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MULTI-PARAMETRIC INDICATORS OF ENERGY PRODUCTS IN BIOMASS DISTRICT HEATING SYSTEM

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MULTI-PARAMETRIC INDICATORS OF ENERGY PRODUCTS IN BIOMASS DISTRICT HEATING SYSTEM

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A 30 ECTS credit units Master´s thesis

Supervisors

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Multi-parametric indicators of energy products in biomass district heating system

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ABSTRACT

In this work integration of an organic Rankine cycle (ORC) into district heating system (DHS) is studied. Two scenarios are considered, namely, low temperature cogeneration and high temperature cogeneration, each of them having two cases. Those cases are full time and part time operation of cogeneration. A general model which was built up in Trnsys simulation environment consists of boiler, cogeneration unit, DHS as a heat load and cooling tower. It calculates heat demand due to space heating and consumption of domestic hot water (DHW) and thus determines parameters of operation that would fulfill the needs. In scenario with high temperature cogeneration an absorption cooling is added in order to evaluate impact of increased heat demand in the summer time on operation of cogeneration.

A study case of biomass district heating in Zagorje ob Savi, Slovenia was carried out. Heat demand was calculated and parameters of cogeneration determined. Into first scenario unit with 280 kW maximum electrical power was installed and in the second one with 427 kW. With several performance indicators it was concluded that low temperature cogeneration has low performance whereas high temperature performs better but also economical analysis would be required.

PREFACE

This thesis is submitted to the School for Renewable Energy Science in Akureyri, Iceland as a part of the master (M.Sc.) degree studies. The work on this thesis is the finishing project of the one year master degree study in Iceland in the geothermal energy specialization. It contains work done from September 2010 to February 2011. My supervisor on the project has been prof. Saso Medved, the University of Ljubljana. The thesis has been made solely by the author; most of the text, however, is based on the research of others, and I have done my best to provide references to these sources.

Writing this thesis has been hard but in the process of writing I feel we have learned a lot and here I would like to thank prof. Saso Medved for his support. Beside him I would also like to thank Boris Vidrih for his help with Trnsys.

The biggest thanks are to my family and boyfriend, who supported me all year round in my studies and also during the time while completing this work. Lastly I would like to thank people who supported me during my stay in Iceland, who were always willing to help me and give me good advice.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF USED SYMBOLS

$A_{envelope,i}$	[m^2]	area of walls separating building and ambient air
A_u	[m^2]	area of walls between heated and unheated space
b	[/]	coefficient of reduced heat losses through unheated space
DD	[°D]	Degree day
H_T	[W/K]	heat transfer coefficient due to heat transfer in construction
$H_{V,m}$	[W/K]	heat transfer coefficient due to ventilation
L_s	[W/K]	coefficient of specific heat losses of construction in contact with ground
$\dot{m}_{cw}$	[°C]	mass flow of cooling water
$\dot{m}_{dw}$	[°C]	mass flow of domestic water
$\dot{m}_{hw,cogen}$	[°C]	mass flow of hot water into cogeneration
$\dot{m}_{hw,hx}$	[°C]	mass flow of hot water in heat exchanger
$\dot{m}_{hw,out}$	[°C]	mass flow of hot water out of boiler
$\dot{m}_{to}$	[°C]	mass flow of thermal oil
n	[h^{-1}]	number of air exchange due to invasion of air
η_{nh}	[/]	efficiency of heat sources
η_{rec}	[/]	temperature efficiency of heat exchanger
P_{gen}	[W]	generated electricity
$\dot{Q}_{biomass}$	[W]	heat flow put into boiler with biomass
$\dot{Q}_C$	[W]	cooling load
$\dot{Q}_{cogen,cw}$	[W]	heat flow transferred from cogeneration to cooling water
$\dot{Q}_{ct}$	[W]	heat flow rejected in cooling tower
$\dot{Q}_{gain}$	[W]	internal gains (electrical appliance, people...)
$\dot{Q}_H$	[W]	heat flow for heating
$\dot{Q}_{heating}$	[W]	heat flow transferred from cooling water into building
$\dot{Q}_{hw,cogen}$	[W]	heat flow transferred from hot water to cogeneration
$\dot{Q}_{hw,cw}$	[W]	heat flow from hot water to cooling water
$\dot{Q}_{load}$	[W]	heat flow needed to cover heat losses
$\dot{Q}_{to,cogen}$	[W]	heat flow transferred from thermal oil to cogeneration
T_{cw}	[°C]	temperature of cooling water
$T_{cw,in\ cogen}$	[°C]	temperature of cooling water into cogeneration
$T_{cw,in\ DH}$	[°C]	temperature of cooling water into district heating

$T_{cw,in\ DHW}$	[°C]	temperature of cooling water before DHW
$T_{cw,in\ heating}$	[°C]	temperature of cooling water before heating
$T_{cw,out\ cogen}$	[°C]	temperature of cooling water out of cogeneration
$T_{cw,out\ DH}$	[°C]	temperature of cooling water out of district heating
$T_{cw,out\ DHW}$	[°C]	temperature of cooling water after DHW
$T_{cw,out\ heating}$	[°C]	temperature of cooling water after heating
T_{dw}	[°C]	temperature of domestic water
$T_{hw,in}$	[°C]	temperature of hot water into boiler
$T_{hw,out}$	[°C]	temperature of hot water out of boiler
$T_{hw,out\ cogen}$	[°C]	temperature of hot water out of cogeneration
$T_{hw,out\ hx}$	[°C]	temperature of hot water out of heat exchanger
$T_{to,in}$	[°C]	temperature of thermal oil into boiler
$T_{to,out}$	[°C]	temperature of thermal oil out of boiler
$U_{envelope,i}$	[W/m^{2}K]	overall heat transfer coefficient
U_u	[W/m^{2}K]	overall heat transfer coefficient of walls between heated and unheated space
$V_{building}$	[m^3]	heated volume of building
Φ_{NH}	[W/m^3]	specific heat power of heating system

1 INTRODUCTION

Directive of the European Parliament and of the Council on the promotion of energy from renewable sources states that each member country was to adopt a national action plan for renewable energy sources for a time period between year 2010 and 2020. These plans should set national targets for country shares in renewable energy sources (RES) in gross final energy use for heating and refrigeration, the use of electricity and transport in 2020.

Slovenia has to achieve at least 25% share of RES in the final energy use by 2020. In 2005, the share of renewables in total final energy consumption in Slovenia was 16,2%. Therefore the objectives of the Slovenian energy policy for renewable energy are:

- to provide 25% share of renewables in final energy consumption and 10% renewable energy in transport by 2020, which according to current projections means a doubling of energy production from renewable energy sources relative to baseline 2005,
- a reduction in greenhouse gas emissions of at least 20% below 1990 levels
- a 20% reduction in primary energy use compared with projected levels, to be achieved by improving energy efficiency.
- long-term increase in share of renewables in final energy consumption by 2030

In order to achieve the goals of renewable energy resources will the Government of the Republic of Slovenia support:

- energy renovation of existing building stock, particularly in the public sector and the construction of buildings that represent the most technologically advanced facilities
- replace fuel oil for heating with biomass and other renewable energy
- district heating systems using renewable energy and combined heat and electricity

2 BACKGROUND AND MOTIVATION

Looking at Figure 2.1 one can quickly notice that 40% of final energy consumption is used for transport, but the remaining 60% is in fact (regardless of sector) needed electricity, heat and cold. Therefore there is a great importance to have influence on the production of desired types of energy.

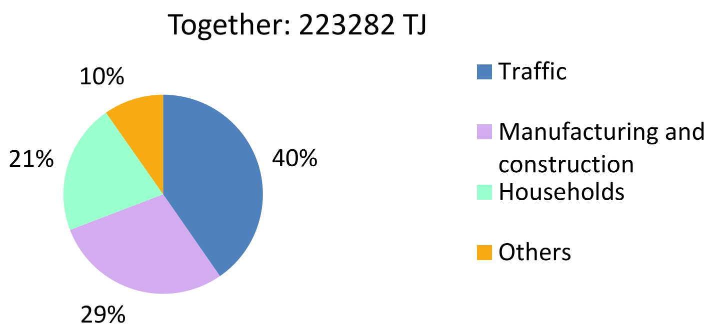


Figure 2.1 Use of final energy in Slovenia in 2008 (Česen, 2009)

HEAT, COLD AND ELECTRICITY DEMAND IN SLOVENIA

To get a sense of how far Slovenia is from the targets (set in the Introduction), this chapter briefly describes current situation of electricity, heat and cold consumption. It also presents the production of electricity in cogeneration units.

2.1 Electricity use in Slovenia

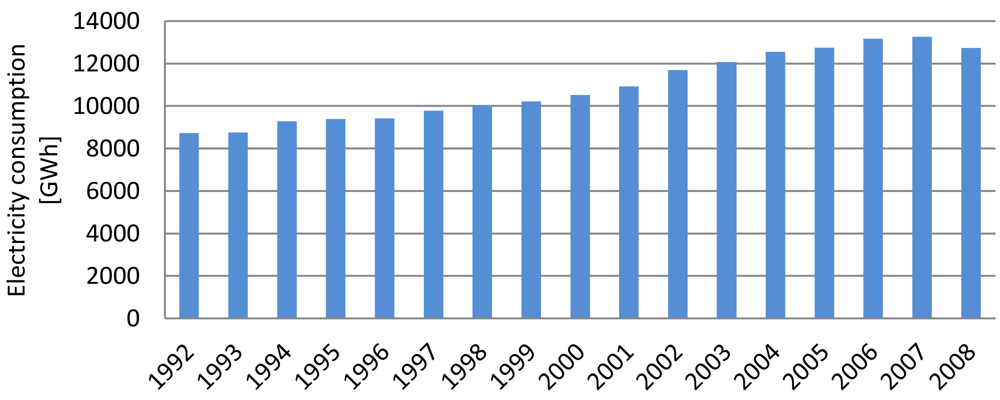


Figure 2.2: Electricity use in Slovenia for a time period between 1992 and 2008 (Česen, 2009)

In 2008 total electricity use was 46% higher than in 1992 and 21% higher than in 2000 (Figure 2.2). In 2007 the lowest annual growth in total of use after 1996 was recorded (0,7%) and a decrease in 2008 happened for the first time in the entire period (Česen,

2009). Probably it happened because of the recession, which has slowed the rate of growth in electricity demand also in 2009.
As it can be seen from Figure 2.3 efficiency of electricity production and efficiency of cogeneration is slowly growing. In 2008 they were 32.6% and 43.3% respectively (Česen, 2009).

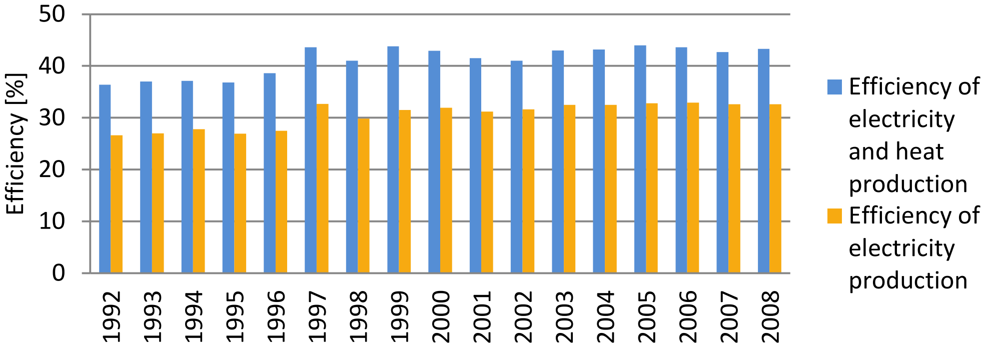


Figure 2.3 Efficiency of cogeneration and only electricity production (Česen, 2009)

2.2 Use of heat

From the figure below we can see that heat consumption varies significantly between years. Greatest influence is exerted by the external temperature, which dictates when the heating season begins. It means there should be a correlation between them and it turns out to be true, as the warmest year were 2002, 2007 and 2009 (Meteorological Agency of the Republic of Slovenia, 2010).

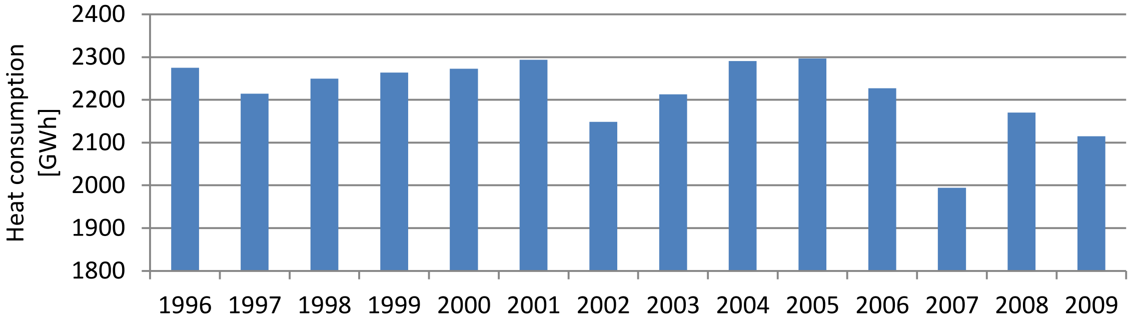


Figure 2.4 Yearly heat consumption in Slovenia (Statistical Office of the Republic of Slovenia)

Figure 2.5 shows monthly heat consumption in district heating in Slovenia for the year 2004. It is logically that the heat consumption is the biggest in the winter time (around 80 GWh) and the smallest in the summer time (around 30 GWh).

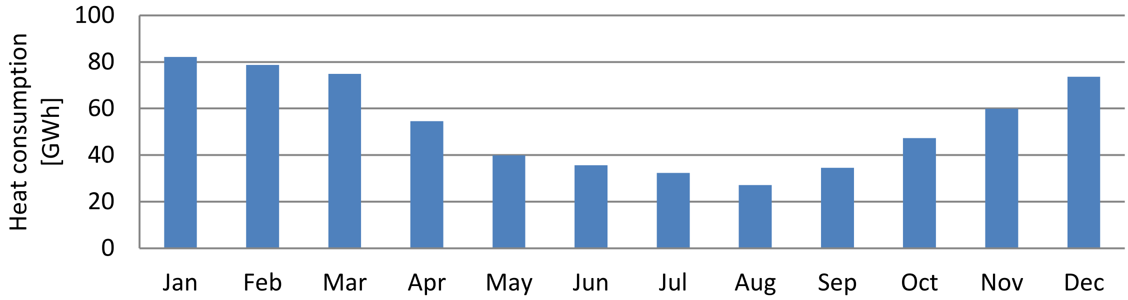


Figure 2.5: Monthly heat consumption in district heating in Slovenia in 2004 (Statistical Office of the Republic of Slovenia)

2.3 Use of cold

There is no data about electricity and heat consumption for cooling. Only available data concerns the number of cooling devices in households. In 2008 90 households out of 1000 were using cooling appliances and until 2020 it is expected that 290 households out of 1000 will have installed cooling appliance. Therefore significant increase in electrical consumption is expected.

2.4 Cogeneration in Slovenia

Electricity production in the systems of combined heat and power (CHP) in 2008 amounted to 1106 GWh, heat production was 3325 GWh. The share of electricity produced in CHP units in total production in Slovenia was 6,7%.Compared with 2007 production increased by almost 2% and compared with 2002 it was higher for 27%.

2.5 Legal framework for cogeneration and renewable energy sources

In order to fulfill requirements set by the directive of the European Parliament presented in Chapter 1, Ministry of the Economy released in 2010 the National Action Plan on Renewable Energy for the period 2010-2020. Some objectives are presented in the following subchapter.

2.5.1 Development of infrastructure for district heating and cooling

The main objectives of the sub-program of the National Energy Program in local energy are formulation, adoption and implementation of intensive development strategies of local energy, which are based on cogeneration of heat and electricity with high efficiency, renewable energy sources and district heating and cooling. Operational sub-objectives of the program are (Ministry of the Economy, 2010):

- to achieve at least 80% of useful energy production with combined heat and power with high efficiency (CHP) or renewable energy use in all systems of local and district heating by 2020
- minimum 20% share of renewable energy sources in the existing district heating that are currently using fossil fuels by 2020

- increase the share of local and district heating in the structure of final energy consumption, despite lower final energy consumption
- construction of new systems of local and district heating based solely on combined heat and power with high efficiency or renewable energy resources and waste heat from industrial processes from 2012 on
- in all new and renovated buildings with consumption of heat above 250 kW_{th} mandatory use of CHP, renewables or district heating where it is technically feasible and economically viable
- installation of at least five district cooling systems by 2015
- transition of five municipalities to the 100% energy supply from renewable energy sources by 2020 and 20 municipalities by 2030
- in the public sector and in all buildings, financed by public funds, to provide heating from CHP, renewable energy sources, or systems of local and district heating:
  - new buildings and major renovations of buildings from 2012
  - the transition of all existing buildings to such a way of heating (40% of buildings until 2020 and 80% until 2030).

2.6 Biomass as a renewable source

One option, how to ensure a sufficient share of renewable energy sources, is to include biomass which represents one of the most important renewable energy sources in Slovenia. Increased use of biomass in modern energy systems is important when reliability and competitiveness of energy supply and environmental protection are taken under consideration. Wood is domestic, renewable and environmentally friendly source of energy, since it is treated as CO_2 neutral. Beside that it is abundant, since 58,5% of Slovenia is covered with forests (Pisek & Dragan, 2009).

2.6.1 Case study: Cogeneration in Ljubljana

Heat and power plant in Ljubljana, Slovenia is an example of replacement of fossil fuels with biomass. Since 2008 in one of the boilers they use wood chips, replacing 20% of coal. Coal is imported from Indonesia because of very low sulphur content. From biomass they produce about 8% of the overall heat and electricity (TE-TOL, 2010).

2.7 Cogeneration

To summarize what was written so far, we could say that actions to reduce CO_2 emissions could be carried out on two fronts. As already mentioned we can increase the share of renewable sources or affect the energy efficiency. The latter is of great importance, since a heavy burden on the environment has electricity generation, where the level of pollution depends on the fuel efficiency. Problem which arises is that the conversion of primary energy into electrical energy has very low efficiency. For example, a new power plant with gas turbines operating in combined cycle is reaching 60% efficiency and a new coal fired power plant between 40 and 50%. But in older technologies efficiencies are much lower, reaching approximately 30%. On the other hand boilers for domestic hot water or water for heating have efficiencies higher than 90%.

If we want to efficiently convert energy into electricity or in other words if we want to increase efficiency of heat and electricity production we can combine these two separated systems (‘monogeneration’) into cogeneration as shown in the Figure 2.6.

Cogeneration (Combined heat and power, e.g. CHP) increases the efficiency of electricity generation since it beneficially uses thermal heat that would otherwise be lost. It encompasses a range of technologies, but will always include an electricity generator and a heat recovery system. Through the utilization of the heat, the efficiency of cogeneration plant can reach 90% or more (The European Association for Promotion of Cogeneration, 2001) and is defined as follows:

$$\eta = \frac{P + Q_h}{Q_{biomass}} \tag{1}$$

Where P is amount of generated electricity in kWh, $Q_{biomass}$ heat supplied with biomass and Q_h useful heat (for heating). It needs to be pointed out that with this equation heat and electricity are brought to the same quality level. In other words they have the same value which is not true since electricity can be 100% transformed in other types of energy, whereas heat cannot be. How much heat (Q) can be possibly converted into useful work is defined with exergy (B). As it can be seen from Equation this value depends on the temperature level of heat (T_{source}) and ambient temperature (T_a).

$$B = Q\left(1 - \frac{T_a}{T_{source}}\right) \tag{2}$$

This study will not include exergy analysis since financial support only depends on amount of heat and electricity supplied. But it is also true that amount of heat supplied to consumers actually depends on temperature level of it since certain temperature has to be provided for heating.

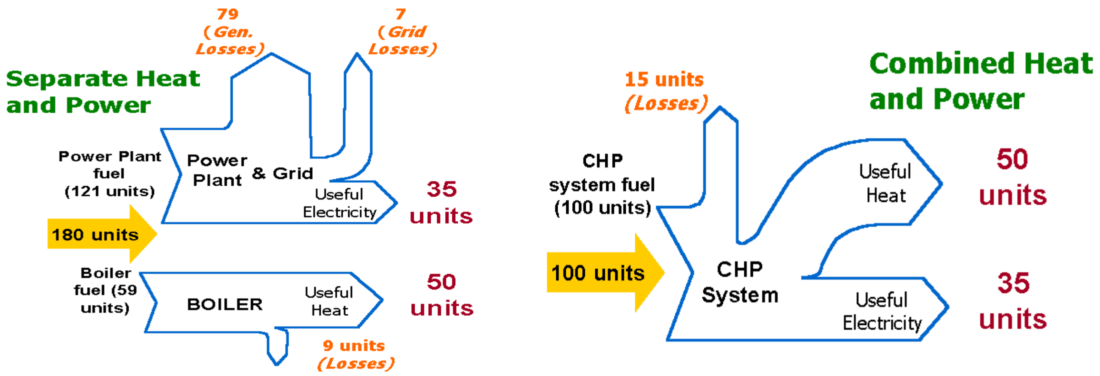


Figure 2.6: Comparison between conventional and combined heat and power plant (Green Power Management)

Because transporting electricity over long distances is easier and cheaper than transporting heat, cogeneration installations are usually sited as near as possible to the place where the heat is consumed and, ideally, are built to a size to meet the heat demand. Cogeneration therefore offers energy savings ranging between 15-40% when compared against the supply of electricity and heat from conventional power stations and boilers (The European Association for Promotion of Cogeneration, 2001).

Two examples of such a system are combined heat and power plants in Lienz based on the ORC technology (see 3.3.1 for details on ORC technology) and in Husavik based on the Kalina Cycle (see 3.3.2 for details on Kalina cycle).

2.7.1 CHP Lienz, Austria

The municipality of Lienz has implemented a showcase project of environmental engineering. The biomass district-heating power plant achieves considerable reduction in air pollutants; carbon monoxide – minus 480 tons, sulphur dioxide – minus 16 tons, and dust – minus 12 tons per heating season. The first biomass power plant (Lienz I) with an overall capacity of 25.5 MW obtains the required energy from biomass, extra-light fuel oil and solar energy. The plant does not only produce heat energy from biomass, but also green electricity by means of an ORC process (Organic Rankine Cycle) with a nominal electric capacity of 1 MW. With the biomass power plant Lienz II, the total capacity of heat production has been raised by 19 MW and a buffer system installed to even out peak loads. The new plant was also equipped with a cogeneration on the basis of an ORC process with a nominal electric capacity of 1.5 MW (Stadtwaerme Lienz).

Table 2.1 Technical data of the biomass CHP plant Lienz (Obernberger, Thonhofer, & Reisenhofer, 2002).

Heating medium	Thermal oil	Yearly production of electricity	11000 MWh/a
Inlet temperature	300°C	Yearly production of heat	74000 MWh/a
Outlet temperature	250°C	Installed thermal boiler output	44.5 MW
Working medium - ORC	Silicon oil	Installed electric capacity	2.5 MW
Cooling medium	Water	Solar thermal collector	630 m²
Inlet temperature	80°C	Buffer storage	400 m³
Outlet temperature	60°C		
Net electrical efficiency at nominal load	18,0%	Investment costs	38.330.000 EUR
Thermal efficiency at nominal load	80,0%		

Figure 2.7 Thermal oil boiler (Stadtwaerme Lienz)

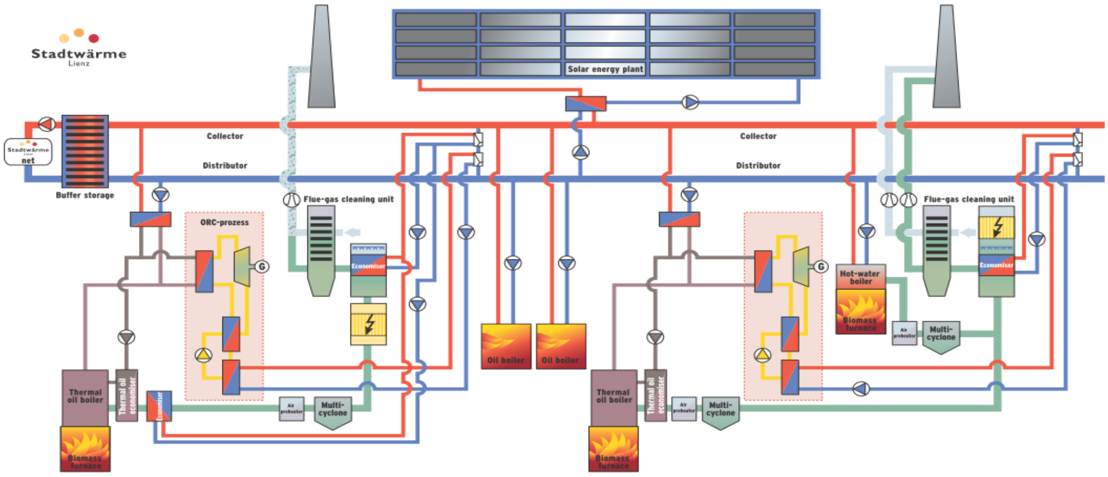


Figure 2.8 Scheme of the biomass CHP plant Lienz (Stadtwaerme Lienz)

2.7.2 Husavik

The plant was installed in 1999 near the small town of Husavik, in Northern Iceland. It produces 2 MWe using the so-called Kalina cycle which is based on a recirculation loop where a mix of water and ammonia is used as the working fluid. Geothermal fluid at about 125°C from the Hveravellir area, about 16 km from Húsavík, is used to produce electricity. Once the geothermal fluid has been harnessed for electricity production, about 100 l/s of 80°C hot water and 200 l/s of 30°C pre-heated cold water are available for the district heating system on the one hand and for potential use in various cultivation farms on the other hand. Some technical data are given in Table 2.2 scheme of power plant and consumers is given in Figure 2.10.

Table 2.2 Technical data of Husavik power plant (Verkis, 2009)

Electricity production	1,7 MW_e
Hot water	41,3 MW_{th}
District heating	13,0 MW_{th}

Green houses	2,4 MW_{th}
Snow melting	1,0 MW_{th}
Industry	1,6 MW_{th}
Fish farming	5,1 MW_{th}
Bathing facility	18,2 MW_{th}
Source	
Temperature	125°C
Mass flow	98 l/s

Figure 2.9 Turbine, tank, separator and pump in geothermal power plant in Husavik (Exorka, Magnus Gehringer, 2006)

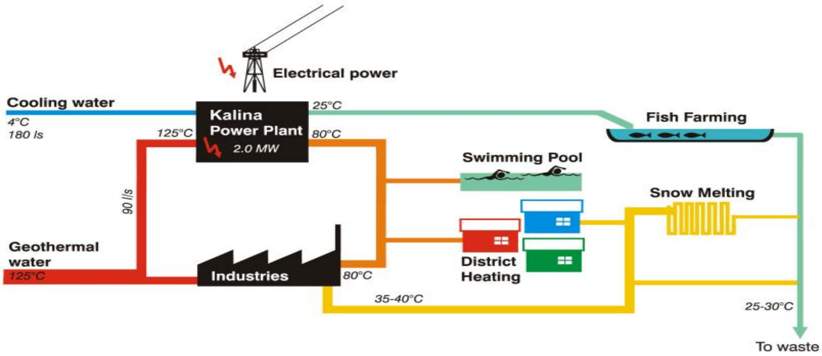


Figure 2.10 Use of different type of energy in Kalina power plant in Husavik (Exorka, Magnus Gehringer, 2006)

It needs to be highlighted that the efficiency of a cogeneration depends on several factors. One of them is the amount of heat generated. Ideally, such a system would work constantly throughout the year, however, a problem occurs during the summer when there is no heat load or it is much smaller (see Figure 2.5). This suggests that it would be wise to introduce systems that require heat during the summer. One possibility for such a system is absorption cooling.

2.8 Cooling of buildings

2.8.1 Cooling in relation to health

Some studies have been done in Europe and North America examining relationship between heatwaves and mortality with elderly people. Evidence for that is an extreme heatwave in August 2003 in Europe, which has attested to the lethality of such events. People living in urban environments are at greater risk than those in non-urban regions. Thermally inefficient housing and the so-called urban heat island effect (whereby inner urban environments, with high thermal mass and low ventilation, absorb and retain heat) amplify and extend the rise in temperatures (especially overnight). In 2003 in Paris many nursing homes and other assisted-living and retirement communities were not air-conditioned and elderly residents might not have been promptly moved to air-conditioned shelters and rehydrated with fluids. If we take a look at population pyramid for the year 2020 and compare it to the current situation, then we can summarize that the population is getting older (Figure 2.11).This suggests that the cooling of buildings in summer time is very important, not only for comfort, but also because it may prevent premature deaths.

Figure 2.11: Predicted population pyramid for Slovenia for 2020. It can be seen that the majority of population will be between 30 and 70 years (Statistical Office of the Republic of Slovenia).

2.8.2 Energy demand of cooling

In most European countries the trend of increasing energy use for cooling and air-conditioning of buildings is expected to continue in the coming decades. For example, sale of smaller devices for cooling of individual rooms (with a cooling power of 5 kW) was in 2008 in the world at around 82 million units, of which Europe has sold 8,6 million(Medved & Carvalho, 2009). Therefore it is not surprising that in some cases peak load in the summer (Figure 2.12) is caused by the power consumption of refrigeration and air-conditioning systems. Although newly developed systems of larger capacity have high efficiencies, we can assume that the standard air-conditioning systems in existing buildings produce on average only 3 kWh cold from 1 kWh of electricity consumed. If a conversion from primary energy to electricity is 2,5:1 (for Slovenia), it consequently means that we spend on average almost 1 kWh of primary energy to produce 1 kWh of usable cold.

In general these two problems (increased demand of electricity and increase of peak demand) are equally important and our aim is to solve them simultaneously. Cooling systems with the main or auxiliary source of heat are one of the possibilities to provide cooling. At the same time surplus heat from cogeneration will be used.

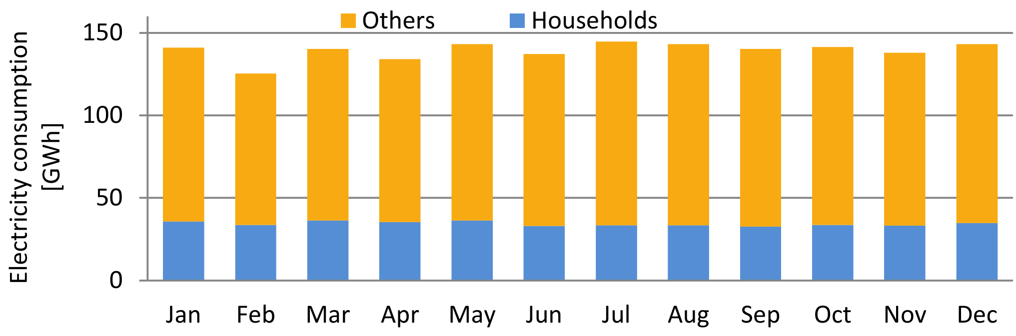


Figure 2.12. Monthly electricity consumption in Municipality of Ljubljana, Slovenia

Even if it is not obviously, one can notice that the peak demand is in July. This is mainly caused by industry (in Figure 2.12 referred as 'others'), which has high cooling load. Another example of peak demand occurring in the summer time is electricity production in U.S.A.

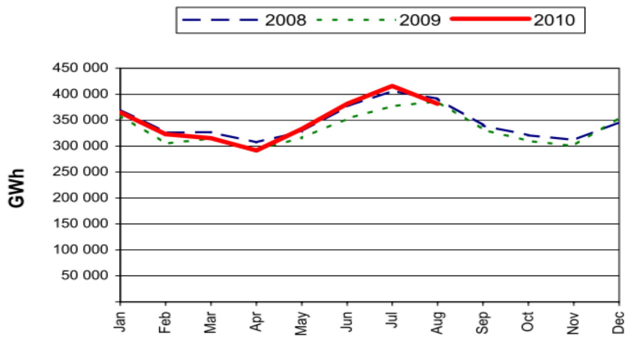


Figure 2.13 Electricity production in the U.S.A. (International Energy Agency, 2010)

2.8.3 Case study: District cooling in the municipality of Velenje, Slovenia

In addition to district heating, which in Slovenia is already well established, also district cooling system deployment has begun. In August 2008 they carried out a pilot project and the Municipality of Velenje building was provided with cold from absorption cooling aggregate. Supplier of heat provides an operating regime in the range from 110ºC to (inlet) and 70ºC to 75ºC (outlet). During the summer time when only heat for domestic hot water is needed the whole system operates at inlet temperature regime from 105 to 110ºC and in the return 70ºC (Zager & Cvet, 2009). Specifications of the system are presented in

Table 2.4 and Table 2.3, investment cost for design of absorption cooling station and first phase of cooling distribution system was 1.175.039 EUR (Zager & Cvet, 2009).

Table 2.3: Specification of the district cooling system (Zager & Cvet, 2009)

Area of cooled buildings	*23.495 m^2*
Yearly cold consumption	1.160.400 kWh/a
Yearly electricity consumption	68.760 kWh/a
Yearly heat consumption	1.546.800 kWh/a

Table 2.4:Specification of absorption cooling station (Zager & Cvet, 2009)

MANUFACTURER	*BROAD*	***COOLING WATER***	
Model	BDH84 X-87/1005-35/28-100	Temperature regime	35/28 ºC
COP	0,775	Flow rate	280 m^3/h
Cooling capacity	980 kW	**HOT WATER**	
CHILLLED WATER		Temperature regime	105/87 ºC
Temperature regime	12/7 ºC	Flow rate	62,5 m^3/h
Flow rate	168 m^3/h	Maximum temperature	125 ºC
		Electrical power	5 kW

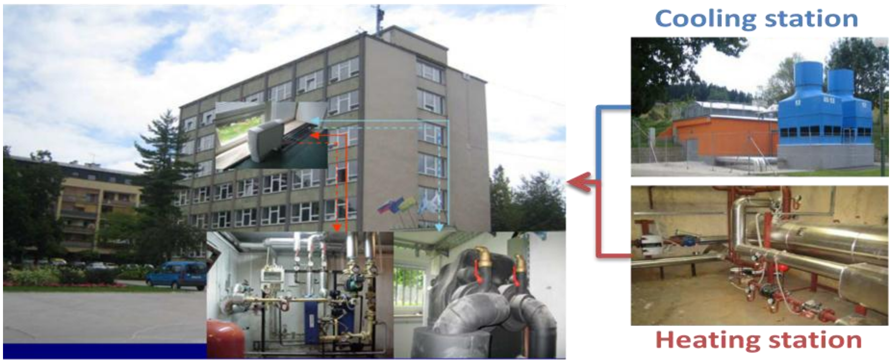


Figure 2.14 District heating and cooling of a building in Velenje, Slovenia (Zager & Cvet, 2009)

2.8.4 Case study: Trigeneration for Technology Park Brdo, Ljubljana

Trigeneration for Technology Park Brdo worth 1.55 million Euros provides electricity, heat and cold for the first seven buildings (21.800 m^2) of Technology Park Brdo, Ljubljana. This system has better electrical efficiency compared to conventional production and therefore saves 1,4 GWh/year and 376 tons/year of CO2 emissions. For electricity generation is used gas engine, which also provides heat but not sufficiently therefore they also have 2 gas boilers. Absorption and compression chillers provide cold from April to October as it can be seen in the Figure 2.15.

Table 2.5 Technical data for trigeneration in Technology Park Brdo

Gas engine Jenbacher	739 kW_{th}, 625 kW_e
Gas boiler Viessmann	2 x 1120 kW_{th}
Absorption chiller Carrier 16JB	600 kW
Compression chiller Carrier	1 x 942 kW cold

	Electricity consumption: 190 kW_e
	COP=4,95
	1 x 1.057 kW cold
	Electricity consumption: 215 kW_e
	COP=4,93
Cooling towers	1 x 1465 kW
	3 x 1130 kW

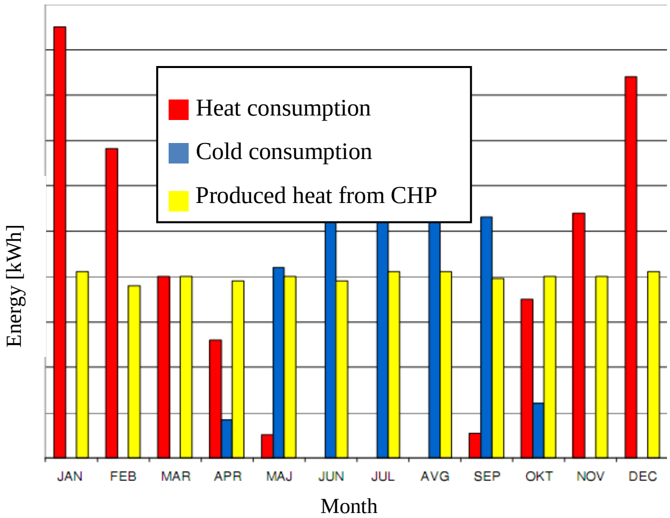


Figure 2.15 Heat production, heat and cold consumption (Valentinčič, 2008)

Figure 2.16 Gas engine (Valentinčič, 2008)

2.9 Incentives for renewable energy

Technologies that pollute less have advantage since Slovenian energy policy offers financial aid for them. If company or seller has the status of qualified electricity producer (must satisfy certain conditions, efficiency must be at least 75%) then the country has decreed a higher purchase price from the market. Duration of the provision of support is determined in the decision to grant support (usually 10 years). Financial support is carried out as feed-in tariff or financial aid for current operations (hereinafter referred to as premium). To determine the amount of financial aid both of them use reference costs, that consist of fixed and variable costs. The fixed part of the reference costs is determined every 5 years or sooner if significant changes occur. The variable component of the reference costs is reviewed annually or more frequently on the basis of forecasted market prices of energy. When determining feed-in tariff or the amount of operating support throughout the duration of contract the fixed component does not change.

2.9.1 Feed-in tariff

Whatever the price of electricity in the market is, support centre buys all net electricity produced by CHP plant at guaranteed prices. It is only valid for micro and mini cogeneration units. Feed-in tariff consists of fixed and variable component:

- Fixed part of the feed-in tariff is equal to the fixed part of the reference costs and does not change during the contract.
- Variable part of the feed-in tariff is equal to the variable part of the reference costs, which changes annually or more frequently depending on the price of fuel.

The amount of money for unchangeable and changeable part for mini cogeneration units can be obtained from the following table as well as a feed-in tariff. For micro units it is determined for each case separately.

Feed-in tariff [EUR/MWh]	**Mini (< 1 MW)**	
	Operating hours < 4000 hours	**Operating hours > 4000 hours**
Unchangeable part	293,27	186,62
Changeable part	33,43	33,43
Price together (2009)	326,70	220,05

Table 2.6. Feed-in tariff for CHP units fueled by biomass (Official gazette of the Republic of Slovenia, 2009)

2.9.2 Premium

Premiums allocated to the net electricity generated by producers that sell it by themselves on the market or consume it by themselves, providing that the production cost of electricity is higher than the price of electricity in the electricity market. It is valid for all sizes of units.

Financial aid for current operation

The amount of operating support is determined by subtracting the price of electricity which could be achieved in the electricity market from the reference costs. It depends on several factors such as type of energy source, size of CHP plant and operating hours.

According to the number of operating hours in the reporting period or calendar year CHP units are classified into two groups:

- First group - CHP units operating up to 4000 hours
- Second group - CHP units operating over 4000 hours

CHP plants that are classified into the first group may obtain financial support only for the annual amount of electricity produced for which they have received a certificate of origin and which do not exceed the product of: the nominal electric power plant x 4000 hours.

In the following table are presented operating supports for biomass CHP units of different sizes and operating hours.

Table 2.7 Operating support for CHP units fueled by biomass (Official gazette of the Republic of Slovenia, 2009)

Size of a production unit	**Operating hours <4000 hours**	**Operating hours >4000 hours**
Mini (<1 MW)	269,50	160,25
Medium - lower (from 1 MW to including 5 MW)	192,28	111,17
Medium - higher (from 5 MW to including 25 MW)	126,56	67,99
Large - lower (from 25 MW to including 50 MW)	93,31	46,46

3 BEST AVAILABLE TECHNOLOGIES FOR COGENERATION

3.1 Internal combustion engine

The reciprocating engines used in cogeneration are internal combustion engines operating on the same familiar principles as their petrol and diesel engine automotive counterparts.

3.1.1 Compression ignition engine

Compression ignition engine for large-scale cogeneration are predominantly four-stroke direct-injection machines fitted with turbochargers and intercoolers. Diesel engines will accept gas oil, HFO and natural gas. Shaft efficiencies are 35 to 45%, and output range is up to 15 MWe (The European Association for Promotion of Cogeneration, 2001). In general, engines up to about 500 kWe (and sometimes up to 2 MWe) are derivatives of the original automotive diesels, operating on gas oil and running at the upper end of their speed range. Engines from 500 kWe to 20 MWe evolved from marine diesels and are dual-fuel or residual fuel oil machines running at medium to low speed.

3.1.2 Spark ignition engine

Spark-ignition engines are derivatives of their diesel engine equivalents. They can use exhaust gases for heat recovery purposes; thus plants can be built with 160°C hot water of 20 bar steam output. Traditionally, shaft efficiency was lower that for compression ignition engines, at between 27% and 35%, and the output range was limited to a maximum of around 2 MWe. The new above 3 MWe spark ignition engines use pre-chamber, where the mixture is stoichometric.

3.2 Gas turbine

The fuel is burnt in a pressurized combustion chamber using combustion air supplied by a compressor that is integral with the gas turbine. In conventional gas turbine, gases enter the turbine at a temperature range of 900 to 1000°C and leave at 400 to 500°C (The National Energy Conservation Centre). The very hot pressurized gases are used to turn a series of turbine blades, and the shaft on which they are mounted. Gas turbine produces mechanical energy. Residual energy in the form of a high flow of hot exhaust gases can be used to meet, wholly or partly, the thermal (steam) demand of the site. Waste gases are exhausted from the turbine at 450°C to 550°C (The National Energy Conservation Centre), making the gas turbine particularly suitable for high-grade heat supply.

3.3 Steam turbine

Steam turbines have been used as prime movers for industrial cogeneration systems for many years. High-pressure steam raised in a conventional boiler is expanded within the turbine to produce mechanical energy, which may then be used to drive an electric generator. The power produced depends on how much the steam pressure can be reduced through the turbine before being required to meet site heat energy needs. For viable power generation, steam input must be at a high pressure and temperature. Residual heat output is relatively low grade. Typical inlet steam conditions are 42 bar/400°C or 63 bar/480°C (The European Association for Promotion of Cogeneration, 2001).

In district heating cogeneration schemes, the turbine condenser may be operated near or even above atmospheric pressure. This ensures that the condenser cooling water picks up enough heat to supply the district heating circuit. Nevertheless, some pass-out steam may still be needed to top up the final temperature of the circulating water.

3.3.1 Organic Rankine cycle

The Organic Rankine cycle (ORC) uses an organic, high molecular mass fluid with a liquid-vapor phase change, or boiling point, occurring at a lower temperature than the water-steam phase change. The fluid allows recover heat from low temperature sources such as industrial waste heat, geothermal heat, solar ponds etc. in a Rankine cycle (Figure 3.1). The low-temperature heat is converted into useful work, which can itself be converted into electricity.Special advantages of the ORC process are its excellent partial load and load changing behavior, which is in particular beneficial in heat controlled operation and for the significant load fluctuations occurring in district heating grids (Stadtwaerme Lienz). The very stable operating behavior as well as the ease of maintenance rank among its further strengths.

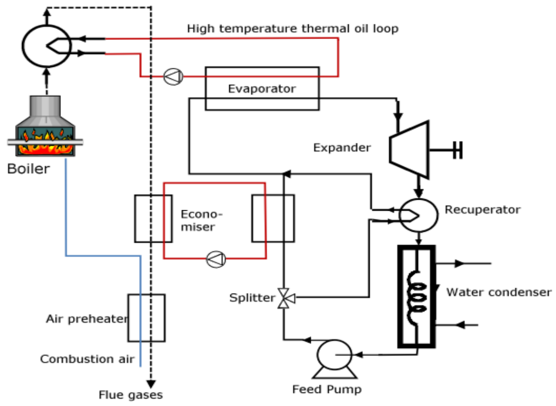


Figure 3.1: Schematic representation of an ORC CHP system (Quoilin & Lemort, 2009)

An important feature for the classification of the ORC cycles is the shape of the saturated vapor line in the temperature versus entropy diagram (T-s diagram). The saturated vapor line may either lead to a bell-shaped saturation curve like in Figure 3.2 or Figure 3.3 or it might be overhanging like in Figure 3.3. The shape depends on the type of working fluids and figures here are only representative (not for a specific fluid). Some common fluids for ORC are toluene, butane, pentane, ammonia, refrigeration fluids, silicone oils…

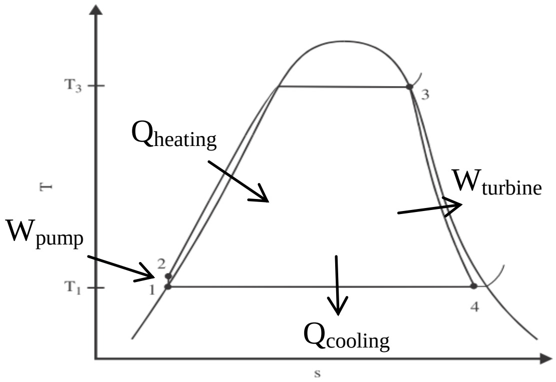


Figure 3.2 ORC cycle for a fluid with bell-shaped saturation curve and saturated vapor at the turbine inlet (Saleh, Koglbauer, Wendland, & Fischer, 2007)

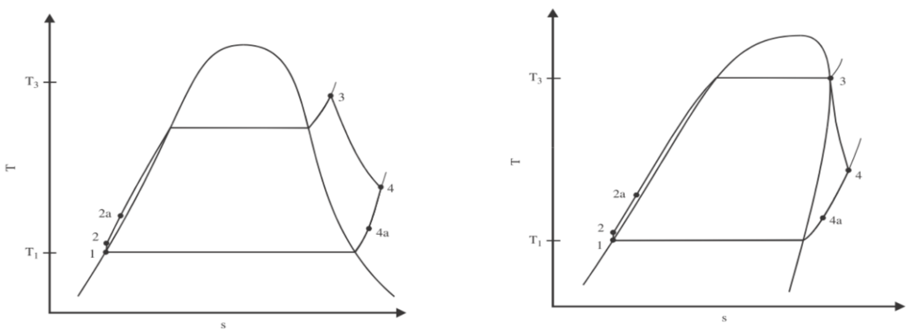


Figure 3.3 a.) ORC cycle for a fluid with bell-shaped saturation curve and superheated vapour at the turbine inlet b.) ORC cycle for a fluid with overhanging saturation curve and saturated vapour at the turbine inlet (Saleh, Koglbauer, Wendland, & Fischer, 2007)

3.3.2 Kalina cycle

The Kalina cycle is a thermodynamic cycle for converting thermal energy to mechanical power, optimized for use with thermal sources which are at a relatively low temperature compared to the heat sink (or ambient) temperature. The cycle uses a working fluid with at least two components (typically water and ammonia) and a ratio between those components is varied in different parts of the system to increase thermodynamic efficiency. There are multiple variants of Kalina cycle systems specifically applicable for different types of heat sources.

Since the phase change from liquid to steam is not at a constant temperature, the temperature profiles of the hot and cold fluids in a heat exchanger can be made closer, thus making the global efficiency of the heat transfer bigger. For this reason, this cycle is increasingly popular in geothermal power plants, where the hot fluid is very often below 100°C.

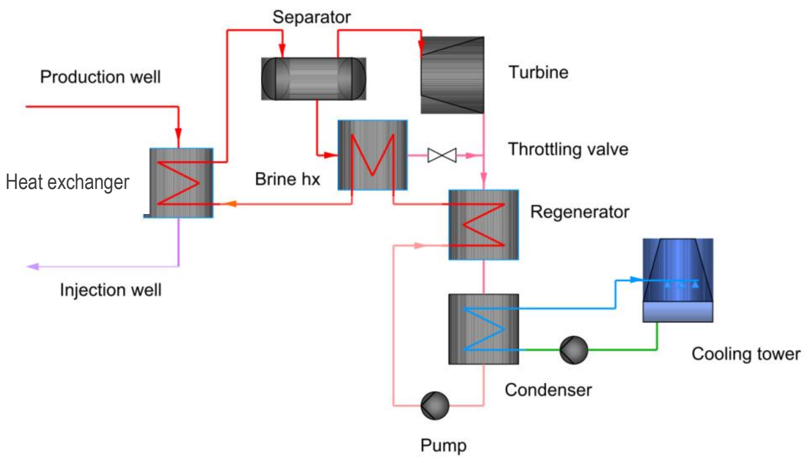


Figure 3.4 Schematic representation of Kalina cycle (Valdimarsson, 2003)

3.4 Combined cycle

Some large systems (power output generally greater than 3 MWe) utilize a combination of gas turbine and steam turbine, with the hot exhaust gases from the gas turbine being used to produce the steam for the steam turbine. The main advantage of CCGT cogeneration is its greater overall efficiency in the production of electricity, compared with the alternatives described above.

3.5 Comparison of different technologies

This chapter summarizes the technologies that are suitable for the cogeneration and it exposes their characteristics. Different indicators are compared in the Table 3.1, then advantages and disadvantages are presented (Table 3.2) and in the end approximate costs are given (Table 3.3).

Table 3.1. Comparison of different technologies for CHP (Žiher, 2003) (Biomass energy center) (Schuster, Karellas, Kakaras, & Spliethoff, 2009) (Obernberger, Thonhofer, & Reisenhofer, 2002)

PRIME MOVER	FUEL USED	SIZE RANGE (MWe)	HEAT TO POWER RATIO	ELECTRICAL EFFICIENCY	TYPICAL OVERALL EFFICIENCY	HEAT QUALITY
Extraction steam turbine	Any fuel	1 to 100+	3:1 to 8:1	10 - 20%	Up to 80%	Steam at 2 press or more
Back pressure steam turbine	Any fuel	0,5 to 500	3:1 to 10:1	7 - 20%	Up to 80%	Steam at 2 press or more
Combined cycle gas turbine	Gas, Biogas, Gasoil, LFO, LPG, Naphtha	3 to 300+	1:1 to 3:1	35 - 55%	73 – 90%	Medium grade steam, high temperature hot water
Open cycle gas turbine	Gas, Biogas, Gasoil, LFO, Heavy fuel oil (HFO), LPG, Naphtha	0,25 to 50+	1,5:1 to 5:1	25 - 42%	65 – 87%	Medium grade steam, high temperature hot water
Copression ignition engine	Gas, Biogas, Gasoil, Naphtha, HFO, LHO	0,2 to 20	0,5:1 to 3:1	35 – 45%	65 – 90%	Low pressure steam, low and medium temperature hot water
Spark ignition engine	Gas, Biogas, LHO, Naphtha	0,003 to 6	1:1 to 3:1	25 – 43%	70 – 92%	Low and medium temperature hot water
Organic Rankine cycle	Any fuel	0,004 to 7,5	5:1	9-21%	~80%	Low and medium temperature hot water,

Table 3.2. Advantages and disadvantages of different technologies for CHP (Žiher, 2003)

PRIME MOVER	PROS	CONS
Gas turbine	High reliability	Relatively short life time
	Heat at high temperature	Fuel limitations
	Wide spectrum of use	
	Exhaust gases with large amount of oxygen	
Steam turbine	High overall efficiency	Low electricity generation
	High level of safety	High investment costs
	Possible use of every fuel	
	Long life time	
	Large spectrum of power	
Ignition engine	High electricity generation	Heat at low temperature
	High electrical efficiency	High O&M costs
	Low investment costs	

Table 3.3 Approximate costs for four different CHP technologies (Žiher, 2003)

Technology	**Steam turbine**	**Gas turbine**	**Combined cycle**	**Reciprocating engine**
Investment cost (ECU/kWe)	1500-1000	1200-530	900-450	960-770
O&M cost	2,3-1,5	5,4-4,6	5,4-4,6	5,8-9,2
Cost per MWe (ECU/MWe) (price of fuel: 13,2 ECU/MWh lower heating value)	20-15	33-30	33-30	29-26
Simple payback time (operating: 7000 h/year, price of electricity: 77 ECU/MWh)	3,5-4	2-3	2-3	2-3
Life time	30	15	15	10

3.6 Manufacturers of the ORC-units

In the Table 3.4 are listed different manufacturers of the ORC technology for waste heat recovery and biomass applications. Available sizes are from 10 to 2400 kW of electrical power, needed temperatures are ranging between 88°C and 350°C.

Table 3.4 Manufacturers of the ORC units

Company	**Enef Tech**	**ElectraTherm**	**Tri-O-Gen**	**BEP Europe**	**Pratt & Whitney**	**Adoratec**	**Turboden**	**Ormat**
Website	www.eneftech.com	www.electratherm.com	www.triogen.nl	www.bepenergy.com	www.pratt-whitney.com	www.adoratec.com	www.turboden.eu	www.ormat.com
Country	Switzerland	U.S.A.	Netherlands	Belgium	U.S.A.	Germany	Italy	U.S.A.
Model	ENEFCOGEN green	Green Machine	n.a.	n.a.	PureCycle	n.a.	n.a.	n.a.
Power [kW]	10-30	30-50	60-165	55, 250	280	300-2400	427-2304	n.a.
Working fluid	n.a.	n.a.	Toluene		R245fa	Thermal oil	Thermal oil	n.a.
Temperature of the heat source [°C]	125-150 (waste heat)	88-116 (waste heat)	350	95/85 (waste heat)	90-149 (waste heat)	320 (biomass application)	300 (biomass application)	n.a.
Required heat [kW]	53-300	430-715	n.a.	700, 3500	n.a.	n.a.	2300-12020	n.a.
Cooling requirement [°C]	20-30	10-21	35-55	20	4-43	n.a.	n.a.	n.a.
Condensing load [kW]	45-255	400-665	350-700	640, 3200	n.a.	n.a.	1854-9601	n.a.

3.6.1 Cost of the ORC units

Prices of ORC units are listed in Table 3.5.

Table 3.5 Prices of ORC-modules (Vanslambrouck, 2010)

Manufacturer	**Size [kW_{el}]**	**Price [€/kW_{el}]**
Turboden	500	1900
	1000	1350
	2000	950
Pure Cycle	250	1350
Tri-O-Gen	150	4300
BEP-Europe	50	2400 module 4000 installed
	250	Lower price compared with the 50 kW unit

4 BEST AVAILABLE TECHNOLOGY FOR COOLING – ABSORPTION COOLING

Cold can be produced with devices that are using electric compressor or devices using heat. Latter ones use sorption, which actually refers to the action of absorption or adsorption. Absorption is the incorporation of a substance in one state into another of a different state (e.g. liquids being absorbed by a solid or gases being absorbed by a liquid). Adsorption is the physical adherence or bonding of ions and molecules onto the surface of another phase (e.g. reagents adsorbed to solid catalyst surface).In principle both technologies are used in refrigeration, whereas absorption cooling is more common.

4.1 Description of the process

The absorption cycle is a process by which the refrigeration effect is produced through the use of two fluids and some quantity of heat input, rather than electrical input as in the more familiar vapor compression cycle. The most widely used refrigerant and absorbent combinations in absorption refrigeration systems (ARS) have been ammonia-water and lithium bromide-water. The lithium bromide-water pair is available for air-conditioning and chilling applications (over 4°C, due to the crystallization of water). Ammonia-water is used for cooling and low temperature freezing applications (below 0°C).

4.1.1 How it works?

In the Figure 4.1 is presented whole chilling cycle of an absorption chiller, which is followed by detailed figures of each part and explanation about how it works.

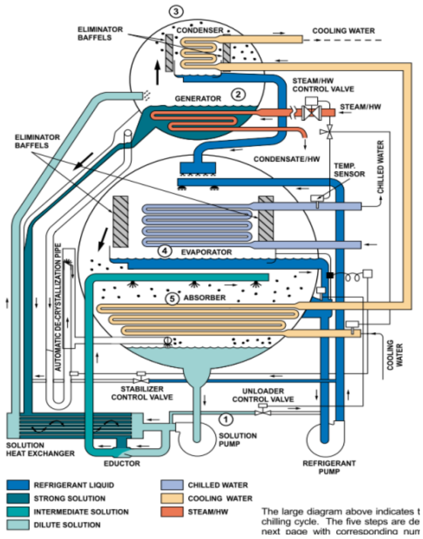


Figure 4.1 Chilling cycle of an absorption chiller (York)

See Figure 4.2 a.): A dilute lithium bromide solution is collected in the bottom of the absorber shell. From here, hermetic solution pump moves the solution through a shell and tube heat exchanger for preheating.

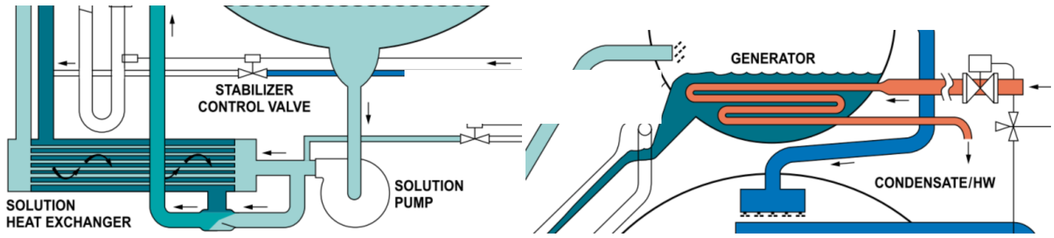


Figure 4.2 a.) Solution pump b.) Generator

See Figure 4.2 b.) After exiting the heat exchanger, the dilute solution moves into the upper shell. The solution surrounds a bundle of tubes which carries either steam or hot water. The steam or hot water transfers heat into the pool of dilute lithium bromide solution. The solution boils, sending refrigerant vapor upward into the condenser and leaving behind concentrated lithium bromide. The concentrated lithium bromide solution moves down to the heat exchanger, where it is cooled by the weak solution being pumped up to the generator.

See Figure 4.3 a.) The refrigerant vapor migrates through mist eliminators to the condenser tube bundle. The refrigerant vapor condenses on the tubes. The heat is removed by the cooling water which moves through the inside of the tubes. As the refrigerant condenses, it collects in a trough at the bottom of the condenser.

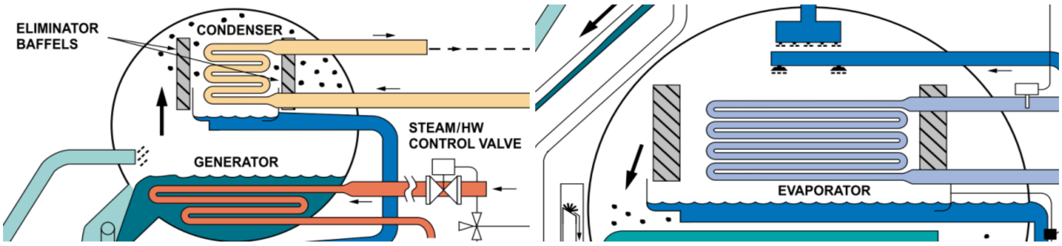


Figure 4.3 a.) Condenser b.) Evaporator

See Figure 4.3 b.) The refrigerant liquid moves from the condenser in the upper shell down to the evaporator in the lower shell and is sprayed over the evaporator tube bundle. Due to extreme vacuum in the lower shell (6mm Hg (0.8kPa) absolute pressure), the refrigerant liquid boils at 39°F (3.9°C), creating the refrigerant effect. (The vacuum is created by hygroscopic action - the strong affinity lithium bromide has for water - in the absorber directly below).

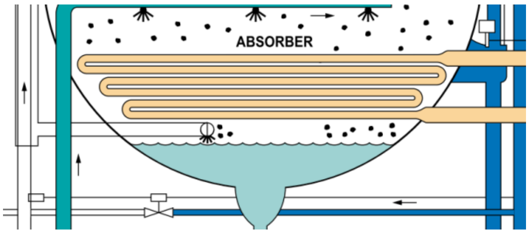


Figure 4.4 Absorber

See Figure 4.4 As the refrigerant vapor migrates to the absorber from the evaporator, the strong lithium bromide solution from the generator is sprayed over the top of the absorber tube bundle. The strong lithium bromide solution actually pulls the refrigerant vapor into the solution, creating the extreme vacuum in the evaporator. The absorption of the refrigerant vapor into the lithium bromide solution also generates heat, which is removed by the cooling water. The now dilute solution of lithium bromide collects in the bottom of the lower shell, where it flows down to the solution pump. The chilling cycle is now completed and the process begins once again.

4.2 Chiller operating performance

There are several metrics, when defining efficiency of absorption chiller e.g. coefficient of performance (COP), integrated part load value (IPLV), applied part load value (APLV).

4.2.1 Coefficient of performance (COP)

COP is defined as the refrigeration effect, divided by the net heat input (Equation 3). Single-effect absorption chillers have COPs of approximately 0.6-0.75 (see Table 4.1) out of an ideal 1.0. Since the COPs are less than one, the single-effect chillers are normally used in applications that recover waste heat such as waste steam from power plants or boilers. That is why we will focus on single-effect using water and lithium bromide as refrigerant and absorbent respectively. As it can be seen from the Table 4.1, this mixture is appropriate for chilling temperature between 6 and 20°C, which is our target temperature.

$$COP_{ABS} = \frac{Q_E}{Q_G + P_{work}} \tag{3}$$

Table 4.1 Comparison of different technologies (Harvey, 2006)

Technology	Closed-cycle absorption (single-effect)	Closed-cycle absorption	Closed-cycle adsorption	Open-cycle solid desiccant	Open-cycle liquid desiccant
Refrigerant	H_2O	NH_3	H_2O	-	-
Absorbent	LiBr	H_2O	Silica gel	Silica gel	CaCl, LiCl
Chilling carrier	Water	Water-glycol	Water	Air	Air
Chilling temperature	6 - 20°C	-60°C to 20°C	6-20°C	16 - 20°C	16 - 20°C
Driving temperature	80 - 110°C	100 - 140°C	55 - 100°C	55 - 100°C	55 - 100°C
Cooling water temperature	Up to 50°C	Up to 50°C	Up to 35°C	Not applicable	As low as possible
Cooling power range	35 - 7000 kW	10 - 10000 kW	50 - 430 kW	20 - 350 kW	
COP	0,6 - 0,75	0,6 - 0,7	0,3 - 0,7		

4.2.2 Part load operation

An absorption water-chiller, like other building heating/cooling equipment, is generally selected to meet the design (maximum) load. Under design conditions, the water-chiller operates continuously at full capacity. During much of the cooling season, however, the load is less than the design value and the water-chiller must be cycled on and off to provide the desired indoor conditions. The seasonal performance of the water-chiller is thus dependent on its performance during part-load operation. The primary absorption chiller performance standard is ARI Standard 560, which provides testing standard conditions, rating requirements, integrated part load value (IPLV) or non-standard part load value (NPLV) (Sakraida, 2009). Values gained from operations of buildings are incorporated into the IPLV equations (Equation 4).

$$IPLV = 0{,}01A + 0{,}42B + 0{,}45C + 0{,}12D \tag{4}$$

Where:

A= COP at 100% capacity (29,44°C)

B= COP at 75% capacity (26,03°C)

C= COP at 50% capacity (22,50°C)

D= COP at 25% capacity (19,03°C)

Non standard operating conditions of an absorption chiller can be evaluated through COP dependency on chiller loading. In the Figure 4.5 three different kinds of absorption chillers are presented and it can be seen that they are the most efficient at 50% part load. For single-effect chiller COP at this value is approximately 0,7. It must be noted that these values are only illustrative.

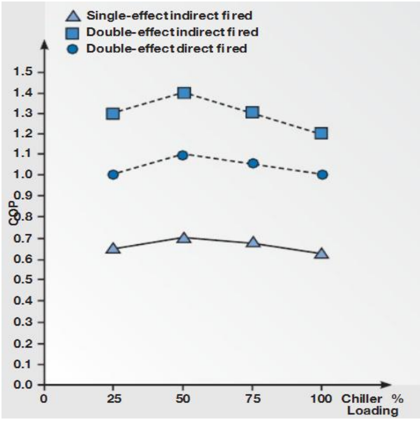


Figure 4.5 Effect of part-load operation on chiller efficiency (Sakraida, 2009)

4.2.3 Applied Part Load Value

The Applied Part Load Value, APLV is calculated using the same IPLV formula, except that actual chilled and condenser water temperatures and flow rates are used. The advantage of using the APLV over the IPLV is that this rating more closely approximates actual operating conditions imposed on the chiller. The disadvantage is the additional performance data that needs to be collected.

4.3 Lithium bromide water

COP of a single-effect absorption chiller can be theoretically expressed, instead of energy, with temperatures of evaporator, condenser and generator (Equation 5).

$$COP = \frac{T_E}{T_C - T_E} \cdot \frac{T_G - T_C}{T_G} \qquad (5)$$

Table 4.2: Water-LiBr absorption chiller thermal energy types and temperature ranges (Yin, 2006)

Heat-driven Absorption Chiller Type	Pressure (kPa)	Temperature (°C)
Direct-fired fossil fuel (natural gas, oil, LPG etc.)	-	1000-1800
Double-stage exhaust gas	-	400-600
Single-stage exhaust gas	-	230-350
Double-stage steam	400 – 1000	144 - 180
Single-stage steam	100 - 400	103 - 133
Double-stage hot water	350 – 1100	140 - 200
Single-stage hot water	40 - 200	75 - 120
Other fuel/steam/hot water/exhaust gas	Same as above	Same as above

4.4 Manufacturers

In the Table 4.3 are presented manufacturers of the absorption chillers, types of technology they use, which medium, what is cooling capacity. In the Figure 4.6 is presented absorption chiller from company York. In the Table 4.4 Prices for absorption chillers of different cooling capacitiesTable 4.4 are presented approximate prices for absorption chillers.

Table 4.3 Manufacturers of the absorption chillers

Manufacturer	Technology	Medium	Nominal cooling capacity (ton[1])
Broad	Direct fired	Natural gas	66 - 3307
	Two stage	Steam	66 - 3307
		Hot water	66 - 3307
	Single stage	Steam	66 - 1984
		Hot water	60 - 1754
Cention	Single stage	Steam	n.a.
		Hot water	30 - 525
Carrier	Single stage	Steam	100 - 700
		Hot water	75 - 525
	Two stage	Direct fired/steam	n.a.
McQuay	Two stage	Direct fired/steam	100 - 1500
Robur	Direct fired	Natural gas	n.a.
Trane	Single stage	Steam	112 - 465
		Hot water	500 - 1350
	Two stage	Steam	380 - 1650
Thermax	Single stage	Steam	100 - 1400
		Hot water	100 – 650
			100 – 1400
			100 – 1400
	Two stage	steam	50 - 1400
Yazaki	Single stage	Water	5 - 30
	Direct fired	Natural gas	n.a.

[1]Air conditioner equipment power in the U.S. is often described in terms of "tons of refrigeration". A "ton of refrigeration" is defined as the cooling power of one short ton (2000 pounds or 907 kilograms) of ice melting in a 24-hour period. This is equal to 12,000 BTU per hour, or 3517 watts (*Wikipedia*).

Figure 4.6 A 172-ton York/Johnson Controls absorption chiller uses exhaust heat from cogeneration to supply space cooling at One Stone Road in Guelph (source: powergenworldwide.com)

Table 4.4 Prices for absorption chillers of different cooling capacities

Cooling capacity [kW]	Price [€]
17,5	17000
35	22000
70	30000
105	36000

4.5 Operating strategies

The selection and operating scheme of a cogeneration system is very much site-specific and depends on several factors, as described below (The National Energy Conservation Centre).

4.5.1 Base electrical load matching

In this configuration, the cogeneration plant is sized to meet the minimum electricity demand of the site based on the historical demand curve. The rest of the needed power is purchased from the utility grid. The thermal energy requirement of the site could be met by the cogeneration system alone or by additional boilers. If the thermal energy generated with the base electrical load exceeds the plant's demand and if the situation permits, excess thermal energy can be exported to other customers.

4.5.2 Base thermal load matching

Here, the cogeneration system is sized to supply the minimum thermal energy requirement of the site. Stand-by boilers or burners are operated during periods when the demand for heat is higher. The prime mover installed operates at full load at all times. If the electricity demand of the site exceeds that which can be provided by the prime mover, then the remaining amount can be purchased from the grid. Likewise, if local laws permit, the excess electricity can be sold to the power utility.

4.5.3 Electrical load matching

In this operating scheme, the facility is totally independent of the power utility grid. All the power requirements of the site, including the reserves needed during scheduled and unscheduled maintenance, are to be taken into account while sizing the system. This is also referred to as a “stand-alone” system. If the thermal energy demand of the site is higher than that generated by the cogeneration system, auxiliary boilers are used. On the other hand, when the thermal energy demand is low, some thermal energy is wasted. If there is a possibility, excess thermal energy can be exported to other facilities.

4.5.4 Thermal load matching

The cogeneration system is designed to meet the thermal energy requirement of the site at any time. The prime movers are operated following the thermal demand. During the period when the electricity demand exceeds the generation capacity, the deficit can be compensated by power purchased from the grid. Similarly, if the local legislation permits, electricity produced in excess at any time may be sold to the utility.

5 TRNSYS

TRNSYS is a transient systems simulation program with a modular structure. It is used by engineers and researchers around the world to validate new energy concepts, from simple domestic hot water systems to the design and simulation of buildings and their equipment, including control strategies, occupant behavior, renewable energy systems (wind, solar, photovoltaic, hydrogen). The user specifies the components that constitute the system and the manner in which they are connected. The TRNSYS library includes many of the components commonly found in thermal and electrical energy systems, as well as component routines to handle input of weather data or other time-dependent forcing functions and output of simulation results (TRNSYS). If desired component does not exist in standard library, then the program allows adding a mathematical model, which describes specific component. This model can be programmed in languages such as C, C++, PASCAL, FORTRAN, etc. Main applications include: solar systems (solar thermal and photovoltaic systems), low energy buildings and HVAC systems, renewable energy systems, cogeneration, fuel cells.

5.1 Tess Libraries

Thermal Energy System Specialists (TESS) was founded in early 1994 by a group of engineers dedicated to providing engineering software and programming expertise to companies and individuals in the energy field. They have developed well over 200 TRNSYS components. In 2001, team under Trnsys selected the best of those components, cleaned up, standardized and commented the source code, created both online and printed manual documentation, and made them into a commercially available product. An expanded second version of the libraries was released along with TRNSYS 16 in 2003 and with TRNSYS 17 in 2010(TRNSYS).

5.2 STEC Libraries

STEC is a collection of TRNSYS models especially developed to simulate solar thermal power generation. It is a supplement to the standard TRNSYS routines featuring components from solar thermal power plants like concentrating collectors, steam cycles, gas turbines and high temperature thermal storage systems. It was developed as a SolarPACES activity and is steadily used, updated and completed by users within the SolarPACES group. The STEC simulation models are intensively used in feasibility studies for solar thermal power projects as well as in research programs for new solar thermal power technologies.

6 MODELING OF HEAT GENERATOR, COGENERATION, DHS AND BUILDINGS

Aim of this work is to make a general model which would include cogeneration and district heating system. It comprises model of heat generator, model of cogeneration, building's heat load (for heating, cooling and DHW) and losses from pipes and hot water tank. This model calculates heat demand, electricity generation and other relevant parameters that help us to evaluate different scenarios proposed by the user. In other words with chosen performance indicators one is able to decide if it is worth rejecting a certain amount of heat in cooling tower and at the same time generate more electricity. Following chapters describe how specific components are modeled.

6.1 Heat generator

For modeling of heat generator one needs its efficiency and a relationship between efficiency and part load operation. Efficiencies vary, for example for biomass boilers efficiencies are between 75% and 92%.

Efficiency changes with load on boiler, but this change is not significant so a constant efficiency was assumed.

6.2 Cogeneration unit

In order to be able to model a cogeneration unit following data need to be known: temperature range of heating medium, mass flow of heating medium, temperature range of cooling water, mass flow of cooling water, electrical power, changing of electrical output with design flow, part load operation. Example of needed data is presented in 7.7.1.

6.3 Heat load modeling

Buildings connected to the district heating system (DHS) via substation represent heat loads or demand of the system. Three components of heat load are assumed to be: domestic hot water (DHW), space heating and needed cold (e.g. heat for absorption cooling).

Following data needs to be known in order to describe a building model in TRNSYS: volume of building, buildings layers composition, number of inhabitants, orientation of walls, area of windows, indoor temperature, type of ventilation, ground temperature.

6.3.1 Building's heating

Buildings heat loss occurs due to several heat transfer mechanisms: conduction through buildings envelope (walls, roof, ground and windows), radiation and ventilation. In order to assure desired temperature a constant heat flow has to be supplied. In such a way heat loss will be compensated. To calculate heat that has to be supplied one should use SIST EN ISO 13790 standard which basically covers following equations:

$$Q_{NH} = (H \cdot DD \cdot 24 - \eta \cdot Q_{gain})/1000 \;\; [kWh] \tag{6}$$

Equations for terms that appear in this equation are given below.

$$H = H_T + H_{V,m} \left[\frac{W}{K}\right] \tag{7}$$

$$H_T = \sum_{i=1}^{n} U_{envelope,i} \cdot A_{envelope,i} + L_s + b \cdot (U_u \cdot A_u) \left[\frac{W}{K}\right] \tag{8}$$

$$H_{V,m} = 0{,}34 \cdot \left(n \cdot V_{building} + (1 - \eta_{rec}) \cdot \dot{V}\right) \left[\frac{W}{K}\right] \tag{9}$$

6.3.2 Building's cooling

In the summer time building's temperature arises due to heat gains such as solar radiation, internal gains (people, electrical appliances…). In order to assure desired temperature a constant heat flow has to be rejected. In such a way heat gains will be compensated. With the help of following equation one can calculate heat flow that has to be rejected. Basically it is the same as for heating only gains and losses are reversed.

$$Q_C = (Q_{gain} - \eta_{nc} \cdot Q_{loss})/1000 \;\; [kWh] \tag{10}$$

6.3.3 Domestic hot water

A certain amount of heat is taken from the district heating network in order to prepare DHW. A DHW consumption in buildings depends on the quantity and temperature of fresh water (12,4 °C (Environment Agency of the Republic of Slovenia, 2002)), amount of water used by persons, number of persons in the building and the temperature to which fresh water has to be heated up (The temperature level of water used is 55°C). Also important is the time when water is consumed and when the peak consumption occurs. To define time of consumption schedules are used. After we define required daily amount of DHW in dependence on number of persons we can assume that there are two peaks. First one occurs from 6 to 9 a.m. and the second one from 6 to 9 p.m. If a consumption profile for a concrete district heating network exists, then it can be entered into model and thus greater accuracy can be achieved.

From the Table 6.1 and Table 6.2 one can calculate a volume of hot water tank and DHW consumption.

Table 6.1: Heat required for DHW and sizes of hot water tanks in multifamily buildings (Medved & Arkar, 2007)

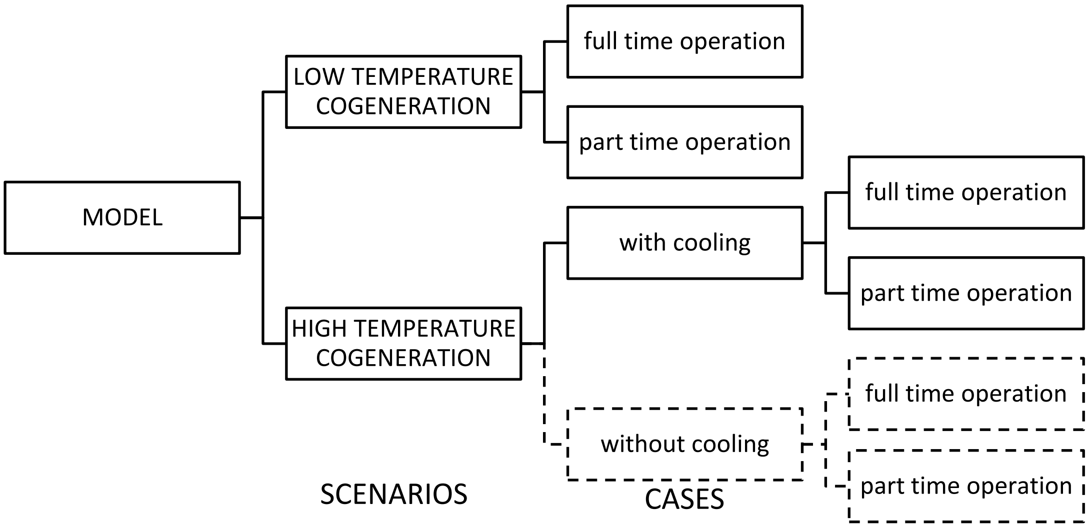
Number of apartments	Max. required heat [kW]	Hot water tank V_s [l] for Z_A [h]			
		0,5	**1**	**2**	**3**
1	8	90	150	200	220
2	12	130	200	300	370
4	18	190	300	450	520
6	24	230	400	600	740
8	28	300	470	690	890
10	33	330	540	835	960
12	39	395	640	985	1180
15	46	455	765	1130	1330
18	53	520	860	1130	1550
20	56	555	910	1380	1620
25	67	665	1110	1670	2000
30	76	750	1250	1870	2220
40	93	910	1525	2260	2730
50	112	1110	1850	2750	3320
60	130	1280	2140	3200	3840
80	162	1600	2660	3990	4800
100	195	1930	3200	4820	5760
120	230	2280	3815	5660	6790
150	275	2700	4550	6790	8120
200	350	3450	5780	8610	10330

Table 6.2 Quantity and energy needed for DHW preparation (Medved & Arkar, 2007)

	DHW [l/day per person]		**Effective heat**
	60°C	45°C	**[Wh/day per person]**
Low consumption	10...20	15...30	600...1200
Medium consumption	20...40	30...60	1200...2400
High consumption	40...80	60...120	2400...4800

6.4 How the model works

In order to observe what the impacts of cogeneration are on DHS and vice versa two configuration of the system were modeled. One assumes that biomass boiler produces hot water and therefore temperature of water is relatively low and the other one assumes biomass boiler which has higher temperatures. Both models have two different scenarios. One adjusts operating of cogeneration to the heat load and the other one assumes that cogeneration is working all the time even if heat demand is smaller than what cogeneration produces. In addition to that two cases were made without absorption cooling and with high temperature cogeneration. This enables to evaluate impacts of heat demand of absorption cooling on the system.



6.4.1 Low temperature cogeneration

As mentioned above this scenario assumes that hot water is used for generating electricity and heat needed for heating is obtained with cooling water. For cogeneration an ORC unit is used because it allows us to use heat also on lower temperature level compared to for example water. Heat generator (boiler) produces hot water where outlet temperature from boiler is fixed and it depends on the type of boiler. Mass flow of water varies with regard to heat demand which is calculated by the program in each hour. Part of hot water feeds cogeneration unit and part of it is directly used for heating of the domestic water (see Figure 6.1. With red color are marked properties that are fixed and constant, with green are marked properties that vary and with orange constraints of specific property. In the figure is also presented how properties depend on each other).

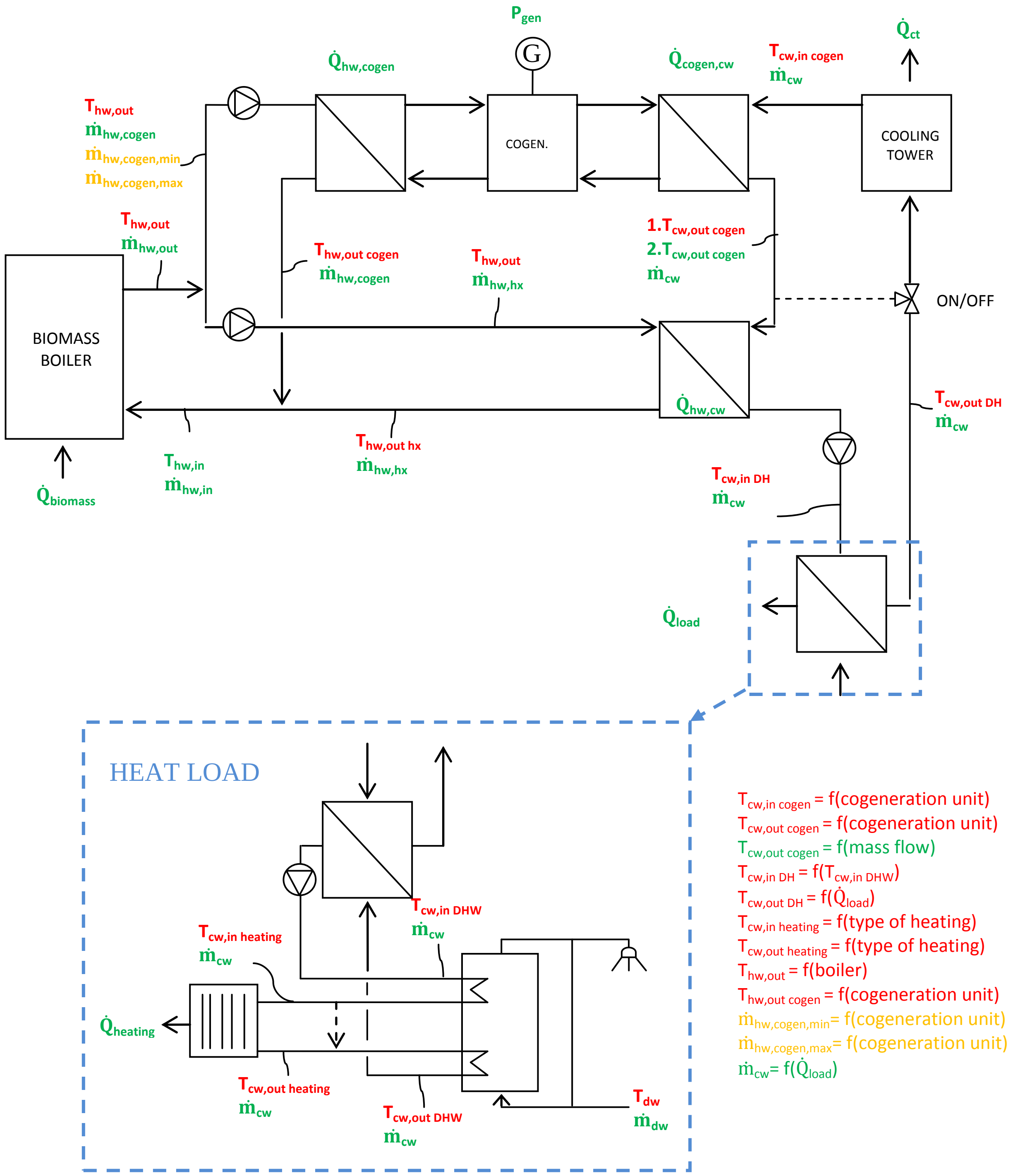


Figure 6.1 Scheme of low temperature cogeneration

Case 1: Part time operation of cogeneration at variable load

Principle of operation and calculation is following: if heat demand is high enough to use heat from cogeneration, then heat from the hot water is transferred to organic medium. Hot water then flows back into boiler and the temperature there is fixed because it is function of required temperature for cogeneration unit. Mass flow of hot water is the same before and after cogeneration. After expanding in turbine organic medium condenses and rejects heat to the cooling water. Temperature of condensation and therefore of cooling water is fixed with the type of ORC unit. Mass flow of cooling water depends on heat load (Equation 11). When latter is known and c_p assumed to be constant and both temperatures fixed, then mass flow can be calculated. So this means that operating of cogeneration and therefore generated electricity will be dictated by the mass flow.

$$\dot{Q}_{load} = \dot{m}_{cw} \cdot c_p \cdot \left(T_{cw,in\,DH} - T_{cw,out\,DH}\right) \tag{11}$$

But this is not valid for the whole range of mass flow. Cogeneration operates in a specific range of load which is characteristic of cogeneration unit and is fixed when unit is chosen. If the heat load is too small, then heat would be wasted and released into ambient. In this case cogeneration is shut down. Another extreme is when heat load exceeds heat that can be released from cogeneration. In this case mass flow increases above the maximal rated by cogeneration. This means that outlet temperature of cooling water from cogeneration decreases according to Equation 12. Heat flow is above certain value constant, so are specific heat and temperature of cooling water into cogeneration and if mass flow increases, the only way for equation to hold true is that and temperature of cooling water out of cogeneration decreases.

$$\dot{Q}_{cogen,cw} = \dot{m}_{cw} \cdot c_p \cdot \left(T_{cw,out\,cogen} - T_{cw,in\,cogen}\right) \tag{12}$$

Cooling water from cogeneration unit is then heated by a hot water from boiler up to a fixed temperature. This gives us heat that has to be supplied with hot water from boiler since mass flow of cooling water, its temperature difference and specific heat are known (Equation 12). Cooling water is then used for warming up DHW (which is later stored in a hot water tank) and after that for low temperature heating. This happens only during heating season, otherwise there is a bypass. Temperature of cooling water out of heating is fixed and constant. In the end water is cooled by the domestic water for which a fixed and constant value is assumed. Temperature of cooling water after heat exchanger with DHW varies, because it depends on consumption of DHW. Since mass flow of cooling water is much greater than mass flow of DHW, first one cools down slightly. When cogeneration is operating water enters into the cooling tower to reject so much heat that the temperature of cooling water out of cooling tower will have value which is prescribed with cogeneration unit. Again rejected heat can be calculated with Equation 12. If cogeneration is not operating then cooling water goes directly to the heat exchanger with hot water from boiler.

In the

Figure 6.2 is presented flow diagram for low temperature cogeneration, which does not operate all the time. It sums up how model is working and what the sequence of calculations is.

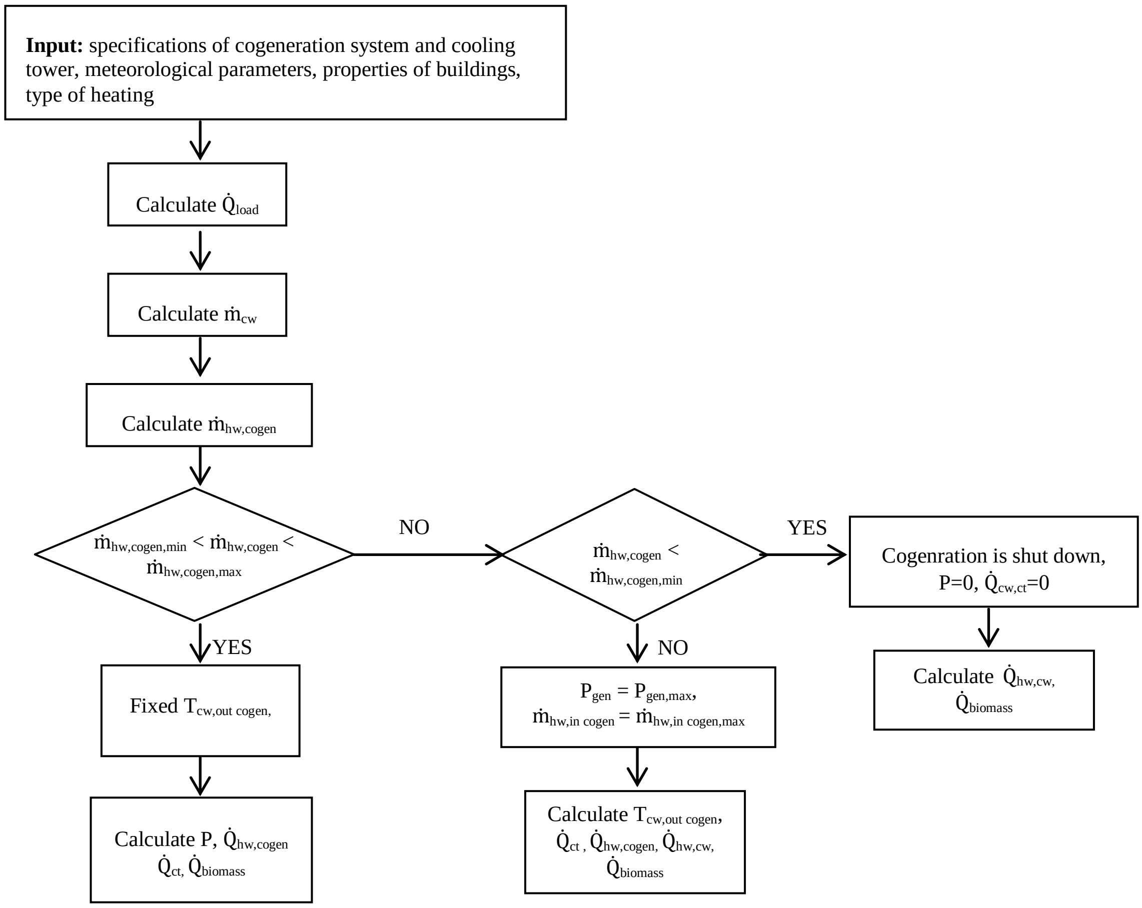


Figure 6.2 Flow diagram for low temperature cogeneration, part time operation

Case 2: Full time operation of cogeneration at variable load

In this case cogeneration is operating all the time. Principle of operation and calculation is almost the same as in previous case, the difference is when the heat load smaller is than heat flow from cogeneration. When this happens cogeneration is operating at minimum load (set with type of cogeneration unit), therefore there is a surplus of heat, which has to be rejected in cooling tower.

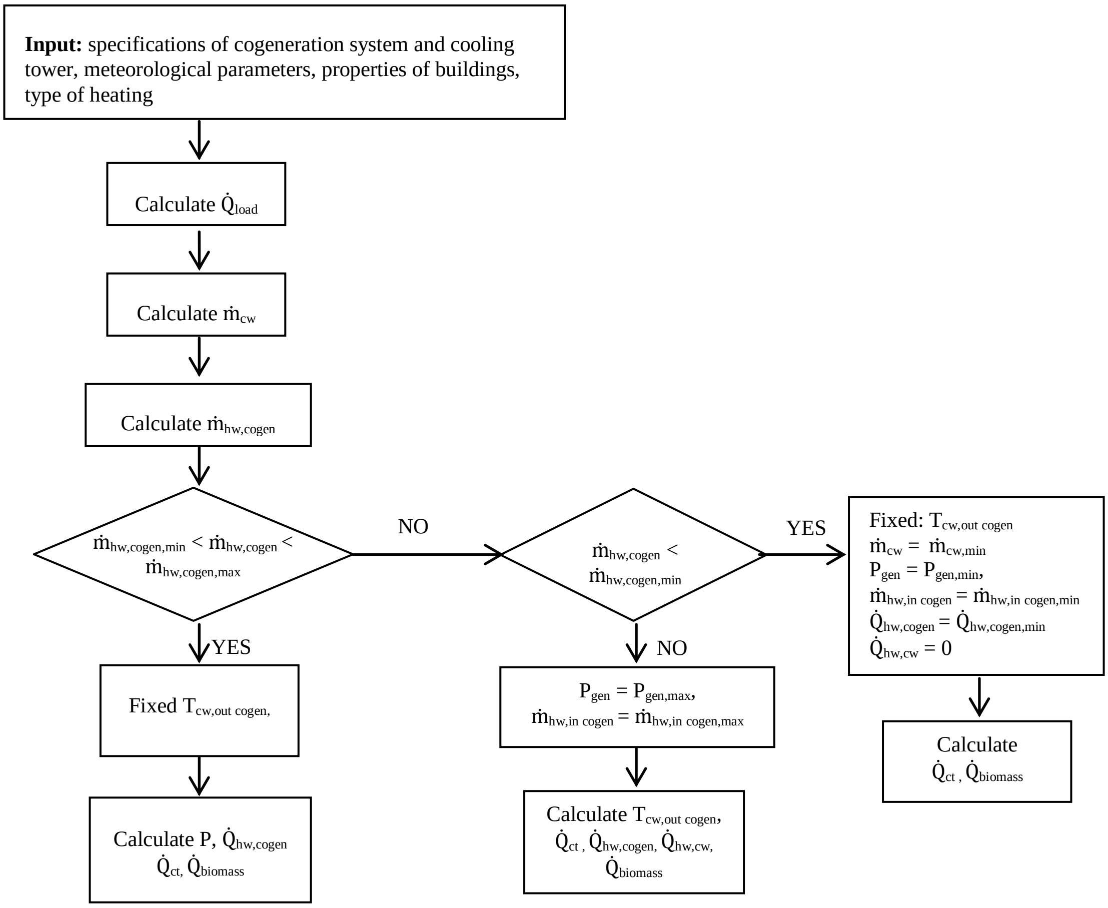


Figure 6.3 Flow diagram for low temperature cogeneration, full time operation

6.4.2 High temperature boiler

This scenario assumes biomass boiler with higher outlet temperatures, which has an influence on temperature of cooling water in a way that it can be higher. This allows use of absorption cooling where heat on high enough temperature level is needed. Besides that we do not need to use low temperature heating. Principle of operation is similar as before, deviations are presented below.

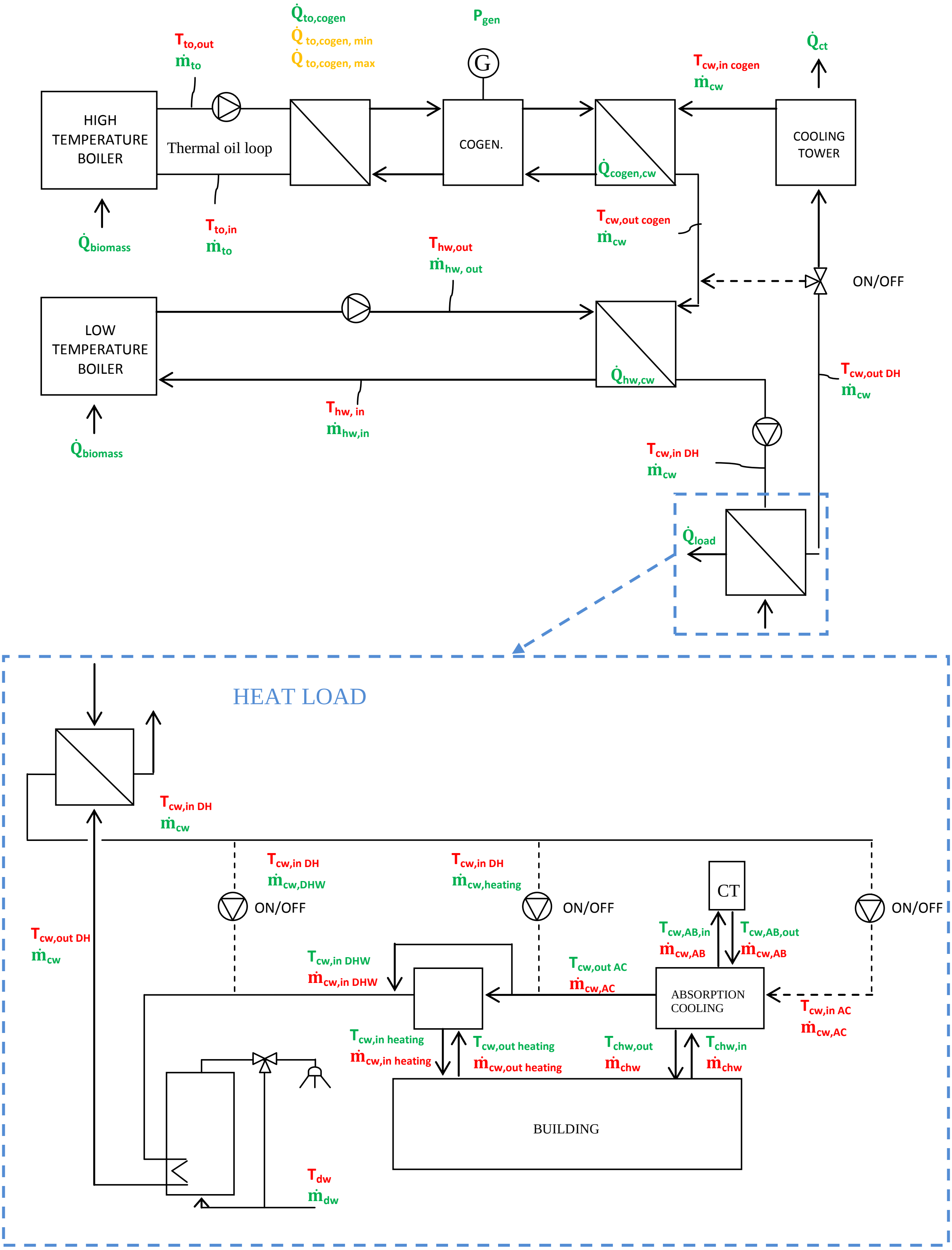


Figure 6.4 Scheme of high temperature cogeneration

Case 1: Part time operation of cogeneration at variable load

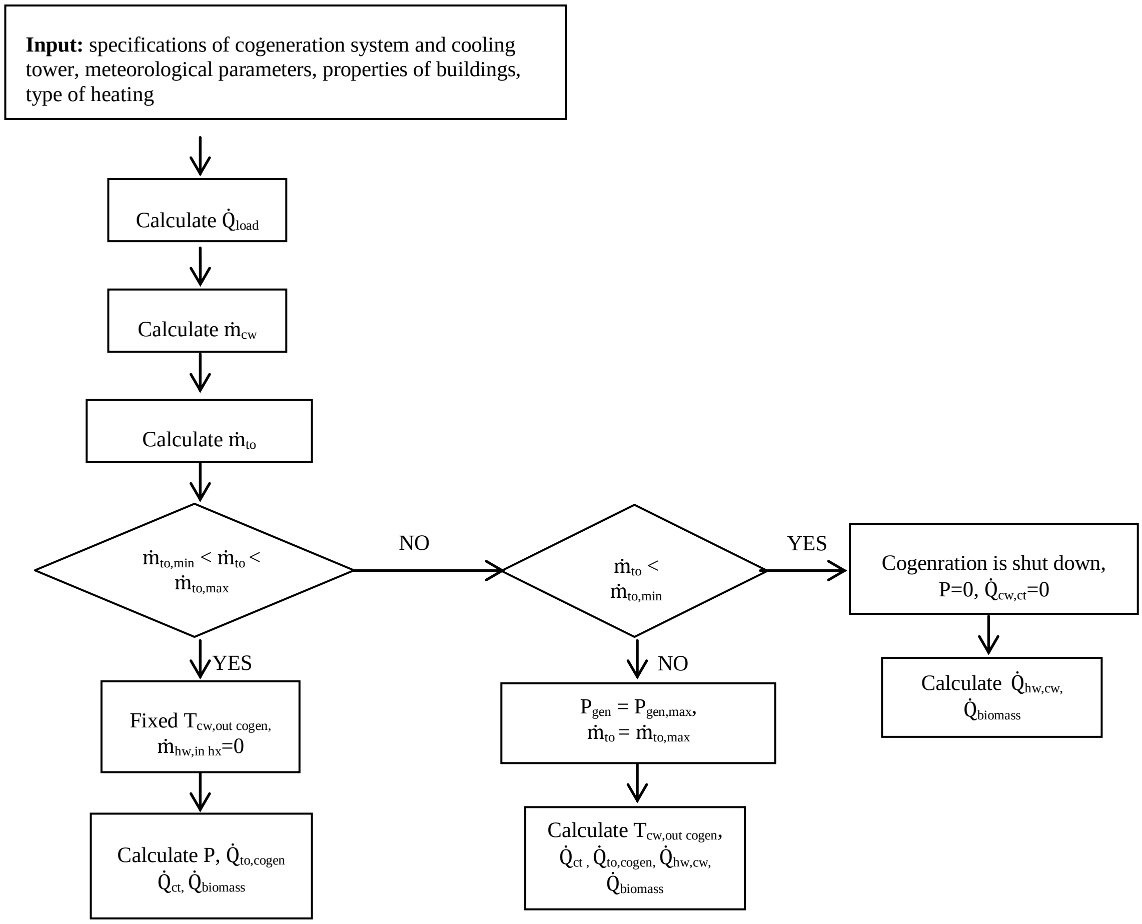


Figure 6.5 Flow diagram for high temperature cogeneration, part time operation

First heat demand has to be known therefore one can calculate mass flow of cooling water. After that generated electricity and mass flow of heating medium (for example thermal oil) can be calculated since inlet and outlet temperatures of heating medium into cogeneration are fixed with the unit. Temperature of cooling water into cogeneration is also fixed with the chosen unit.

If needed mass flow of cooling water is below minimum mass flow acceptable for cogeneration, then cogeneration is shut down and low temperature boiler supplies needed heat. If needed mass flow of cooling water is in the range of mass flow acceptable for cogeneration, then cogeneration is operating and temperature of cooling water out of cogeneration is fixed. Third option is when needed mass flow of cooling water exceeds maximum mass flow acceptable for cogeneration and in that case temperature of cooling water out of cogeneration drops and low temperature boiler supplies the rest of needed heat.

After cogeneration, cooling water transfers heat to the heat load. In the Figure 6.4 heat load of all substations is presented as one substation which has absorption cooling unit, space heating and DHW. So, cooling water first enters into absorption cooling unit (when cold is needed) because the highest temperature is needed there. Inlet temperature is fixed and besides that also mass flows of cooling water for absorption unit and chilled water. Since temperature regulation of energy flows is used, temperatures vary. After absorption unit cooling water supplies heat for space heating and DHW where again mass flow is fixed and temperatures vary. In the end it mixes with the water from other substations and goes back to cogeneration unit.

Case 2: Full time operation of cogeneration at variable load

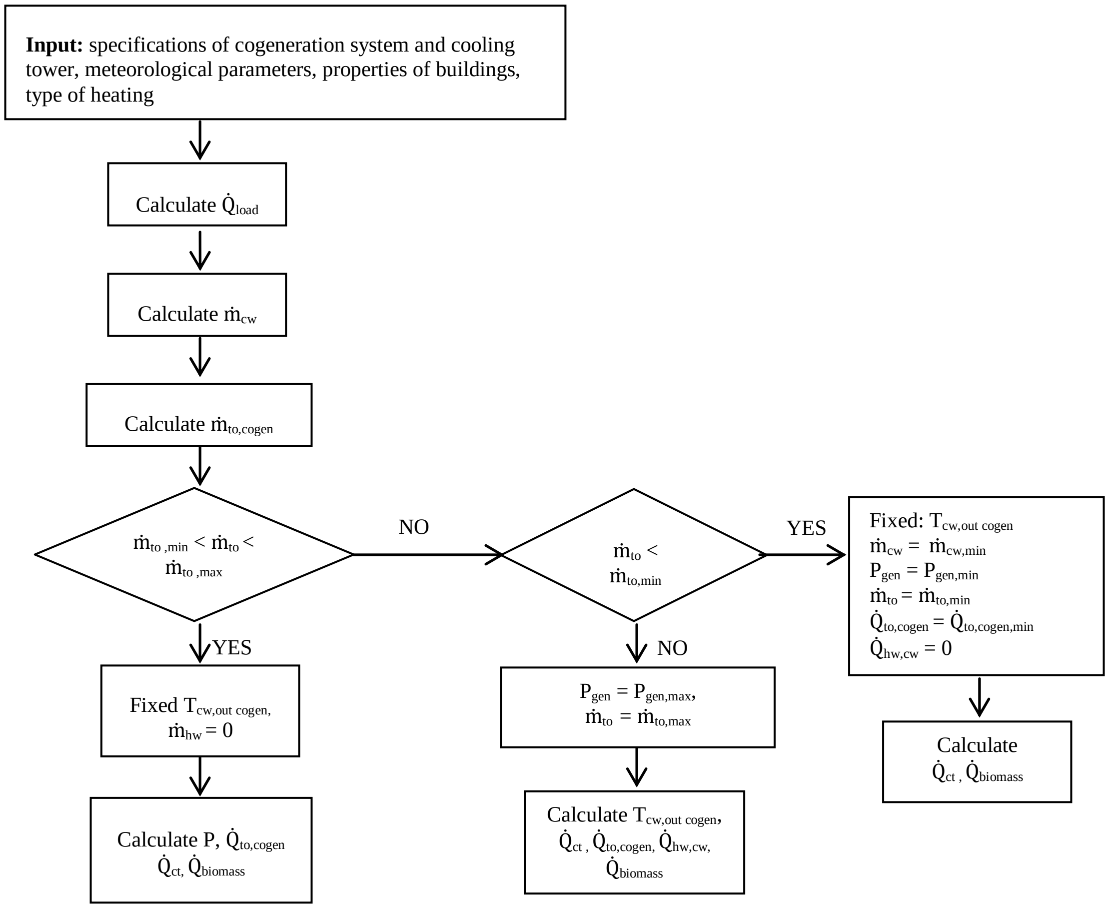


Figure 6.6 Flow diagram for high temperature cogeneration, full time operation

In this case cogeneration is operating all the time. Principle of operation and calculation is almost the same as in previous case the difference is when the heat load smaller than heat flow from cogeneration is. When this happens cogeneration is operating at minimum load (set with type of cogeneration unit), therefore there is a surplus of heat, which has to be rejected in cooling tower.

6.4.3 Summary of working of the model

Following table sums up what was presented in this chapter. User has to provide specific inputs, then program calculates different parameters such as: mass flows, temperatures etc. But they are not indicators that would allow proper evaluation of the operation therefore program calculates also generated electical energy, heat delivered with biomass etc.

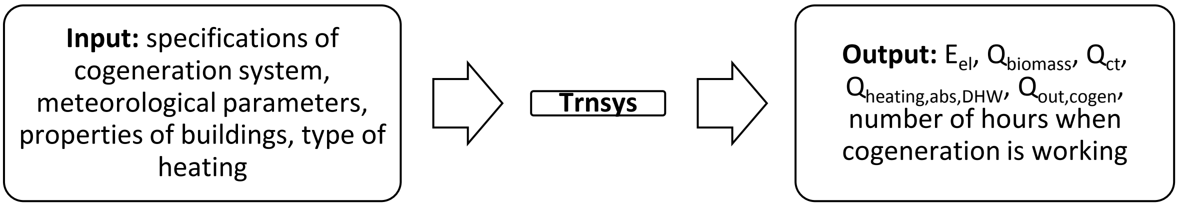


Figure 6.7 Graphical presentation of working of the model

6.5 Limitations of the Trnsys DHS model

This is energy model and it does not take into account pressure conditions in the pipelines. Only primary network is modeled. It is assumed, that all buildings have the same properties and each of them is modeled as single thermal zone, which means that there are no differences in air temperatures between e.g. kitchen, bedroom and hallway. Model allows user to define building's properties in detail in order to obtain accurate results. It means we can change any of the construction properties and monitor the impact of this step on peak thermal power, use of energy. Since the model of buildings is coupled with the district heating network we can evaluate the effects such as improved thermal envelope of the buildings on thermal demand.

One of the limitations of the model is that the supplied energy for DHW changes only during day, but it does not change according to season demand.

Model does not take into account changes of thermodynamic properties such as: specific heat and density of water (temperature dependence).

6.6 Performance indicators

In order to evaluate different options of system's configuration and operation some performance indicators have to be chosen. They are:

- Ratio of generated electricity and used biomass in terms of heat: $E_{el}/Q_{biomass}$
- Ratio of heat used for heating/absorption cooling/DHW and heat from cogeneration: $Q_{heating,abs,DHW}/Q_{out\ cogen}$
- Ratio of heat used for heating/absorption cooling/DHW and heat from biomass: $Q_{heating,abs,DHW}/Q_{biomass}$
- Ratio of heat rejected in cooling tower and heat from biomass: $Q_{ct}/Q_{biomass}$
- Ratio of sum of heat used for heating/absorption cooling/DHW and generated electricity and heat from biomass: $(Q_{heating,abs,DHW} + E_{el})/Q_{biomass}$

6.7 User friendly report

In order to acquire results in a user friendly format, graphs representing the most important variables were made. User can copy results given by Trnsys into Excel spreadsheet and as a result he will get report showing graphs. In the first column are graphs with hourly values, in the second one are graphs with monthly values and in the third one with yearly values. Added is also histogram of needed heat flow for DHW, space heating (and absorption cooling) in a year. How the graphs look like can be seen in Results.

7 CASE STUDY – LOCAL COMMUNITIY ZAGORJE OB SAVI

Town Zagorje ob Savi, where studied DHS system is located lies in Zasavje region in central Slovenia. It is a former brown coal mining town, but since year 2000 the mine has been closed (STAT, 2002). The problem of polluted environment remains today since Zasavje region is the most polluted region in Slovenia. Emissions of SO_2 that come from burning low quality fossil fuels have been dropping for last decade, but prognosis for emissions of heavy metals and complex organic compound are rather pessimistic. One of the reasons for this is waste incineration in cement plant in neighboring community.

7.1 Weather conditions

Average yearly temperature for the area of town Zagorje is around 10°C. Average degree days for period 2001 – 2007 are 3083 degree days (18°C base line temperature). In simulations 3300 degree days are used, same as for Ljubljana (ARSO, 2009). Test reference year for Ljubljana was used in simulations.

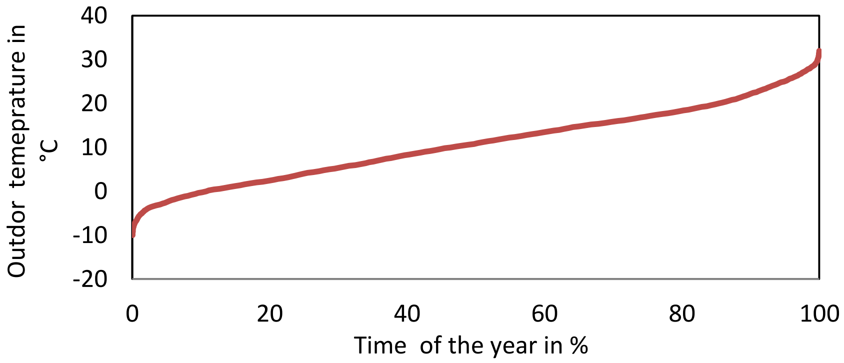


Figure 7.1: Outdoor temperature duration curve for test reference year for Ljubljana

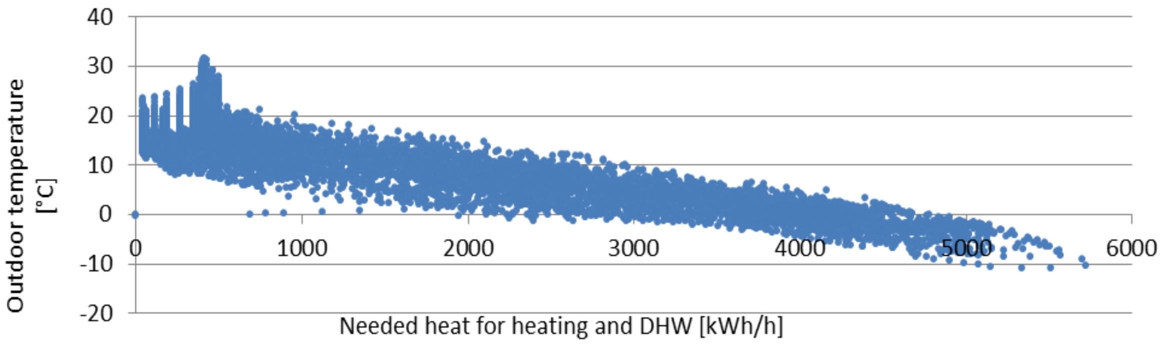


Figure 7.2 Needed heat for heating and DHW in dependance on outdoor temperature

7.2 Availability of biomass for heat and electricity generation

Municipality Zagorje ob Savi has an area of 147 km^2 and forests cower 65% of this area (9564 ha). According to national Forestry Institute the largest sustainable possible cut is 28794 m^3/year, where around 56 % is yearly realized.

7.3 Current utilization of biomass

The share of household heated with wood biomass in municipality is 31% which represents around 2000 apartments. Large consumer is also biomass district heating in town Zagorje ob Savi, which is subject of this thesis, where around 4000 tons of biomass is yearly used for heat production. Heat is mostly used in multifamily buildings connected to DHS network.

7.4 Description of district heating system in Zagorje

Wood chips district heating system in Zagorje ob Savi supplies heat to residential, public and industrial consumers in the town. The production facility consists of two wood chip biomass combustion plants, a hot water boiler with a nominal capacity of 2.5 MW_{th} each. As a back-up oil boiler is used with nominal capacity **7** MW. Yearly heat production (from 1995 to 2006) is around 14 GWh. Fuel is supplied by local wood industry SVEA. It consists mainly of saw dust and chips and has relatively high water content. Current yearly consumption of biomass is around 7000 m^3. Total heated area is around 60.000 m^2. Design temperature of the system is -18°C. This temperature is rarely reached and it means that system is over-dimensioned. In reality the nominal power of consumers is less than 12 MW. For detailed description of the system look (Vetršek, 2010).

7.5 Volume of hot water tank

District heating network consists of 19 heat substations, which are presented in the Table 7.2. Number of apartments for each of them is determined and therefore we can calculate volume of hot water tank for each substation (see also Table 6.1). Since our model does not differentiate between them, then the volume of hot water tank can be defined as a sum of volumes for each substation divided by 19. This gives us average volume of hot water tank.

$$Average\ volume = 53m^3/19 = 2800l$$

Next step is to model hot water tank in Trnsys. We need properties such as thickness and thermal conductivity of insulation, dimensions of it and they are given in a Table 7.1.

Table 7.1 Properties of hot water tanks (Haase GFK-Technik)

Volume [l]	Diameter [cm]	Height [cm]	Thickness of insulation – cover [cm]	Thickness of insulation – walls [cm]	Thermal conductivity [W/mK]
1000	79	225	14	12,5	0,037
2200	130	206	14	12,5	0,037

Table 7.2 Consumers in DHS Zagorje

Heat substation	Name	Max required heat power for heating [kW]	Number of occupants	Number of apartments	Max required heat power of DHW [kW]	Area [m^2]
0	SVEA	1000	0	/	/	/
1	KCDD - Cesta 9. avgusta 1	2126	500	200	/	12577
2	Cesta zmage 14	963	261	123	/	5948
3	Cesta zmage 16 (STV)	629	199	78	158	4812
4	Cesta zmage 22 (STV)	827	246	92	180	5232
5	Polje 10 (STV)	278	99	36	83	1944
6	Polje 19	1031	316	139	/	6176
7	Polje 26/1 - Polje 26	777	256	116	/	6217
8	Cesta 20. julija 2	631	150	28	/	4415
9	SPO,Cesta 9. avgusta 8	1394	300	90	/	6866
10	Vrtec MAJA	183	20	/	/	689
11	Salon SVEA	103	5	/	/	1260
12	Mercator - ŽIVA	205	10	/	/	1640
13	Cesta zmage 16b	84	50	5	/	1500
14	Dežman, Drnovšek	50	8	2	/	360
15	Polje 26/2	544	143	68	/	3207
16	Cesta zmage 7	689	45	24	/	4812
17	Ulica talcev 1	300	92	45	/	2198
18	SVEA	80	0	/	/	2000
TOGETHER		**11894**	**2700**	**1046**	**421**	**71853**

7.6 Hot water consumption

Number of persons is 2700 as from the Table 7.2 and average consumption per day per person was chosen as medium consumption and water with 45°C. Therefore from the Table 6.2 average consumption is 60 l/day/person. Having this number one can calculate average consumption per heat substation:

$$2700 \cdot 60/19 = 8526l = 8{,}5m^3 \quad (11)$$

7.7 System in Trnsys

This chapter describes model in Trnsys which was made for situation in Zagorje. As mentioned before biomass boiler already exists and can supply hot water with 110°C. In the following subchapters are presented two scenarios.

7.7.1 Cogeneration with existing boilers

This scenario assumes that into the existing system an ORC unit is installed. A 280 kW unit PureCycle was chosen and some characteristics of it are presented in the Table 7.3.

Table 7.3 Characteristics of the ORC unit

	116°C Resource		93°C Resource	
Hot Resource				
Inlet temperature [°C]	116	116	93	93
Exit temperature [°C]	78	90	78	78
Flow rate [l/s]	21	31	66	66
Cooling water				
Inlet temperature [°C]	16	27	16	27
Exit temperature [°C]	26	37	26	37
Flow rate [l/s]	58	62	70	77
Power output [kW]	280	280	280	205

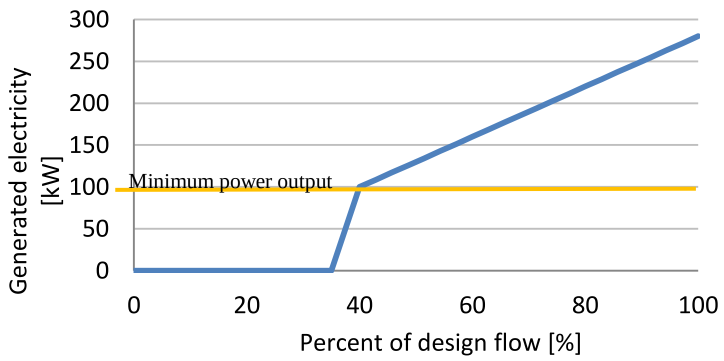


Figure 7.3 PureCycle Model 280 power output as a function of design flow rate

When modeling also changing of generated electricity as a function of percent of design flow has to be known. This relationship is presented in the *Figure 7.3*. Moreover next two relationships have to be known: changing of mass flow of hot water into cogeneration with mass flow of cooling water and changing of generated electricity with mass flow of cooling water. They can be represented as in *Figure 7.4* or as following conditions:

Table 7.4 Relations between characteristics in low temperature cogeneration

Mass flow of cooling water m_{cw} [t/h]	**Mass flow of hot water m_{hw} [t/h]**	**Generated electricity P [kW]**	**Temepature of cooling water out of cogeneration $T_{cw, out cogen}$ [°C]**
$93600 < m_{cw}$	$m_{hw} = 0$	P = 0	$T_{cw, out cogen} = T_{cw,out DHS}$
$93600 < m_{cw} < 234000$	$m_{hw} = 0{,}54 * m_{cw}$	$P = 1{,}28 * m_{hw} - 20$	$T_{cw, out cogen} = 37$
$m_{cw} > 234000$	$m_{hw} = 126000$	P = 280	$T_{cw, out cogen} = 1{,}0*10^{-4} * m_{cw}^2 - 9{,}35*10^{-2} + 52{,}7$

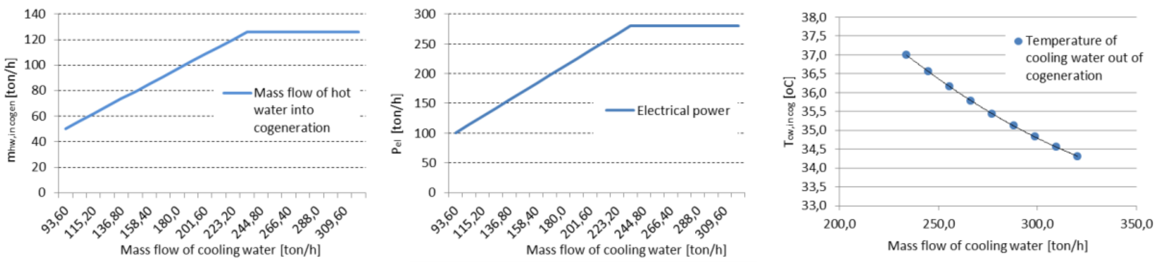


Figure 7.4 Relations between characteristics for low temperature cogeneration

As the temperature of the water from the boiler is lower than that of typical ORC cogeneration applications the system is slightly modified. Biomass boiler produces hot water (110°C) which feeds cogeneration unit and some of it is directly used for heating the domestic water (see Figure 7.5).

Principle of working was already explained in the Chapter 6.4.1, now specific data have to be presented that were applied to the model. Temperature that has to be provided for DHW in order to eliminate a chance for legionella is 55°C. In order to reach this temperature cooling water has to be heated up to 60°C. For calculating mass flow of it, constant specific heat of water (4,2 kJ/kgK) and temperature difference between 55°C and 32°C were assumed (Equation 12). 55°C is inlet temperature in heating and DHW system and 32°C is outlet temperature.

$$\dot{m} = \frac{\dot{Q}}{c_p \cdot \Delta T} = \frac{\dot{Q}}{4{,}2kJ/kgK \cdot 23K} \tag{13}$$

After heat exchanger with DHW cooling water enters into floor heating (during heating season, otherwise there is a bypass). Inlet temperature into floor heating depends on the way of use (shops, home, office...) but in our case manufacturers advise that water reaches 45°C, but it should not exceed 55°C. To ensure comfort maximum temperature difference between inlet and outlet temperature of water could be maximal 10 - 15 K (Floor heating). Outlet temperature from floor heating is 32°C. In the end water is cooled by domestic water for which it is assumed temperature of 12°C and in the case when cogeneration is operating also cooled by the cooling tower. Inlet temperature of cooling water into cogeneration is 27°C.

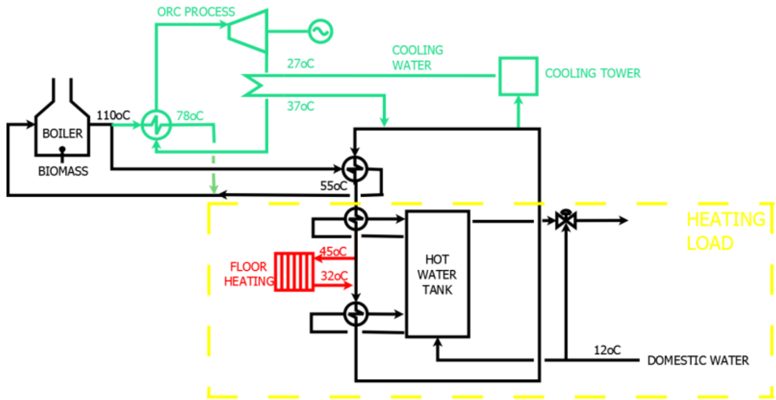


Figure 7.5 Proposed system with existing biomass boiler

7.7.2 Cogeneration with thermal oil boiler

This scenario assumes that one boiler is replaced with new thermal oil boiler allowing bigger cogeneration unit from manufacturer Turboden. Some characteristics of three units are presented in the *Table 7.5*.

Table 7.5 Characteristics of Turboden CHP units (Turboden)

	TURBODEN 4	TURBODEN 6	TURBODEN 10
INPUT – Thermal oil			
Inlet temperature [°C]	300	300	300
Outlet temperature [°C]	240	240	240
Thermal power input [kW]	2300	3240	3815
OUTPUT – Hot water			
Hot water temperature (in/out) [°C]	60/80	60/80	60/80
Thermal power to the cooling water	1854	2565	3038
PERFORMANCES			
Gross active electric power [kW]	427	641	737
Net electric efficiency	0,177	0,189	0,184
Biomass consumption [kg/h]	1106	1558	1834
LHW=2,6 kWh/kg,			
Boiler efficiency=0,8			

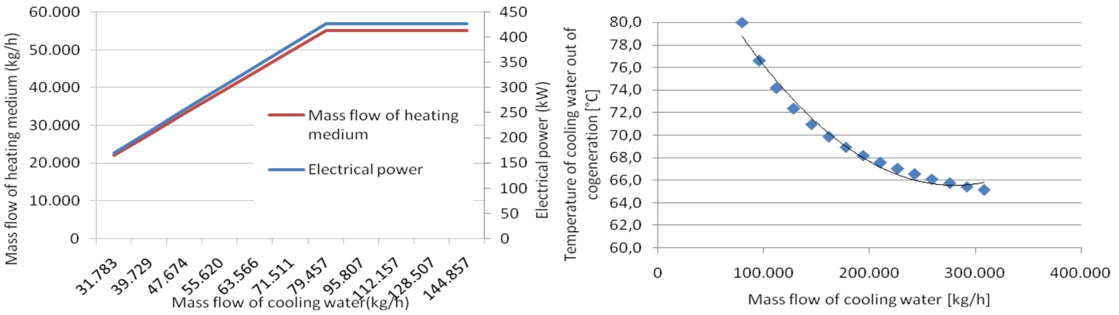


Figure 7.6 Relations between characteristics for high temperature cogeneration

For a first unit (Turboden 4) next relationships have to be known: changing of mass flow of thermal oil with mass flow of cooling water, changing of generated electricity with mass flow of cooling water and temperature of cooling water out of cogeneration. They can be represented as in Figure 7.6 or as following conditions:

Table 7.6 Relations between characteristics in high temperature cogeneration

Mass flow of cooling water m_{cw} [kg/h]	**Mass flow of hot water m_{hw} [kg/h]**	**Generated electricity P [kW]**	**Temepature of cooling water out of cogeneration $T_{cw,\ out\ cogen}$ [°C]**
$m_{cw} < 31783$	$m_{hw} = 0$	$P = 0$	$T_{cw,\ out\ cogen} = T_{cw,out\ DHS}$
$31783 < m_{cw} < 79457$	$m_{hw} = 0,69 * m_{cw}$	$P = 5,37*10^{-3} * m_{cw}$	$T_{cw,\ out\ cogen} = 80$
$m_{cw} > 79457$	$m_{hw} = 55200$	$P = 427$	$T_{cw,\ out\ cogen} = 3,27*10^{-10} * m_{cw}^2 - 1,83*10^{-4} * m_{cw} + 91,24$

In this case one existing boiler is replaced with thermal oil boiler, which provides higher temperature (inlet into cogeneration is at 300°C and return at 240°C). Higher temperatures have also inlet and outlet temperature of cooling water (60°C and 80°C respectively). Since latter one has 80°C, it can be used for absorption cooling, which operates when indoor temperature exceeds 25°C. Besides hot source we also need cooling water which can have 31°C at the inlet into absorption cooling. Outlet of chilled water has 7°C and inlet changes according to heat load but it does not exceed 12°C. Due to high temperature of cooling water a conventional system for space heating can be used, where temperatures are 65/55°C. Again inlet temperature for DHW has to be above 55°C which is fulfilled, since constraint is 60°C at the inlet into cogeneration.

When heat demand is too low for cogeneration to operate, then biomass boiler is used to provide heat for DHW, heating/cooling.

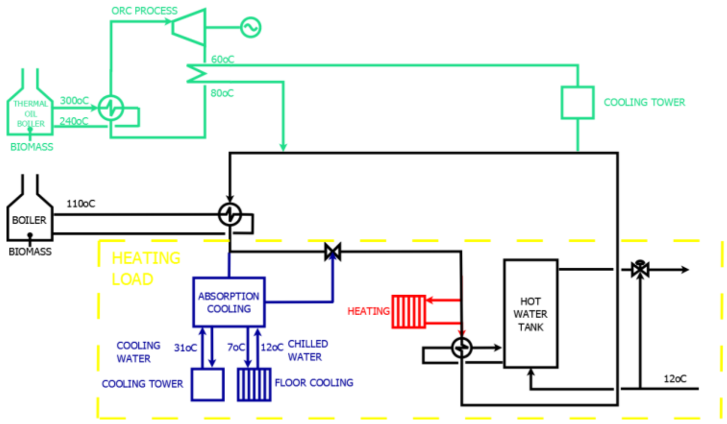


Figure 7.7 Proposed system with high temperature cogeneration

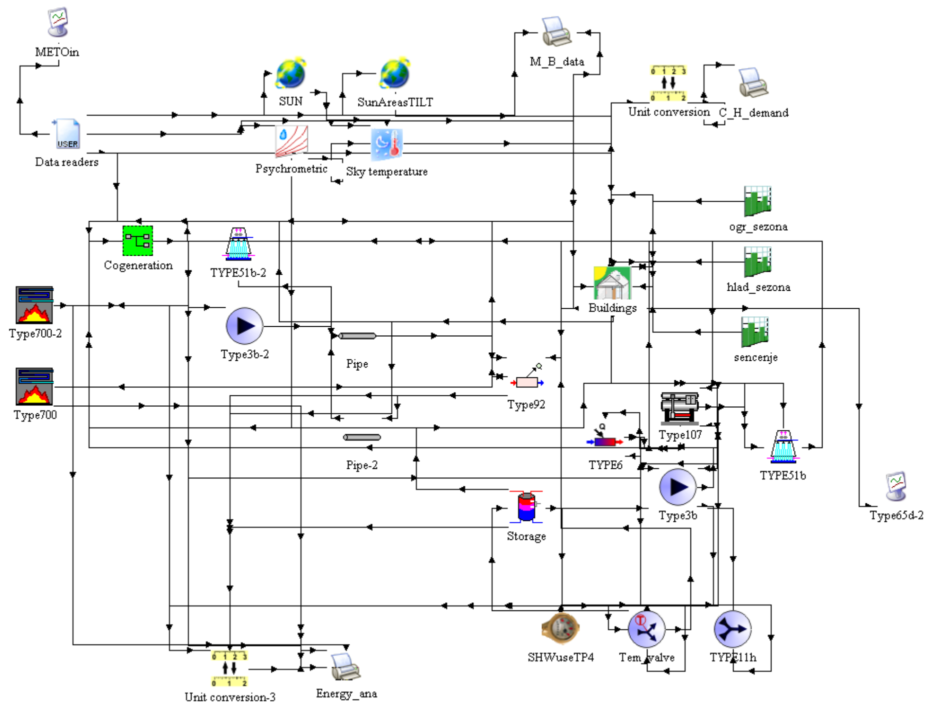


Figure 7.8 System in Trnsys

8 RESULTS

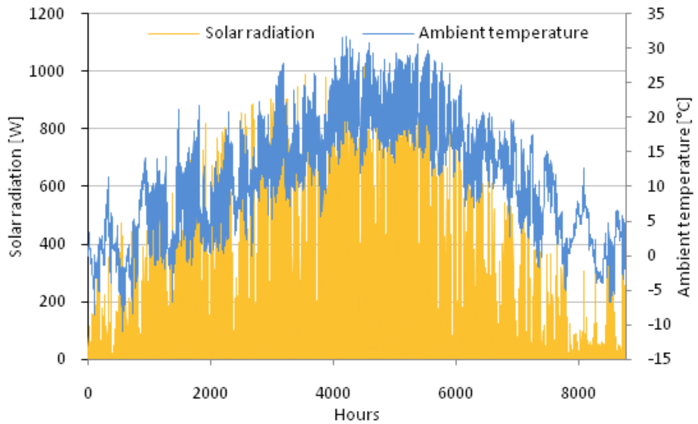
On the following pages are presented results of simulations.

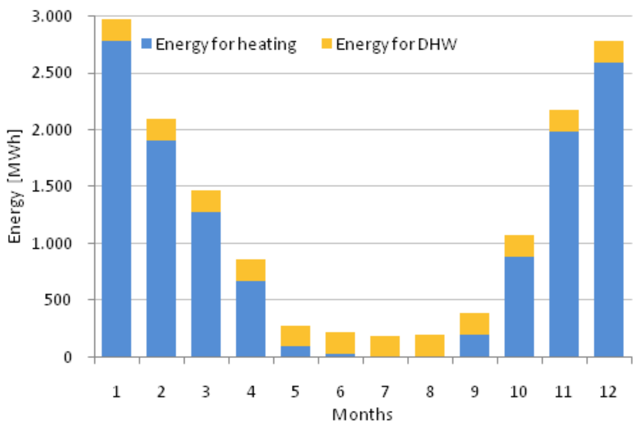
SCENARIO WITH LOW TEMPERATURE COGENERATION

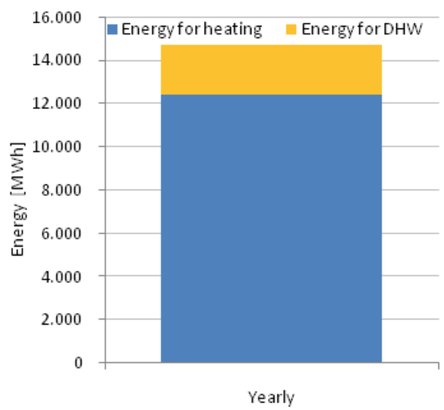
BIOMASS BOILER	
Thermal power	2 X 2,5 MW
Outlet temperature	110°C
BUILDINGS	
Peak load	5524 kW
Floor area	60000 m^2
Indoor temperature - summer	25°C
Indoor temperature - winter	20°C
HOT WATER TANK	
Required inlet temperature	55°C

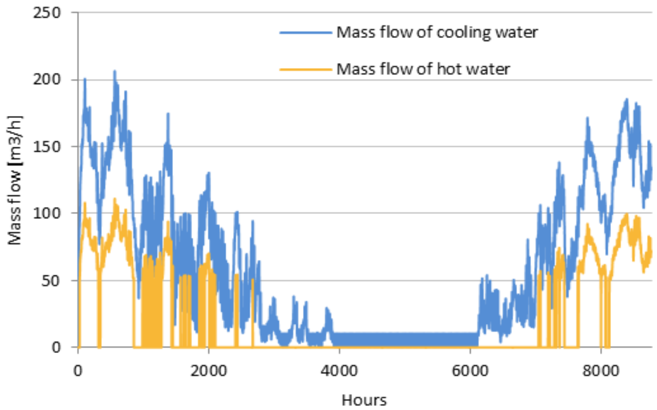
COGENERATION UNIT: ORC – PURE CYCLE	
Max. thermal power	
Range of generated electrical power	100 kW - 280 kW
Part load operation	40 – 100%
Inlet temperature of hot water	110°C
Outlet temperature of hot water	78°C
Inlet temperature of cooling water	27°C
Outlet temperature of cooling water	Varies, max.: 37°C
FLOOR HEATING	
Required inlet temperature	45°C
Maximum temperature difference between inlet and outlet	10 K
DOMESTIC WATER INLET	12°C

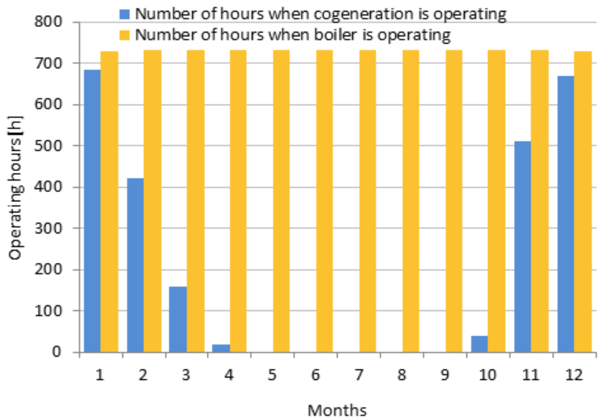
PART TIME OPERATION

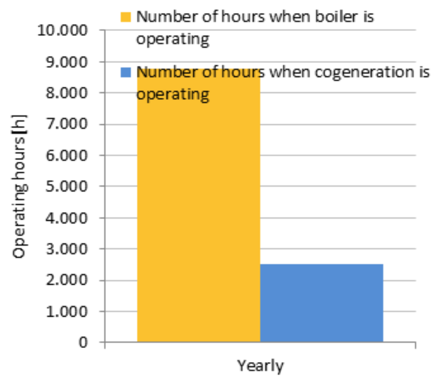


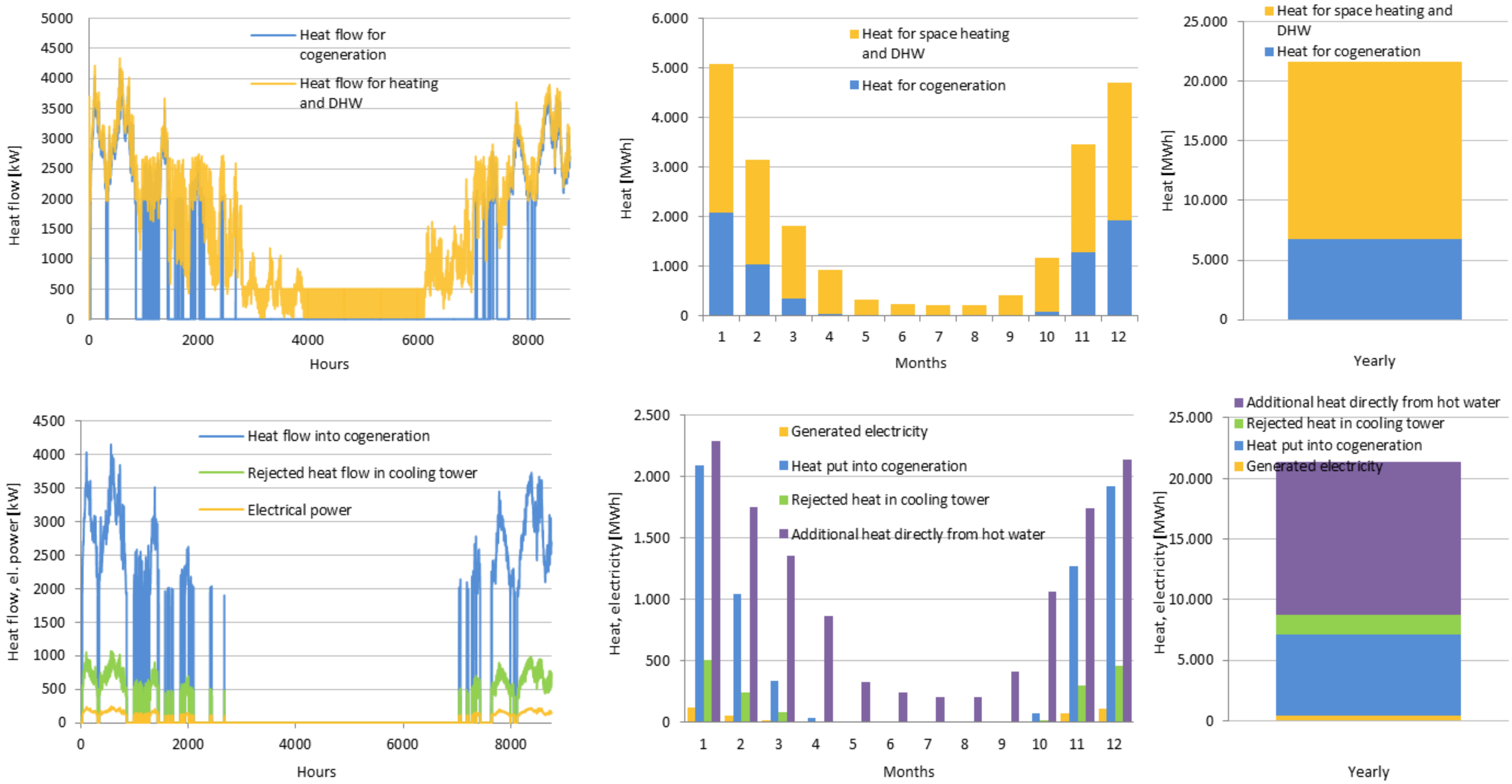











Heat flow for cogeneration
Heat flow for heating and DHW
Heat flow [kW]
Hours
Heat for space heating and DHW
Heat for cogeneration
Heat [MWh]
Months
Yearly
Heat flow into cogeneration
Rejected heat flow in cooling tower
Electrical power
Heat flow, el. power [kW]
Generated electricity
Heat put into cogeneration
Rejected heat in cooling tower
Additional heat directly from hot water
Heat, electricity [MWh]

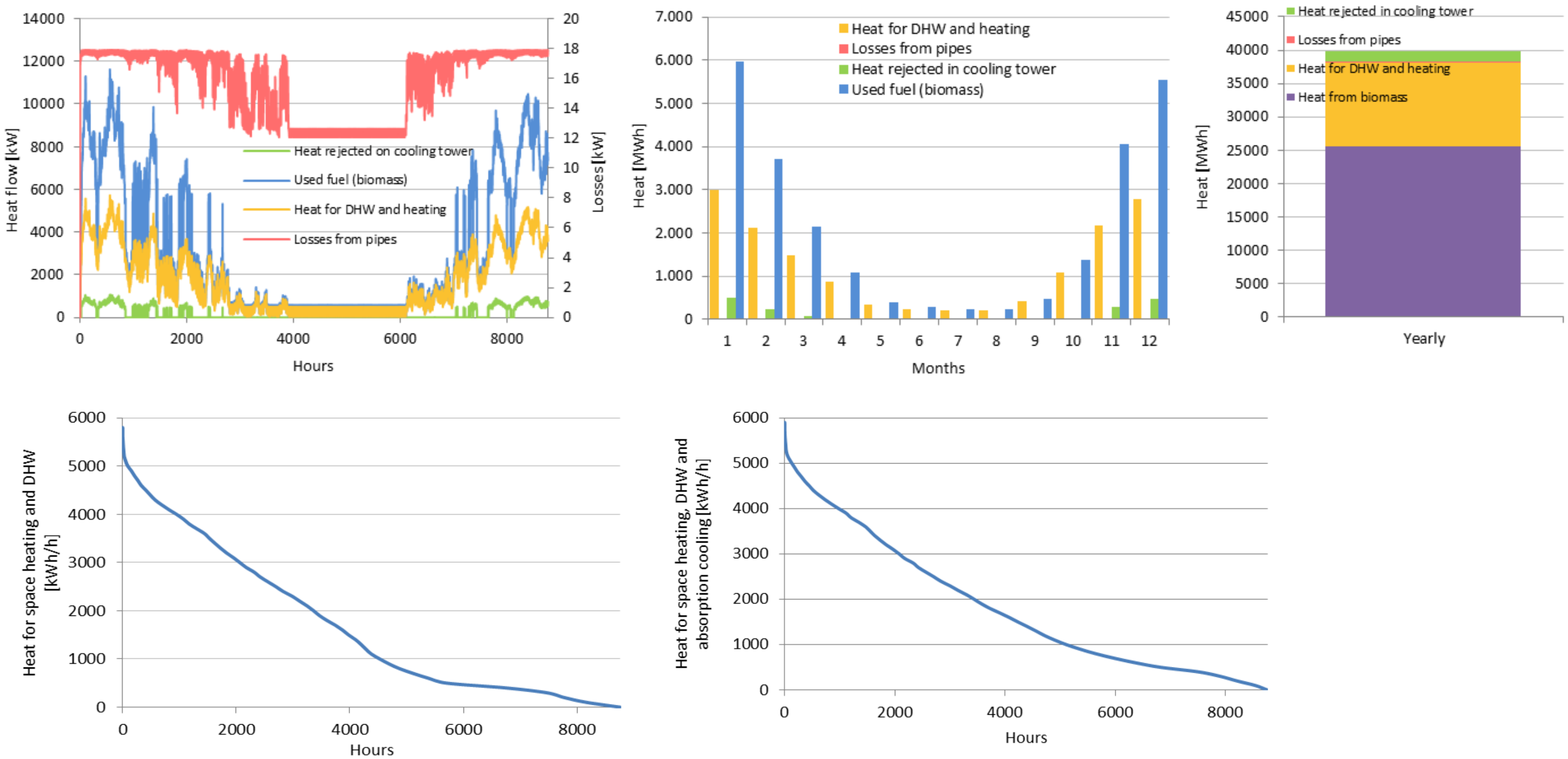


Figure 8.1 Histogram of needed heat for space heating, DHW (and absorption cooling)

First row shows how solar radiation and ambient temperature change during the year, what is needed energy for DHW and heating on a monthly basis and also yearly. From the second picture it can be seen that energy for heating varies between months significantly from around 2800 MWh in January to 0 MWh in summer months. Energy for DHW remains the same for all months. Needed energy for both of them is just a little less than 15000 MWh.

Second row shows how mass flow of cooling and hot water change during the year. One can notice that there is correlation between them and heat demand for heating and DHW, which means that they are almost zero in the summer time. The same trend have number of hours when cogeneration is operating in the second figure, which means that the heat load between May and September is too small for cogeneration to work. On the other hand boiler is operating all the time only load is changing.

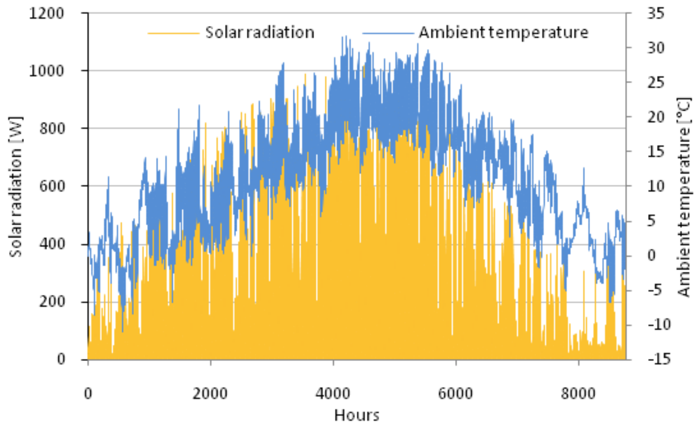
Third row shows heat flow for cogeneration and also for heating and DHW during year. Heat flow for heating and DHW is greater than that for cogeneration, which is expected since cogeneration unit is relatively small (280 kW). Heat is always required, also in the summer time when it is used for DHW preparation. All in all system has to be provided with 19344 MWh yearly.

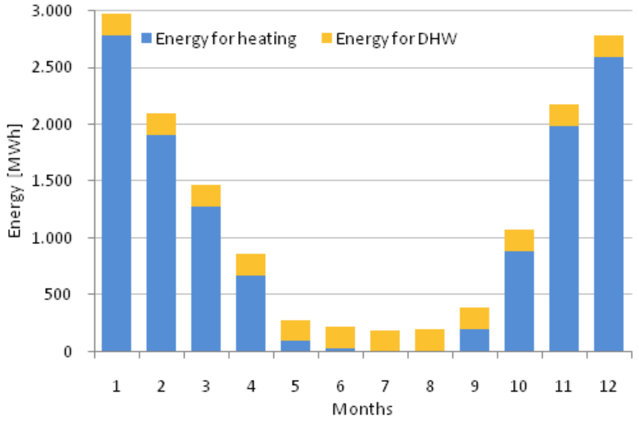
Fourth row, first figure shows heat flow into cogeneration, rejected heat flow in cooling tower and electrical power from cogeneration. First thing that leaps into the eyes is that electrical power really small is in comparison to heat flow into cogeneration. The same can be concluded from second and third figure. On a yearly basis generated electricity amounts to 382 MWh and heat put into cogeneration is almost 18 times bigger (6768 MWh).

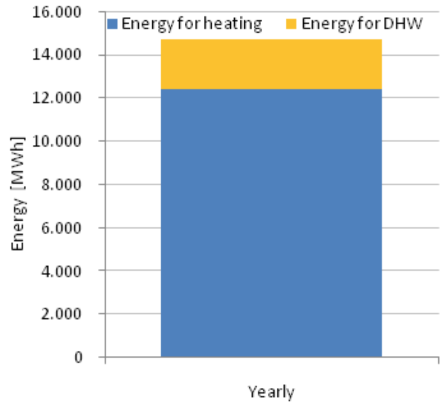
Fifth row shows comparison of different heat flows. Used fuel refers to heat flow delivered with biomass and is calculated as a sum of heat flow into cogeneration and heat flow for heating and DHW taking into account efficiency of biomass boiler (different sources state different efficiencies, but the best approximation is 85%). On a yearly basis required primary energy to cover all needs equals 22758 MWh. Next thing that can be concluded is that losses from pipes are so small that they can be neglected.

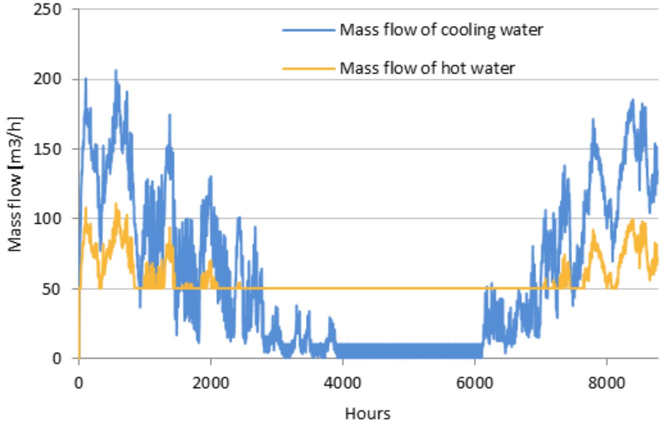
From the histograms it is possible to see what the peak demand in this system is. Beside that a number of hours when certain amount of heat has to be supplied can be obtained. Peak demand is around 5800 kWh/h and it occurs in the winter time. Curves have approximately the same shape, except in the second scenario consumption is a bit bigger due to absorption cooling

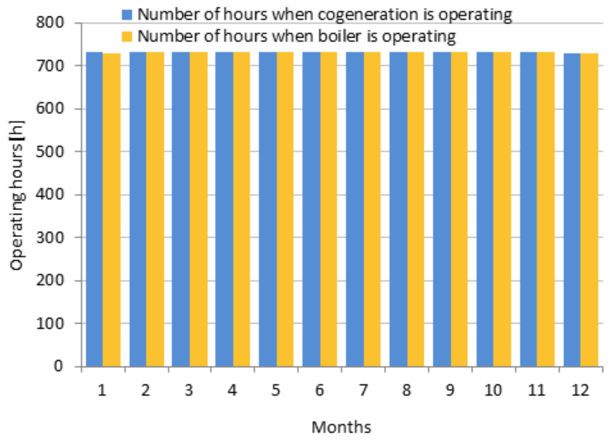
FULL TIME OPERATION

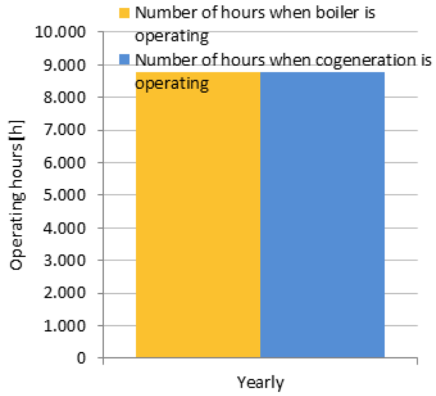




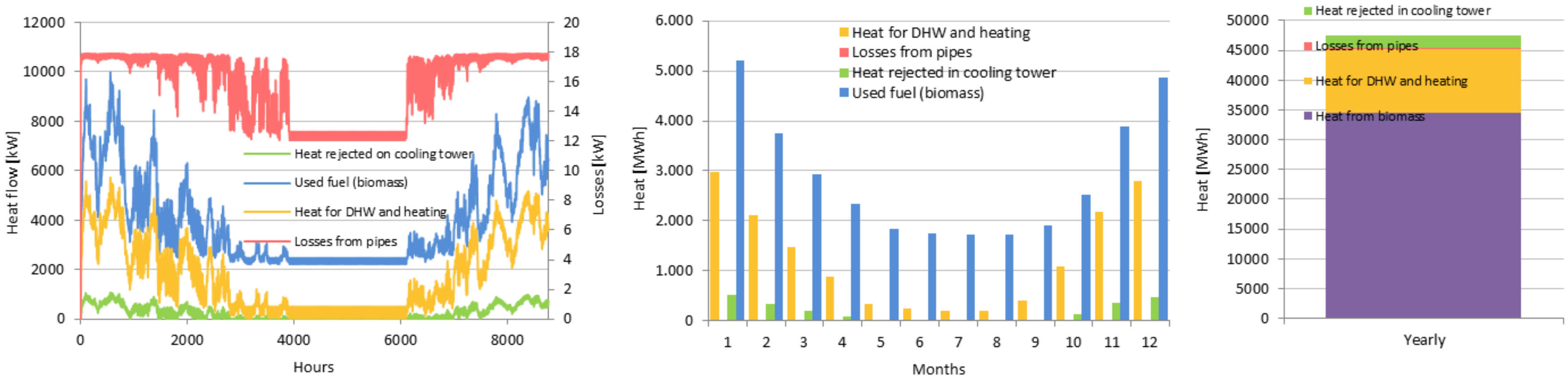









Heat flow [kW]
Hours
Heat flow for cogeneration
Heat flow for heating and DHW
Heat [MWh]
Months
Heat for heating and DHW
Heat for cogeneration
Heat [MWh]
Yearly
Heat for heating and DHW
Heat for cogeneration
Heat flow, el. power [kW]
Hours
Heat flow into cogeneration
Rejected heat flow in cooling tower
Electrical power
Heat, electricity [MWh]
Months
Generated electricity
Heat put into cogeneration
Rejected heat in cooling tower
Additional heat directly from hot water
Heat, electricity [MWh]
Yearly
Additional heat directly from hot water
Rejected heat in cooling tower
Heat put into cogeneration
Generated electricity


Heat flow [kW]
Losses [kW]
Hours
Heat rejected on cooling tower
Used fuel (biomass)
Heat for DHW and heating
Losses from pipes
Heat [MWh]
Months
Heat for DHW and heating
Losses from pipes
Heat rejected in cooling tower
Used fuel (biomass)
Heat [MWh]
Yearly
Heat rejected in cooling tower
Losses from pipes
Heat for DHW and heating
Heat from biomass

Diagrams in the first row are the same as in previous case.

From the second row one can notice that mass flow of hot water in this case remains constant during summer because cogeneration is operating and hot water needs to be supplied. When looking at number of hours when boiler and cogeneration are operating it is obvious that they are the same and operate all the time.

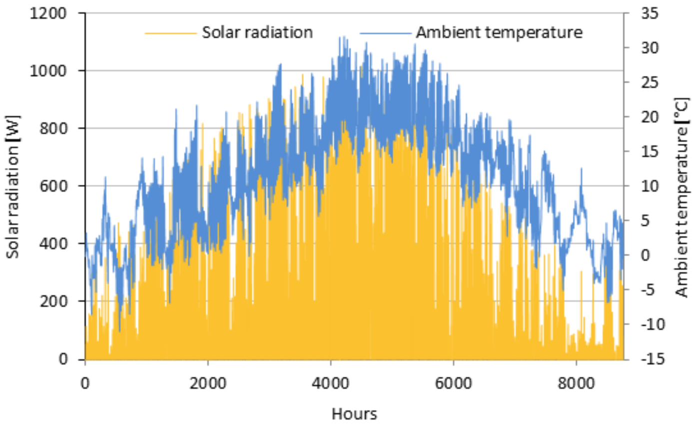
Heat flow for cogeneration has the same pattern as mass flow of hot water since there is correlation between them. When comparing heat for cogeneration in both cases it is evidently that in this case heat for cogeneration on a yearly basis increases from about 6500 MWh to 18500 MWh. One would expect that the rejected heat in cooling tower would increase significantly but this is not the case (third row, second figure). The reason is that only a needed portion of mass flow of cooling water is heated up to 55°C and rest goes directly to the cooling tower. On a yearly basis this gives 2100 MWh (in previous case was 1600 MWh) and electrical energy increases for 600 MWh (from 400 MWh to 1000 MWh).

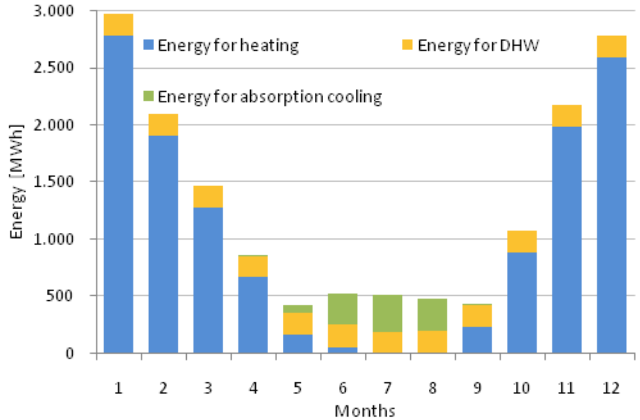
The main conclusion from fifth row would be that heat from biomass has increased for around 13000 MWh/year up to 34449 MWh/year . What the relationships between heat and electrical energy are will be presented later.

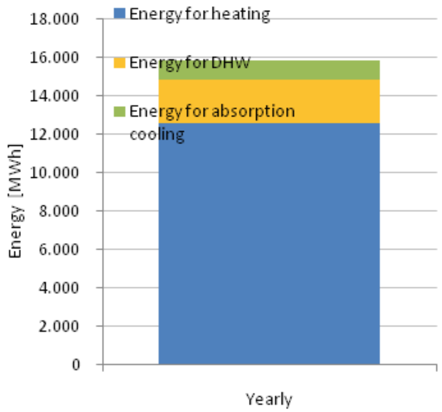
SCENARIO WITH HIGH TEMPERATURE COGENERATION

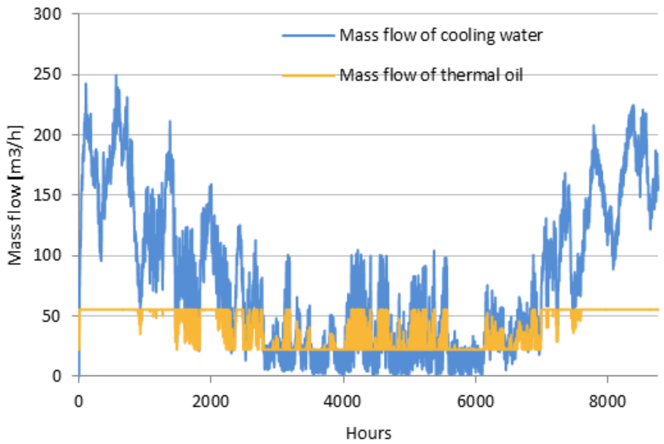
BIOMASS BOILER - existing		**COGENERATION UNIT: ORC - TURBODEN**	
Thermal power	1 X 2,5 MW	Max. thermal power input	2,3 MW
THERMAL OIL BOILER		Range of generated electrical power	kW - 751 kW
Outlet temperature	300°C	Part load operation	40 – 100%
Inlet temperature	240°C	Outlet temperature of cooling water	80°C
		Inlet temperature of cooling water	60°C
DOMESTIC WATER INLET	12°C		
		BUILDINGS	
SPACE HEATING		Peak load	5524 kW
Required inlet temperature	65°C	Floor area	60000 m^2
		Area of walls	
		Indoor temperature - summer	25°C
HOT WATER TANK		Indoor temperature - winter	20°C
Minimum inlet temperature	55°C		

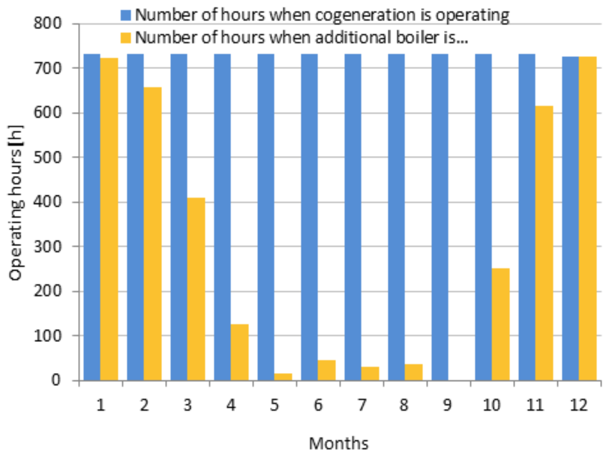
FULL TIME OPERATION

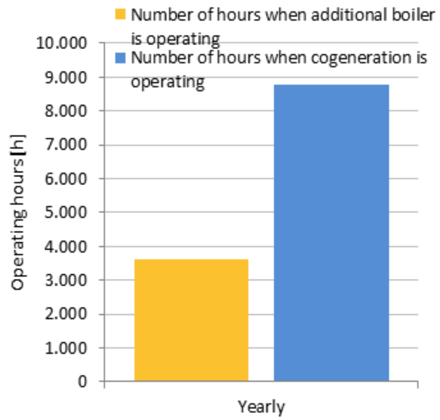


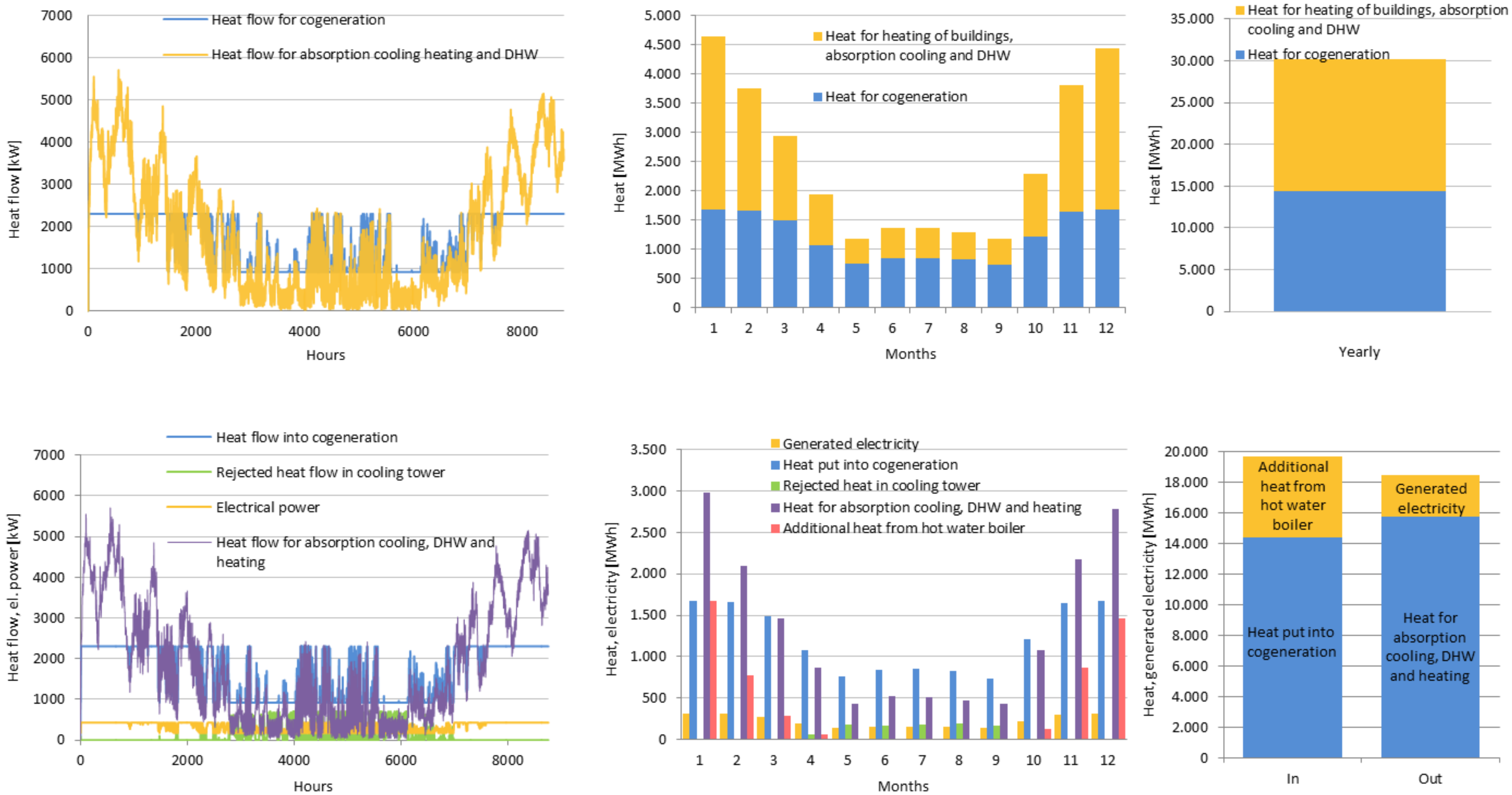




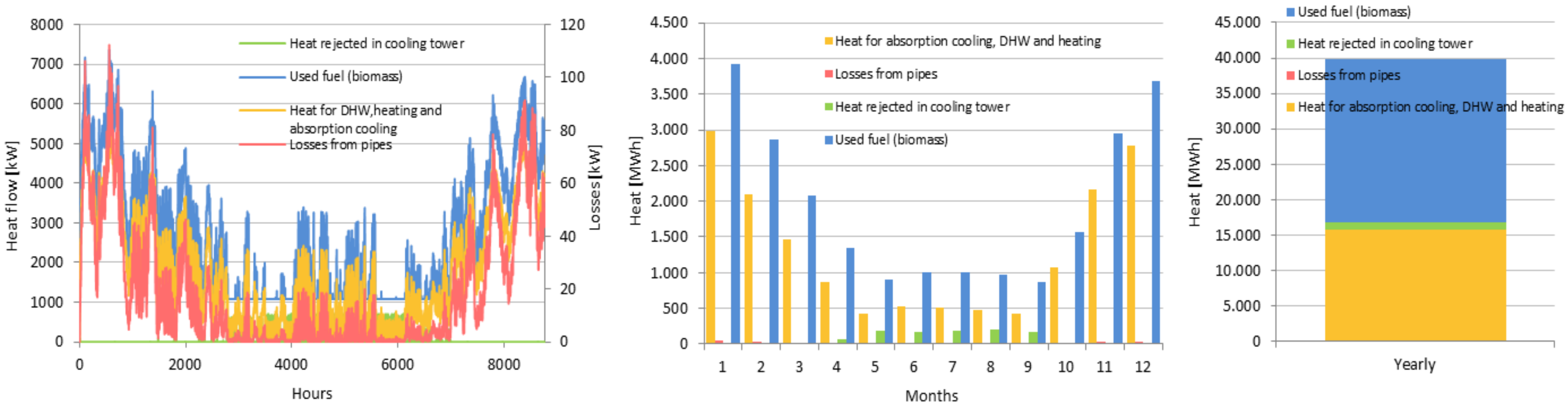








Heat flow for cogeneration
Heat flow for absorption cooling heating and DHW
Heat flow [kW]
Hours
Heat for heating of buildings, absorption cooling and DHW
Heat for cogeneration
Heat [MWh]
Months
Heat for heating of buildings, absorption cooling and DHW
Heat for cogeneration
Heat [MWh]
Yearly
Heat flow into cogeneration
Rejected heat flow in cooling tower
Electrical power
Heat flow for absorption cooling, DHW and heating
Heat flow, el. power [kW]
Hours
Generated electricity
Heat put into cogeneration
Rejected heat in cooling tower
Heat for absorption cooling, DHW and heating
Additional heat from hot water boiler
Heat, electricity [MWh]
Months
Additional heat from hot water boiler
Heat put into cogeneration
Generated electricity
Heat for absorption cooling, DHW and heating
Heat, generated electricity [MWh]
In
Out



Heat flow [kW]
Losses [kW]
Hours
Heat rejected in cooling tower
Used fuel (biomass)
Heat for DHW,heating and absorption cooling
Losses from pipes
Heat [MWh]
Months
Heat for absorption cooling, DHW and heating
Losses from pipes
Heat rejected in cooling tower
Used fuel (biomass)
Heat [MWh]
Yearly
Used fuel (biomass)
Heat rejected in cooling tower
Losses from pipes
Heat for absorption cooling, DHW and heating

First row shows again how solar radiation and ambient temperature change during the year, what is needed energy for DHW, heating and absorption cooling. We can see that heat load in summer increases for about 300 MWh per month due to absorption cooling, which means this could have impact on operation of cogeneration. Yearly heat demand increases for a little bit less than 1000 MWh since we also account cooling need and this gives us about 16000 MWh needed heat per year.

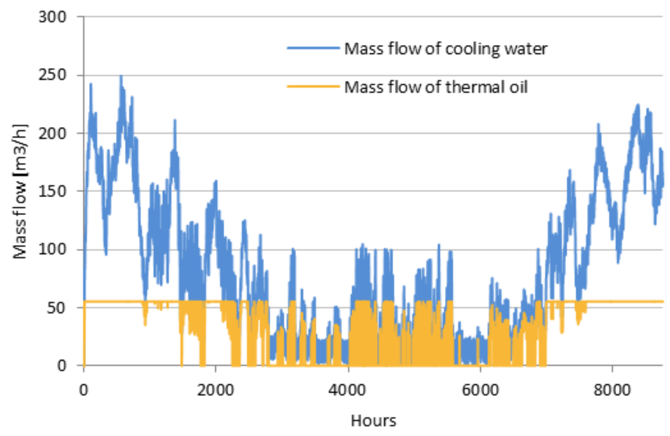
Second row shows how mass flows of cooling water and thermal oil change during the year. Mass flow of cooling water decreases during summer because heat load decreases at that time. On the other hand mass flow of thermal oil changes only in a certain range. This is due to the fact that this scenario assumes full time operation of cogeneration. When heat load is high enough, than mass flow is at its maximum (winter time, middle of the summer), whereas in the transitional period (spring, autumn) drops down to the minimum. When looking at second figure it is obviously that cogeneration is operating all the time, whereas boiler has to run mostly in the winter time. On a yearly basis this means that cogeneration will work 8760 hours and additional boiler for a bit less than 3600 hours.

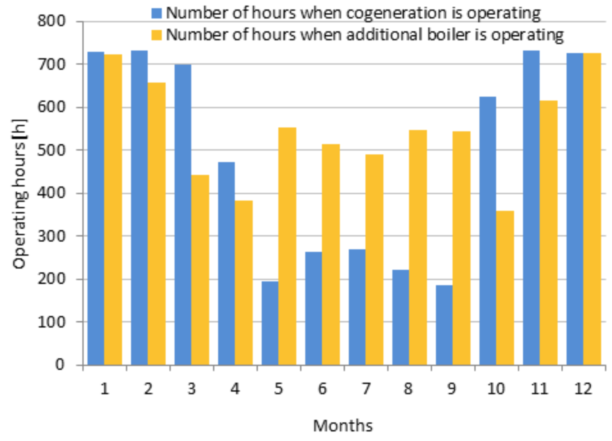
Third row shows heat flow for cogeneration and heat flow for absorption cooling, heating and DHW during year. Heat flow for absorption cooling, heating and DHW is in the winter time greater than that for cogeneration, which means that cogeneration will work at 100% and also additional boiler will operate. On a yearly basis one can notice that heat put into cogeneration and additional heat (marked as 'in') are bigger than heat for absorption cooling, heating and DHW and generated electricity (marked as 'out'). The difference is heat that is rejected in cooling tower. If 'out' is devided by 'in' than this ratio gives us 0,937.

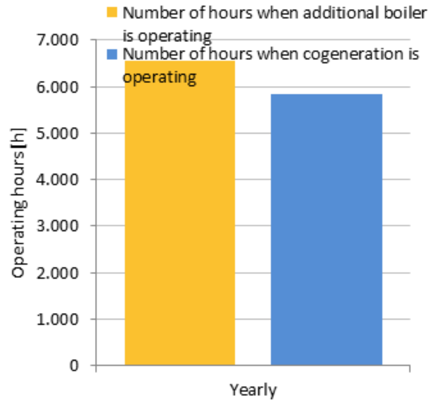
In the first figure in fourth row are compared: heat flow into cogeneration, rejected heat flow in cooling tower, heat flow for absorption cooling, heating and DHW and electrical power. When comparing heat flow into cogeneration and electrical power it is obvious that efficiency is not really high, but heat can be used for heating. On a monthly basis it is interesting to compare rejected heat in cooling tower and additional heat from hot water boiler. Cooling tower works only in the summer time whereas hot water boiler at that time reaches its minimum. This is expected since the heat load decreases. It needs to be mentioned that it should not happen that heat is being rejected from cooling tower at the same time as hot water boiler is operating (valid on an hourly basis). This was checked and so we could confirm correct functioning of the program. One can notice that quite a lot of heat needs to be supplied additionally in the winter time, which suggests that bigger unit could be installed. Unit with 5,1 MW thermal input was chosen. In that case we would generate more electricity (5000 MWh/year) and cover more heat load in the winter time, but in the summer time more heat would be rejected from the cooling tower. Also additional boiler would work for less than 1000 hours. Graphs for this case could be found in Appendix I.

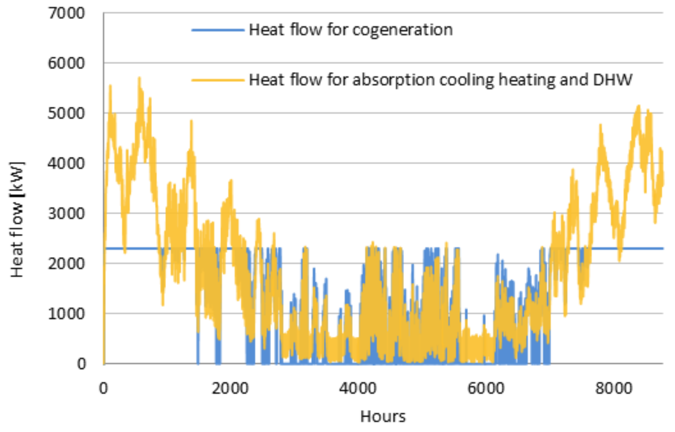
Fifth row again shows comparison of different heat flows. Compared to the first scenario now heat demand increases but biomass consumption remains approximately the same. The reason for that is that efficiency of cogeneration is in this scenario higher.

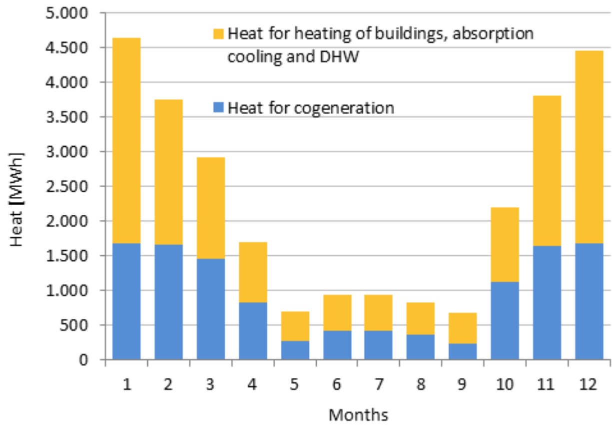
PART TIME OPERATION

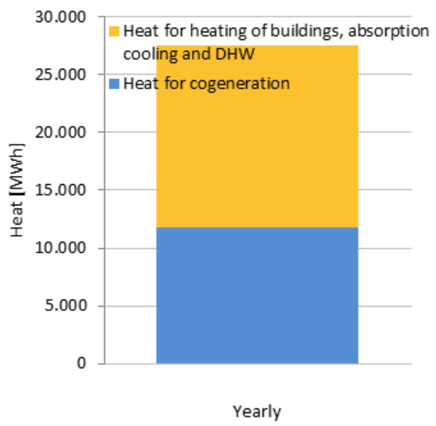


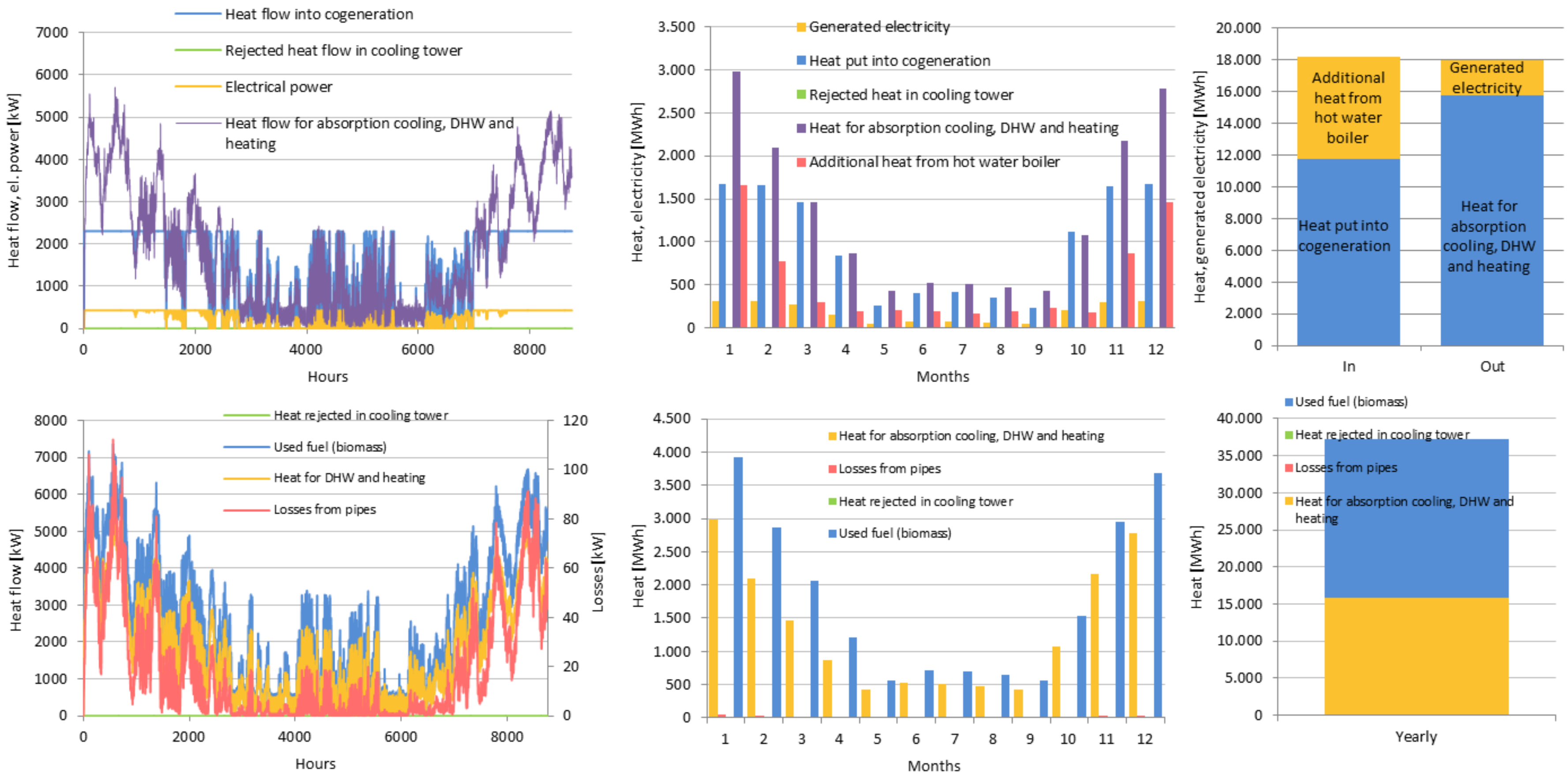












Heat flow into cogeneration
Rejected heat flow in cooling tower
Electrical power
Heat flow for absorption cooling, DHW and heating
Heat flow, el. power [kW]
Hours
Generated electricity
Heat put into cogeneration
Rejected heat in cooling tower
Heat for absorption cooling, DHW and heating
Additional heat from hot water boiler
Heat, electricity [MWh]
Months
Heat, generated electricity [MWh]
Additional heat from hot water boiler
Heat put into cogeneration
Generated electricity
Heat for absorption cooling, DHW and heating
In
Out
Heat rejected in cooling tower
Used fuel (biomass)
Heat for DHW and heating
Losses from pipes
Heat flow [kW]
Losses [kW]
Hours
Heat for absorption cooling, DHW and heating
Losses from pipes
Heat rejected in cooling tower
Used fuel (biomass)
Heat [MWh]
Months
Used fuel (biomass)
Heat rejected in cooling tower
Losses from pipes
Heat for absorption cooling, DHW and heating
Heat [MWh]
Yearly

From the first figure in the second row one can notice that mass flow of thermal oil is not anymore in specific range but it drops down to zero. This happens when the heat load so small is that we would need to reject heat in cooling tower, which is not acceptable for this case. Because of this reason more heat needs to be supplied from hot water boiler. Compared with previous case, where additional boiler was operating for 3633 hours, now this value has increased up to 6546 hours. On the other hand number of hours when cogeneration is operating has dropped down to 5846 hours per year.
When comparing heat for cogeneration (third row) in both cases it is logically that it is bigger in the first case. On a yearly basis this means 14419 MWh for full time operation and 11739 MWh for part time.
In fourth row it is interesting to compare the last figure with figure from previous case. Now 'in' and 'out' are almost the same (ratio is 0,987), since there is no heat rejected in cooling tower. Increased has additional heat from hot water boiler whereas heat put into cogeneration and also generated electricity are in this case smaller.
From the last row, last figure, it can be concluded that in this case needed primary energy from biomass is approximately 2000 MWh smaller than when cogeneration is operating full time. In that case is 23165 MWh.

Additional case was assuming high temperature cogeneration but without cooling. Whole system is the same as before except heat load is lower. If cases with full time operation are compared (Figure 8.2) than it can be concluded that less heat is being rejected when there is cooling load. In this case 969 MWh is rejected, but in case without cooling this number is 1449 MWh.

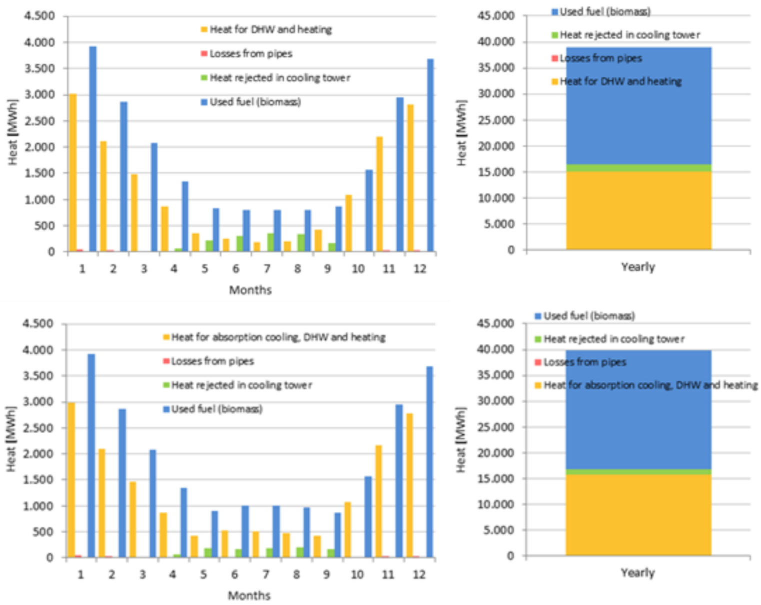


Figure 8.2 Results for high temperature cogeneration, full time, a.) without and b.) with cooling

When looking at the same scenario, but part time operation, the most interesting result is to compare number of hours when cogeneration is operating. One can notice that number of hours increases from almost 5846 to around 5074 and at te same time boiler operates for 646 hours less. It means when absorption cooling is added to the DHS we will make the best of cogeneration and generate more electricity.

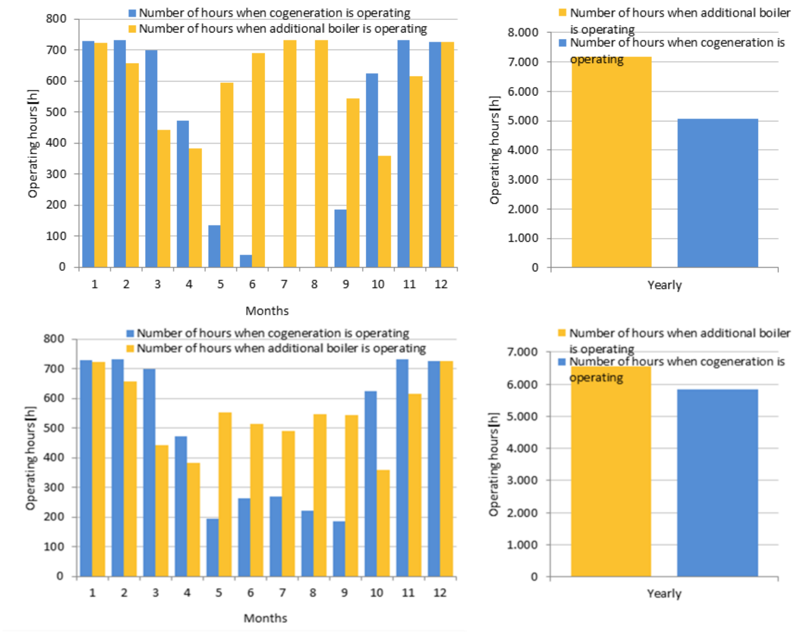


Figure 8.3 Results for high temperature cogeneration, part time, a.) without and b.) with cooling

Table 8.1 Performance indicators for different cases

	Low temperature cogeneration		High temperature cogeneration			
			With cooling		Without cooling	
	full time	part time	full time	part time	full time	part time
$E_{el}/Q_{biomass}$	0,026	0,015	0,116	0,102	0,115	0,097
$Q_{heating,abs,DHW}$ $/Q_{out\ cogen}$	0,877	0,901	0,943	1,00	0,912	1,00
$Q_{heating,abs,DHW}$ $/Q_{biomass}$	0,379	0,585	0,690	0,735	0,666	0,742
$Q_{ct}/Q_{biomass}$	0,0542	0,0634	0,0418	0	0,064	0
$(E_{el} + Q_{heating,abs,DHW})$ $/Q_{biomass}$	0,404	0,599	0,804	0,839	0,781	0,839
Number of hours when cogeneration is working	8760	2499	8760	5846	8760	5074

In order to evaluate different scenarios and cases several performance indicators were chosen that are presented in the Table 8.1. When looking at the first one (generated electricity divided by heat from biomass) it can be concluded that cases with full time operation have higher ration than part time. Moreover high temperature has higher ratio than low temperature, which holds true for all performance indicator (except fourth one). Second indicator (heat load divided by heat from cogeneration which is used for heat load) has the highest values and in the case of part time high temperature cogeneration is equal to 1 which means no heat is wasted. Third indicator (heat load divided by heat from biomass) represents how much heat from biomass is used for heat load. One can see that this value is low for full time low temperature cogeneration. Next indicator connects heat rejected in cooling tower and generated electricity. Logically is this value for part time operation smaller since this case is optimized in a way that Q_{ct} approaches 0. It should not be overlooked that for part time low temperature cogeneration this ratio equals 4,24. Last indicator is basically efficiency of cogeneration. Sum of heat supplied to DHS and generated electricity is divided by heat supplied with biomass. The highest efficiencies have cases with part time operation.

9 CONCLUSIONS

Main idea in the frame of this thesis was to observe impacts of heat demand on operating and performance of cogeneration and to develop model which will include thermal response of district heating system (heating and cooling of buildings) and electricity generation. General model was applied to the study case and two scenarios were developed each of them having two cases.

For cases where low temperature heating is needed low temperature cogeneration scenario with 280 kW maximum electrical power was made. After analysis the main conclusion is that it has relatively poor performance. This is logical because temperatures of hot water entering and leaving cogeneration are low (110°C and 78°C respectively) and do not allow higher efficiencies.

Simulation was also carried out for high temperature cogeneration with 427 kW maximum electrical power and in this scenario performance is better for two reasons: higher temperatures and heat consumption in summer time (absorption cooling) but the problem is that new boiler has to be installed. (In the first case this is not necessary since only hot water from existing boiler is being used). Efficiency is higher for part time operation (83,9%), because we are minimizing rejected heat in cooling tower. Idea was also to install bigger cogeneration unit (with 427 kW maximum electrical power 737 kW) which would cover bigger share of heat load and in that case efficiency is 83,8% for part time operation. When choosing the best option on the basis of defined efficiency then high temperature cogeneration seems to be the best one.

In this point it has to be mentioned that exergy analysis wasn't included even if heat and generated electricity do not have the same value.

9.1 Suggestions for future work

Proposed was best option from thermodynamic point of view but when also investment and operating costs were included then solution could change. Therefore economical analysis would be necessary which would link together performance indicators and costs.

Another suggestion is to observe the impacts of lowering heat losses of the buildings on DHS. As known more and more people are deciding for passive standard of houses which lowers needed heat. This idea could be even extended to supply consumers with cold from central absorption cooling unit whereas heat would be partly produced from renewable energy sources (for example solar collectors).

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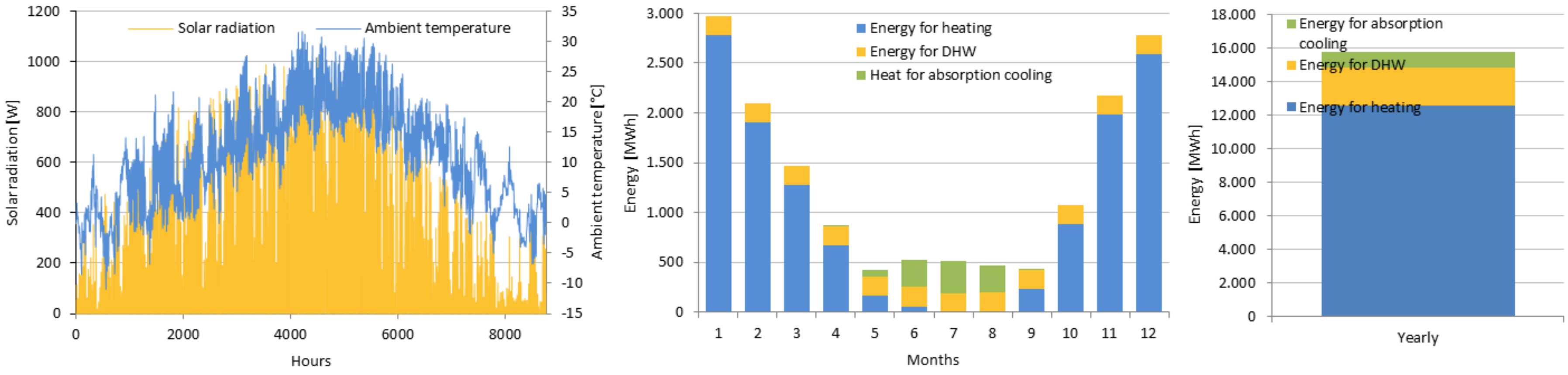
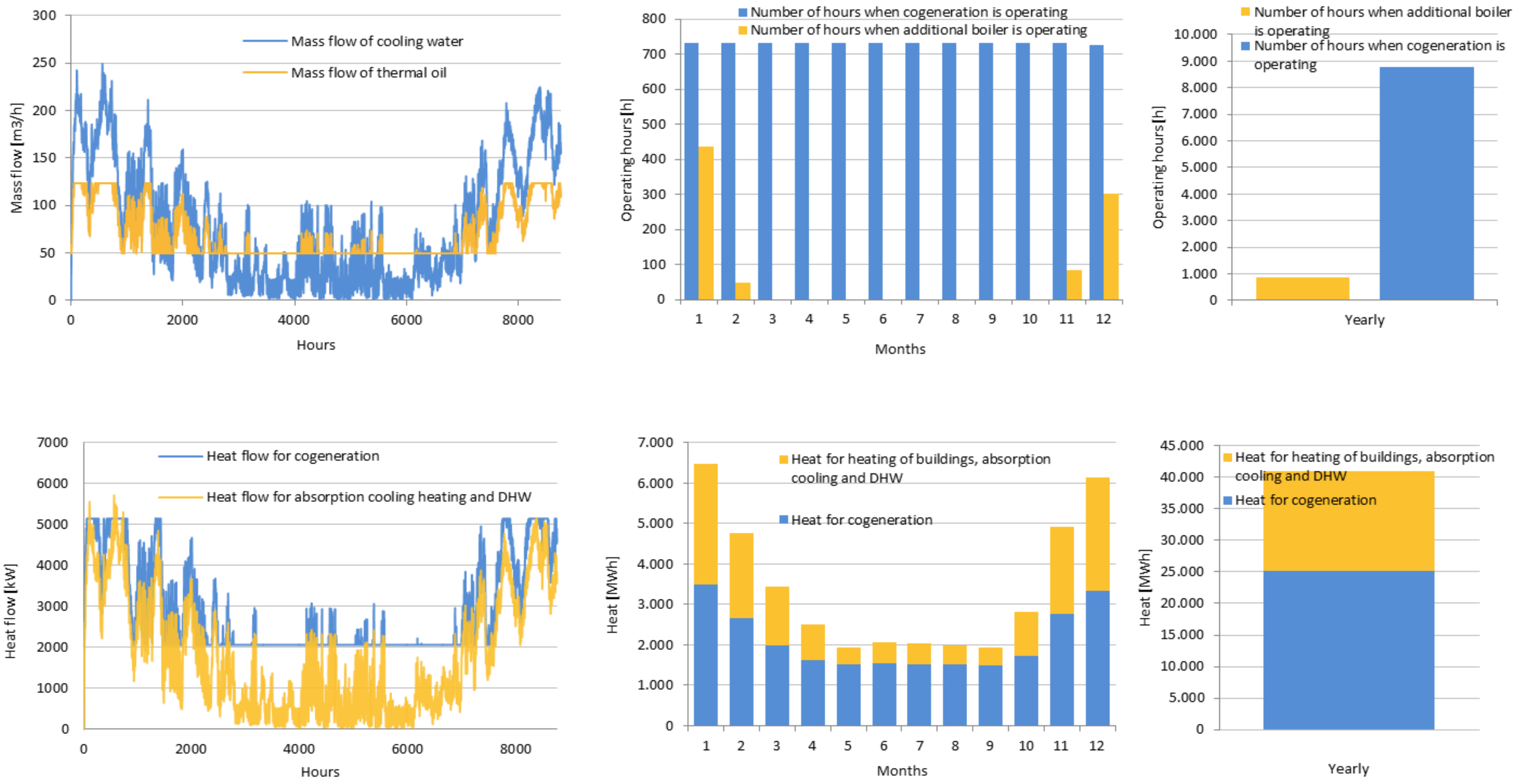
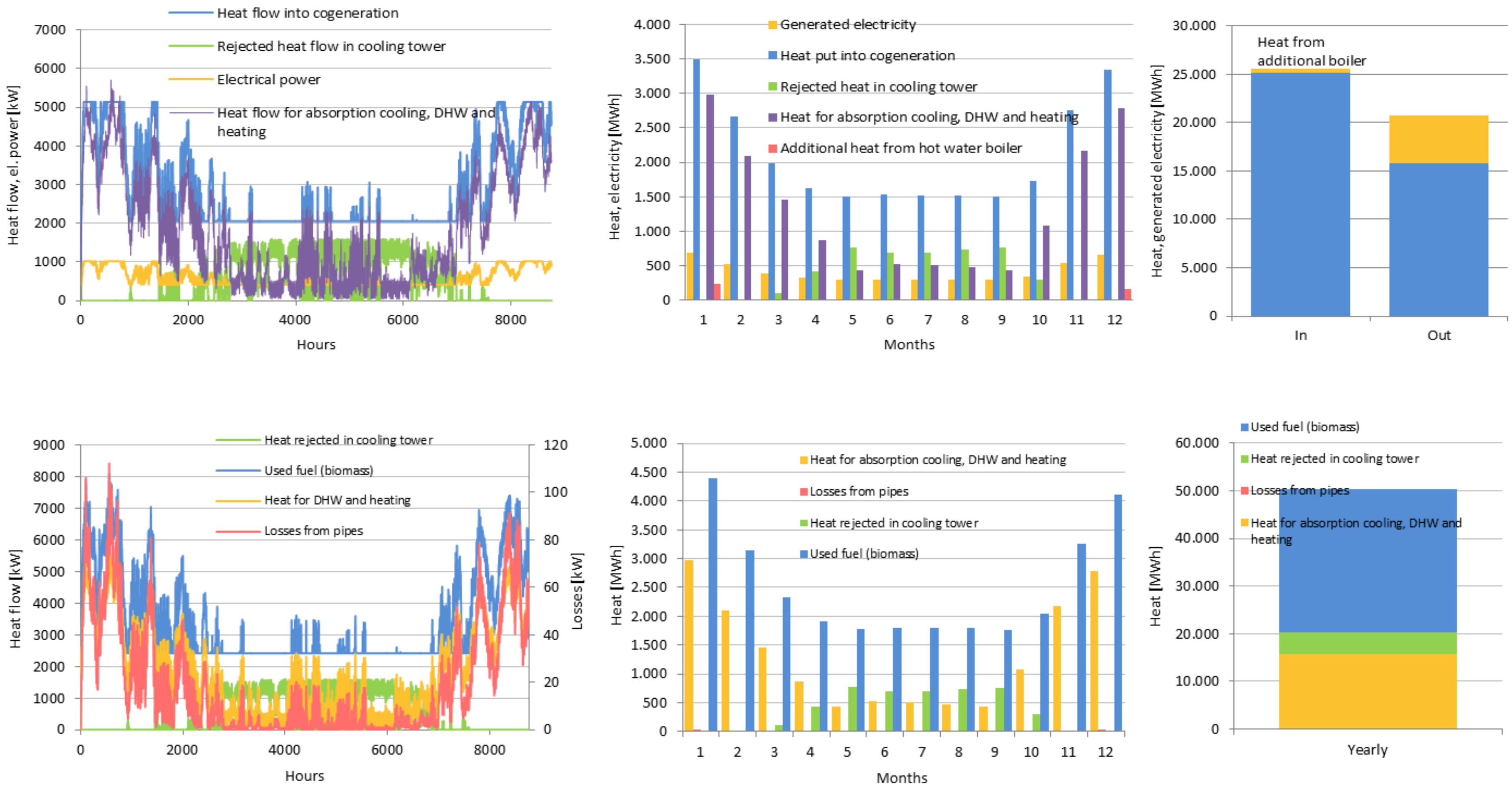
APPENDIX A

FULL TIME OPERATION OF 5,1 MW THERMAL INPUT

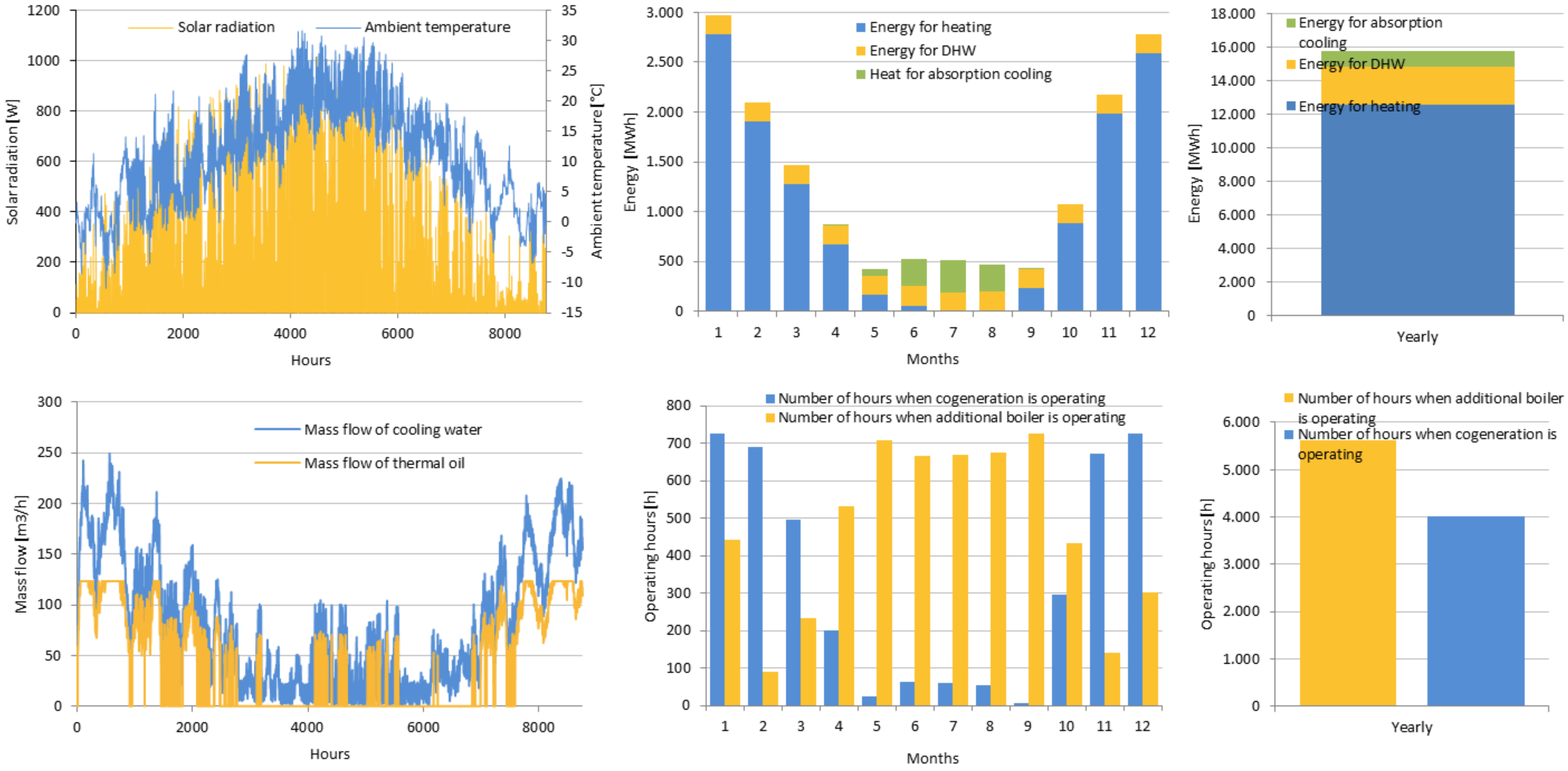




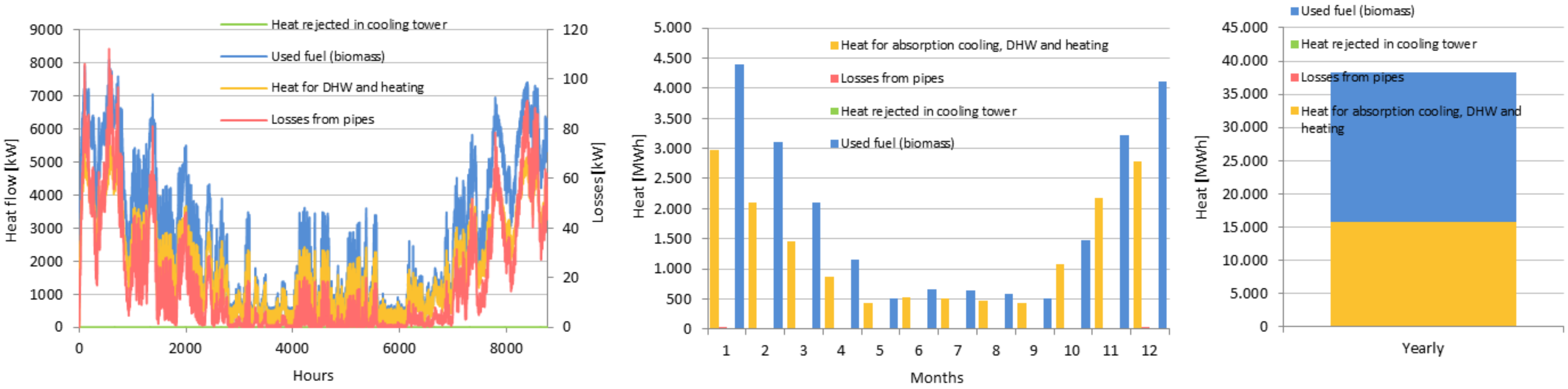
Mass flow of cooling water
Mass flow of thermal oil
Mass flow [m3/h]
Hours
Number of hours when cogeneration is operating
Number of hours when additional boiler is operating
Operating hours [h]
Months
Number of hours when additional boiler is operating
Number of hours when cogeneration is operating
Operating hours [h]
Yearly
Heat flow for cogeneration
Heat flow for absorption cooling heating and DHW
Heat flow [kW]
Hours
Heat for heating of buildings, absorption cooling and DHW
Heat for cogeneration
Heat [MWh]
Months
Heat for heating of buildings, absorption cooling and DHW
Heat for cogeneration
Heat [MWh]
Yearly


Heat flow into cogeneration
Rejected heat flow in cooling tower
Electrical power
Heat flow for absorption cooling, DHW and heating
Heat flow, el. power [kW]
Hours
Generated electricity
Heat put into cogeneration
Rejected heat in cooling tower
Heat for absorption cooling, DHW and heating
Additional heat from hot water boiler
Heat, electricity [MWh]
Months
Heat from additional boiler
Heat, generated electricity [MWh]
In
Out
Heat rejected in cooling tower
Used fuel (biomass)
Heat for DHW and heating
Losses from pipes
Heat flow [kW]
Losses [kW]
Hours
Heat for absorption cooling, DHW and heating
Losses from pipes
Heat rejected in cooling tower
Used fuel (biomass)
Heat [MWh]
Months
Used fuel (biomass)
Heat rejected in cooling tower
Losses from pipes
Heat for absorption cooling, DHW and heating
Heat [MWh]
Yearly

PART TIME OPERATION OF 5,1 MW THERMAL INPUT



Heat flow for cogeneration
Heat flow for absorption cooling heating and DHW
Heat flow [kW]
Hours
Heat for heating of buildings, absorption cooling and DHW
Heat for cogeneration
Heat [MWh]
Months
Yearly
Heat flow into cogeneration
Rejected heat flow in cooling tower
Electrical power
Heat flow for absorption cooling, DHW and heating
Heat flow, el. power [kW]
Generated electricity
Heat put into cogeneration
Rejected heat in cooling tower
Heat for absorption cooling, DHW and heating
Additional heat from hot water boiler
Heat, electricity [MWh]
Heat, generated electricity [MWh]
In
Out


Heat rejected in cooling tower
Used fuel (biomass)
Heat for DHW and heating
Losses from pipes
Heat flow [kW]
Losses [kW]
Hours
Heat for absorption cooling, DHW and heating
Losses from pipes
Heat rejected in cooling tower
Used fuel (biomass)
Heat [MWh]
Months
Used fuel (biomass)
Heat rejected in cooling tower
Losses from pipes
Heat for absorption cooling, DHW and heating
Heat [MWh]
Yearly