three columns can be subtracted from the cost values in tables 9.2, 9.4 and 9.5.

	Model 1	Model 2	Model 3	Model 1	Model 2	Model 3
System	MW	MW	MW	MSEK	MSEK	MSEK
4P-100	26,3	24	24	0,46	0,64	0,64
3P-100	31,4	22,3	30,2	0,05	0,78	0,14
4P-50	26,3	25,1	25,1	0,46	0,55	0,55
3P-50	29,2	24,1	28,7	0,22	0,63	0,26
4P-25	27,9	25,2	27,6	0,33	0,54	0,35
3P-25	26,3	25,7	25,7	0,46	0,50	0,50

Table 9.7 shows the resulting annual system costs for the cost base presented in the Tables 9.2 –9.3 a,b,c. Generally speaking it can be seen that the cost reduction is bigger for the 4P system than for the 3P system with the exception of 3P Model 2 at higher cooling loads, but the difference 4P-3P is mostly only some tenth of MSEK/yr.

Table 9.7: Comparison of annual system costs including the cost differences for different heating plant capacities of the different systems according to Table 9.6. Costs of biomass fuel 80 SEK/ MWh, electricity 350 SEK/MWh.

	Model 1	**Model 2**	**Model 3**
4P-100	23,18	21,71	21,71
3P-100	30,38	27,96	24,67
4P-50	18,51	17,76	17,76
3P-50	22,51	20,65	19,49
4P-25	16,02	15,63	15,82
3P-25	17,27	16,84	16,11

Comparing Tables 9.3 and 9,7 it can be seen that the cost reduction, which is largest for the systems of Model 2, is not changing the mutual ranking of the systems. Hence the real heating plant capacity does not have an important influence on the comparison of 4P and 3P systems at the basic energy cost conditions.

10 Summarising discussion and conclusions

10.1 General discussion

An attempt was made in this study to analyse the possibility of constructing combined district heating and cooling systems with a common return pipe, i.e. three-pipe systems (3P) instead of conventionally four-pipe (4P) systems. In a 3P system the return temperatures varies widely during the season, from being closest to the heating system return temperature in winter time to being close to the cooling system return temperature in summer time, depending on the combination between heating demand and cooling demand. In the chosen example of this paper (50 % cooling load case) the temperatures are about 48°C (winter) and 25 °C (summer), respectively. These lower return temperatures can be used for more effective operation of the biomass cooling plant by using stack gas condensers more effectively. On the other hand, this also means that more energy has to be supplied to the heating circuit and at the same time a higher cooling capacity is necessary to cool down the return flow to the supply temperature. In practice this means that in the 3P system energy is transferred from the heating circuit to the cooling circuit and it must either be rejected by the cooling tower of the cooling circuit or partially recovered by means of a heat pump. This heat pump can be connected so that its condenser is preheating the water of the heating circuit.

The general presumptions for the 3P systems investigated in the study are that the 3P system uses the same pipe trench as the 4 P system and that both the heating and cooling systems, 4P as well as 3P, are layed at the same occasion over the same connection area. However, it was assumed that in average only each 10th customer was connected to the district-cooling network. Furthermore, it was assumed that the network out of the heating/cooling plant is divided into two main stems. In the basic analysis, only the cost of the cooling equipment and pipes were included in the economic analysis, whereas the biomass heating plant was taken to be same in all models. As to the operating cost, the price for electricity was chosen to 350 SEK/MWh and that of biomass fuel to 80 SEK/MWh. Hence this cost difference reflects a typical situation in the Swedish energy market. The investment costs were transformed to annual costs by means of an annuity rate of 8 % (20 years, 5% interest).

In this study, three different models are compared for 4P and 3P system, respectively. ***Model 1*** is the base case for completely separated heating and cooling systems in the 4P case; in 3P heat exchangers are used as recovering parts of the energies, keeping the heating return temperature at 30°C whenever the common return temperature T_R is below this value, and keeping the cooling return temperature at 30°C, whenever T_R is above that temperature.

In ***Model 2***, a heat pump is interconnecting the supply pipes of the cooling and heating circuits, recovering energy by means of its evaporator taking up energy from the cooling supply pipe and preheating the heating medium at the entrance of the biomass plant. The maximum preheating temperature (summer time) is 80°C (two-stage heat pump). By that way a large amount of the heat rejected in Model 1 can be recovered and the total energy of such a system is quite close to that of the totally separated 4P-Model 1 system. However, an important part of the heating energy is electrical (compressor) work, and hence normally this is an expensive way of producing heat.

For avoiding this disadvantage of Model 2, in ***Model 3*** part of the cooling was achieved by a pre-cooling tower (3P system only), bringing down a large amount of the common return temperature to the cooling supply temperature in summer time and leaving only as much energy as necessary for delivering the summer load to the heat pump. This system uses totally more energy than Model 2, but much less electricity and can therefore under some circumstances be more economic than Model 2.

In all 3P cases, extra or larger equipment (heat exchangers, cooling towers, heat pumps) are used compared to the 4P systems. But the cost for this equipment can be compensated by lower costs for omitting one return pipe. So generally speaking, if cooling investment and pipe investment costs are compared, it can be seen that ***3P systems generally have lower investment costs compared to 4P systems.***

The opposite holds for the operating costs (fuel and electricity). Those of the 3P systems are far above (Model 1 and 2) or slightly above (Model 3) that of 4P systems**.** Which also means that ***3P systems are more energy intensive than 4P and therefore 3P systems should in general not be applied for the aspect of energy economy.*** However, the difference for low cooling loads (25 %) is very small and such special cases should be further analysed more in detail.

In some occasions, where ***electricity prices are low,*** the comparison gives results indicating that the total annual costs of the 4P systems and 3P systems are quite similar at low cooling loads (25 %), i.e. cases were the energy consumption of 3P and 4P systems does not differ too much. For these cases with low electricity prices, some favourable conditions for 3P systems could emerge. However, the differences are quite small and such systems need more detailed analysis of every application.

10.2 Conclusions

The following conclusion can be drawn from the analysis:

- In *three-pipe systems* (3P) for combined district heating and cooling systems, the return pipe is common for the returns of the heating and the cooling systems. That means that the return temperature T_R will be on an intermediate level between $T_{R,heating}$ and $T_{R,cooling}$.
- This means in practice that energy is principally transferred from the heating to the cooling circuit and has to be either rejected or recovered by means of heat exchangers or a heat pump.
- For this reason, en energy efficient system will always need more or larger cooling and heat recovery equipment than a four-pipe (4P) system with separate heating and cooling circuits.

- Which also means that 3P systems are more energy intensive than 4P and therefore 3P systems should in general not be applied for the reason of energy economy.
- For the same reason, operating costs of 3P systems will in general be higher than for 4P systems.
- However, the total investment costs for 3P systems can be lower than for 4P systems because of saving costs of one return pipe.
- In the investigated base case with a large difference between costs for supplied heat (80 SEK/MWh) and electricity (350 SEK/MWh), the operating costs will dominate the annual saving of investment costs (annuity 8 %) and hence 3P systems show higher annual costs than 4P systems.
- The study of three different system types show that systems with a heat pump recovering cooling energy from the cooling circuit to the heating circuit (Model 2) are the most economic solutions for both 4P and 3P systems, if the electricity costs are not too high.
- Under conditions, where electricity has a very low price, 3P-Model 2 could be an alternative possibility for combined heating and cooling systems. The energy consumption of this system is only slightly higher compared to 4P systems and the total annual costs can be lower.
- From these examples the general conclusion can be drawn that there might exist conditions on the market on which a 3P system can at least be economically equal to 4P systems. However, these conditions must be analysed carefully and with more detailed cost approximations than those used in this study.
- In systems where fuel prices are very low, 3P systems can use an extra pre-cooling tower for reducing the temperature of the return flow before entering the heat pump evaporator (Model 3). This increases the supplied amount of fuel but can reduce the total costs.
- As a general conclusion we can state that 4P systems should be preferred to 3P systems.

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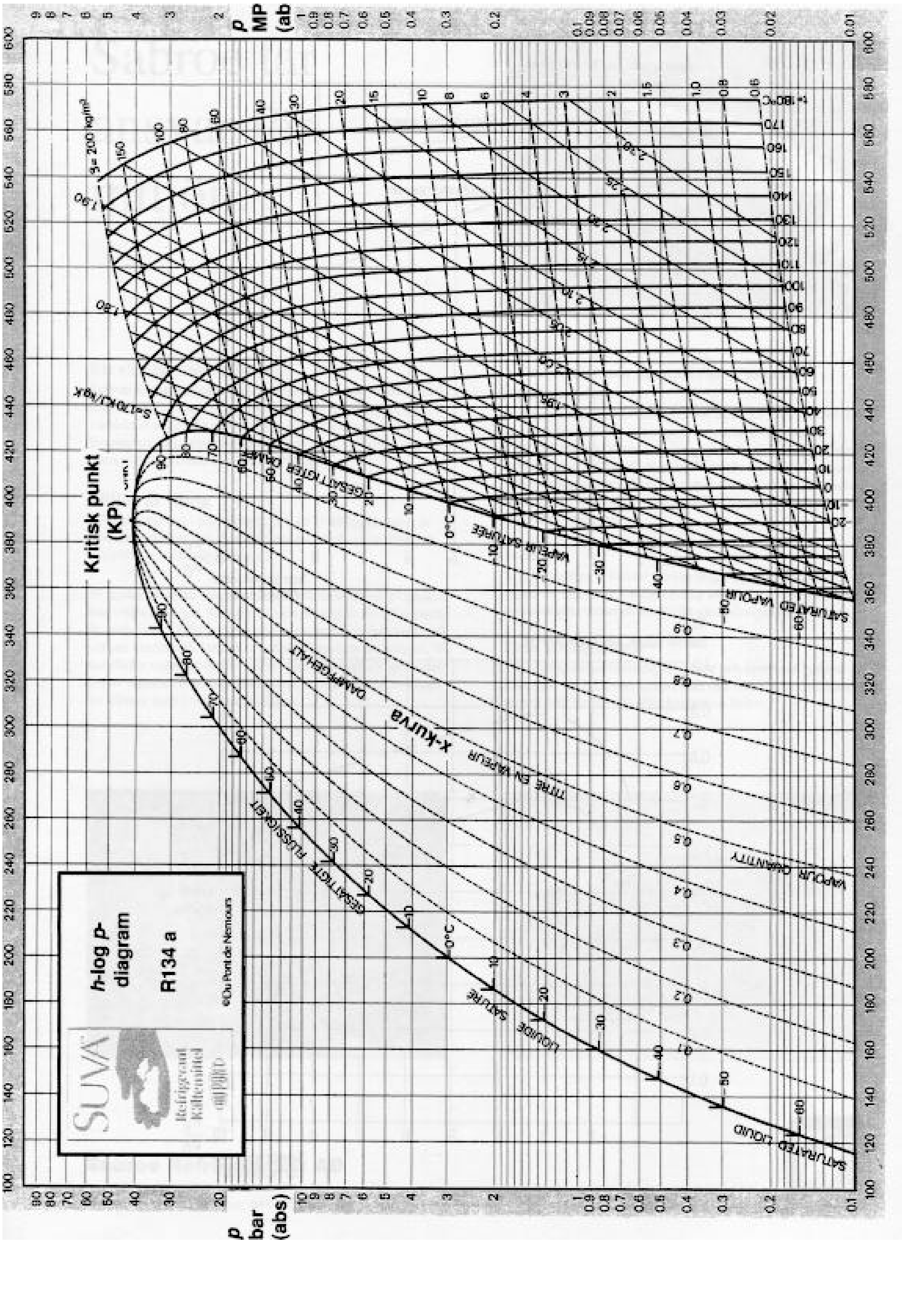
Appendix 1

Phase diagram for the chiller and heat pump cooling medium R 134a [12].



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19	Projekt och Resultat 1994-1997	Anders Tvärne	mars-98

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20	Analys av befintliga fjärrkylakunders kylbehov	Stefan Aronsson Per-Erik Nilsson	mars-98
21	Statusrapport Trycklösa Hetvattenackumulatorer	Mats Lindberg Leif Breitholtz	maj-98
22	Round Robin test av isolerförmågan hos fjärrvärmerör	Ulf Jarfelt	maj-98
23	Mätvärdesinsamling från inspektionsbrunnar i fjärrvärmesystem	Håkan Walletun	juni-98
24	Fjärrvärmerörens isolertekniska långtidsegenskaper	Ulf Jarfelt Olle Ramnäs	juni-98
25	Termisk undersökning av koppling av köldbärarkretsar till fjärrkylanät	Erik Jonson	juni-98
26	Reparation utan uppgrävning av skarvar på fjärrvärmerör	Jarl Nilsson Tommy Gudmundson	juni-98
27	Effektivisering av fjärrvärmecentraler – metodik, nyckeltal och användning av driftövervakningssystem	Håkan Walletun	apr-99
28	Fjärrkyla. Teknik och kunskapsläge 1998	Paul Westin	juli-98
29	Fjärrkyla – systemstudie	Martin Forsén Per-Åke Franck Mari Gustafsson Per-Erik Nilsson	juli-98
30	Nya material för fjärrvärmerör. Förstudie/litteraturstudie	Jan Ahlgren Linda Berlin Morgan Fröling Magdalena Svanström	dec-98
31	Optimalt val av värmemätarens flödesgivare	Janusz Wollerstrand	maj-99
32	Miljöanpassning/återanvändning av polyuretanisolerade fjärrvärmerör	Morgan Fröling	dec-98
33	Övervakning av fjärrvärmenät med fiberoptik	Marja Englund	maj-99
34	Undersökning av golvvärmesystem med PEX-rör	Lars Ehrlén	apr-99
35	Undersökning av funktionen hos tillsatser för fjärrvärmevatten	Tuija Kaunisto Leena Carpén	maj-99
36	Kartläggning av utvecklingsläget för ultraljudsflödesmätare	Jerker Delsing	nov-99
37	Förbättring av fjärrvärmecentraler med sekundärnät	Lennart Eriksson Håkan Walletun	maj-99
38	Ändgavlar på fjärrvärmerör	Gunnar Bergström Stefan Nilsson	sept-99
39	Användning av lågtemperaturfjärrvärme	Lennart Eriksson Jochen Dahm Heimo Zinko	sept-99
40	Tätning av skarvar i fjärrvärmerör med hjälp av material som sväller i kontakt med vatten	Rolf Sjöblom Henrik Bjurström Lars-Åke Cronholm	nov-99

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41	Underlag för riskbedömning och val av strategi för underhåll och förnyelse av fjärrvärmeledningar	Sture Andersson Jan Molin Carmen Pletikos	dec-99
42	Metoder att nå lägre returtemperatur med värmeväxlardimensionering och injusteringsmetoder. Tillämpning på två fastigheter i Borås.	Stefan Petersson	mars-00
43	Vidhäftning mellan PUR-isolering och medierör. Har blästring av medieröret någon effekt?	Ulf Jarfelt	juni-00
44	Mindre lokala produktionscentraler för kyla med optimal värmeåtervinningsgrad i fjärrvärmesystemen	Peter Margen	juni-00
45	Fullskaleförsök med friktionsminskande additiv i Herning, Danmark	Flemming Hammer Martin Hellsten	feb-01
46	Nedbrytningen av syrereducerande medel i fjärrvärmenät	Henrik Bjurström	okt-00
47	Energimarknad i förändring Utveckling, aktörer och strategier	Fredrik Lagergren	nov-00
48	Strömförsörjning till värmemätare	Henrik Bjurström	nov-00
49	Tensider i fjärrkylenät - Förstudie	Marcus Lager	nov-00
50	Svensk sammanfattning av AGFWs slutrapport ”Neuartige Wärmeverteilung”	Heimo Zinko	jan-01
51	Vattenläckage genom otät mantelrörsskarv	Gunnar Bergström Stefan Nilsson Sven-Erik Sällberg	jan-01
52	Direktförlagda böjar i fjärrvärmeledningar Påkänningar och skadegränser	Gunnar Bergström Stefan Nilsson	jan-01
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55	Tryckväxlare för fjärrkyla	Lars Eliasson	mars-01
56	Beslutsunderlag i svenska energiföretag	Peter Svahn	sept-01
57	Skarvtätning baserad på svällande material	Henrik Bjurström Pal Kalbantner Lars-Åke Cronholm	okt-01
58	Täthet hos skarvar vid återfyllning med befintliga massor	Gunnar Bergström Stefan Nilsson Sven-Erik Sällberg	okt-01
59	Analys av trerörsystem för kombinerad distribution av fjärrvärme och fjärrkyla	Guaxiao Yao	dec-01

FORSKNING OCH UTVECKLING – ORIENTERING

1	Fjärrkyla: Behov av forskning och utveckling	Sven Werner	jan-98

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2	Utvärdering av fjärrkyla i Västerås. Uppföljning av Värmeforsk rapport nr 534. Mätvärdesinsamling för perioden 23/5 – 30/9 1996.	Lars Lindgren Conny Nikolaisen	jan-98
3	Symposium om Fjärrvärmeforskning på Ullinge Wärdshus i Eksjö kommun, 10-11 december 1996	Lennart Thörnqvist	jan-98
4	Utvärdering av fjärrkyla i Västerås. Uppföljning av Värmeforsk rapport nr 534. Mätvärdesinsamling för period 2. 1/1 – 31/12 1997.	Conny Nikolaisen	juli-98
5	Metodutveckling för mätning av värmekonduktiviteten i kulvertisolering av polyuretanskum	Lars-Åke Cronholm Hans Torstensson	sept-99

Svenska Fjärrvärmeföreningens Service AB och Statens Energimyndighet bedriver forskningsprogram inom området fjärrvärme hetvattenteknik och fjärrkyla.

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