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Design and Optimization of Standardized Organic Rankine Cycle Power Plant for European Conditions

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DESIGN AND OPTIMIZATION OF STANDARDIZED ORGANIC RANKINE CYCLE POWER PLANT FOR EUROPEAN CONDITIONS

Maciej Lukawski

A 30 credit units Master's thesis

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ABSTRACT

This paper investigates the possibility of introducing universally designed binary power plants into European energy markets. ORC cycles are found to be particularly useful not only for the production of electricity from geothermal water, but also for the recovery of waste heat from engine exhaust gasses, furnaces and drying ovens. In this dissertation, an analysis of market demand and thermodynamic characteristics of different heat sources is performed in order to find an optimal set of design boundary conditions maximizing unit performance for the most promising types of applications.

A thermodynamic model of a power plant using a wet mechanical-draft cooling tower is created in EES software and a detailed analysis of component configuration and parameters of working fluids is carried out.

Optimal plant configuration and size of components is found by thermoeconomic optimization. Exergy flow rates of all streams in the system as well as rates of exergy destruction and loss are quantified. Detailed economic analysis of the unit is made for four different applications: geothermal plants using water from conventional hydrothermal wells, former oil and gas boreholes, waste heat recovery plants coupled with diesel engines, and a clinker cooler in a cement plant.

Finally, a sensitivity analysis shows the impact that changes in heat source characteristics and macroeconomic variables have on levelized cost of power.

PREFACE

During the last few years there has been a significant increase in the European renewable energy market. It is believed that geothermal energy, as one of the most reliable and inexpensive sources of green energy will contribute much to the reduction of fossil fuel dependency and help to fulfill ambitious targets set by the European Commission for Member Countries. Since the temperature of obtainable geothermal waters in the majority of European countries is not extraordinary high, electricity production from these resources can be targeted almost exclusively by binary power plants. Another on-going challenge is improving the efficiency of thermal processes in industry. With continually increasing use of energy, the potential for waste heat recovery becomes significant.

High market demand for binary power plants presents an opportunity for mass production of units. This thesis investigates a possibility of introducing standardized Organic Rankine Cycle power plants to the European energy market. The unit is designed in such a way that it will be able to operate in various types of applications with different boundary conditions.

The traditional approach to design and optimization of power plants relies on thermodynamics and aims in maximizing fuel utilization efficiency. However, such a simple method which is commonly applied for fossil fuel power plants is not suitable for binary units. This paper presents a method of thermoeconomic optimization of power plants, which is based on exergy analysis. By assessing the magnitude of irreversibilities occuring in the system and estimating costs related to each component in the system, this method allows the cost-optimal design of each single component to be found. In contrast to thermodynamic optimization this approach does not minimize the inefficiencies of the system, but allows an optimum to be found, from the point of view of final product, values.

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1 MARKET RESEARCH

1.1 Geothermal applications

The European Union is currently the world´s dominant driving force towards the sustainable production of energy and a leader in implementing renewable energy policies. But, at the same time, it is the world´s biggest importer of energy. In order to change this situation by ensuring security of supply and competitive prices, reducing climate changes, which are already considered unavoidable, the European Commission set ambitious targets for member states to increase their share of energy produced from renewable sources. Apart from the Kyoto protocol, which obligates EU members to reduce their greenhouse gas emissions by 8% below the 1990 level in years 2008-2012, Europe also aims at increasing their share of power production from renewable sources to 12% by 2010 and 20% by 2020.

All these targets cannot be met without simultaneous development of different types of renewable energy: solar, wind, biomass hydro- and geothermal. During the last decade, much attention has been paid to the development of wind and solar power and such systems in many countries were highly subsidized. Both these sources of energy are important as they have vast potential, but they also share an important drawback which is a low stability of supply. Neither wind, nor solar energy can be used as a base load supply. A low capacity factor, varying between 20% and 40%, allows using them only as an auxiliary power supply to the electric grid. Geothermal energy shares many advantages of these renewable sources. Yet it may and should be used as a base load capacity, as the capacity factor of geothermal power plants usually exceeds 85%. The availability factor, which is the percentage of the time a power plant can operate at its installed capacity equals or even exceeds 90%. This is a very high value even when compared to fossil fuel power plants.

According to research made by Frost & Sullivan, the costs of geothermal power are also dropping. The generation costs of geothermal electricity used to be €50 to €150 per MWh in 2005. This is expected to fall to €40 to €100 per MWh in 2010 and €40 to €80 per MWh in 2020. As electricity generation from geothermal water becomes more affordable, interest in Europe continues to rise.

At the current stage of development, utilization of geothermal energy in Europe is limited to hydrothermal systems. Although research in Enhanced Geothermal Systems (EGS) has been carried out intensively during the last few years and a project in Soultz-sous-Forêts is in its advanced stage, this technology is not yet proven and cannot be expected to become economically feasible in Europe very soon. Currently, almost all reservoirs utilized for electricity production in Europe are of the hydrothermal type and a majority of them are of the high-enthalpy type. Operating power plants in Europe are located only in a few sites, where the best conditions exist. They substantially contribute to the production of electricity in Iceland and Italy and have their share in the market in Turkey and Portugal. Table 1-1 includes data for geothermal power production in Europe proper in 2005 (Antics, Sanner) updated with available data from 2007 (DiPippo, 2008; Earth Policy Institute).

Table 1-1 Geothermal power production in continental Europe in 2005

Country	Installed capacity [MWe]	Running capacity [MWe]	Annual energy production [GWh/yr]	Number of units	% of national electricity demand
Austria	1,2	1,1	3,2	2	Negligible
Germany	0,2	0,2	1,5	1	Negligible
Iceland[1)]	422,4	420,4	2893	24	27,1
Italy	810,5	711	5200	32	1,0
Portugal (San Miguel Island)	16	13	90	5	25 (Azores only)
Turkey	30	30	108	2	Negligible
Total	1280,3	1175,7	8295,7	66	-

[1)] Data from 2007

Apart from small units in Germany and Austria, all plants included in Table 1-1 use resources from high temperature water or dry steam reservoirs, so they operate in conditions which are not typical for the majority of European countries. It is obvious that, because of simpler technology and lower costs of electricity production, high-temperature resources were explored as the first and for a long time the only ones in Europe. However, the quantity and overall potential of these fields are limited. If Europe is to considerably increase its production of electricity from geothermal resources, other geothermal fields with worse temperature characteristics have to be exploited as well. In Europe, such geothermal fields exist on the north and south side of the Alps, Northern Greece, the Pannonian basin of Hungary and the borders of Slovakia, Slovenia, Romania and Serbia, Southern Spain, Upper Rhine and Rhone structures as well as in Northern Germany and Central and Southern Poland (Antics, Sanner) . The locations of the largest high-temperature basins are shown at Figure 1.1.

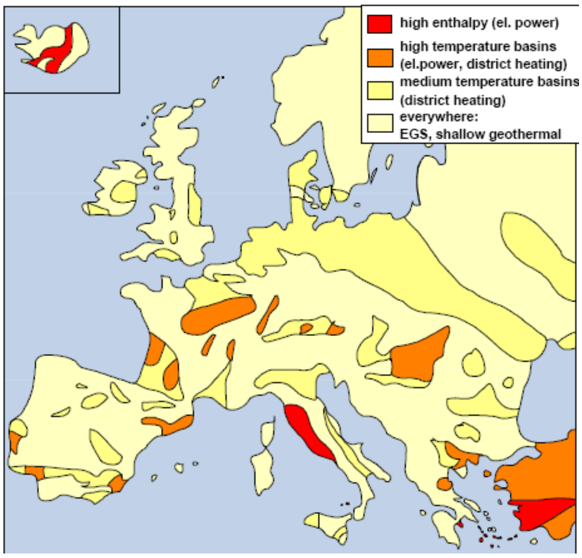

Figure 1.1 Location of geothermal basins in Europe (Antics, Sanner)

Because of the low temperature of geothermal fluid in these reservoirs, geothermal electricity production can be targeted almost exclusively by binary type power plants. These power plants, with over 160 units in operation in 2007, are the most widely used type of geothermal power plants around the world (DiPippo, 2008). Most of them are small units - they generate less than 5% of total power of geothermal power plants. However, they are continuously increasing their share in the market. Binary power plants, either of Organic Rankine Cycle (ORC) or Kalina type, operate usually on 100 to 170°C hot water. However, according to Kestin et al., for temperatures even up to 200°C binary power plants are favored if the content of noncondensable gases in brine is higher than 2%.

As Table 1-1 shows, in 2005 total power capacity and annual generation of electricity in ORC geothermal power plants in Europe was negligible. Only three units rated in total at 0.91 MW were operating on the continent: one of them in Germany (Neustadt-Glewe, online since 2003), and two in Austria (Blumau and Altheim, which started operation in 2001 and 2002, respectively). Another eight units were installed in Iceland: seven in Svartsengi Power Plant working in bottoming cycle and one Kalina type Power Plant in Husavik (DiPippo, 2008). Since 2005, many new projects were initiated, especially in Germany where, after implementing the 2004 Renewable Energy Sources Act, for each kWh of electricity produced in geothermal power plant rated under 5MW, utility receives 0,15 € (Kreuter).

Movement in the geothermal market is also observed in Hungary, where MOL (Hungarian Oil & Gas Company) is considering installing a few ORC units utilizing waste water extracted from oil and gas wells. The exact flow rates which would be possible to obtain from these boreholes are not known yet because initial exploration was focused on hydrocarbons. However, according to studies of Árpási and Kujbus in Hungary, there are 70 to 80 wells suitable for production of geothermal fluids, where expected well-head temperature is above 100°C. The highest well-head temperature measured in one of them was 171°C. According to MOL's present concept, it is possible to establish 3-4 geothermal power plants with 2-5 MW capacity by 2012.

Table 1-2 includes estimations for the existing and ongoing projects in Europe in 2007, where binary power plants (P) were to be installed or the temperature of geothermal water used for district heating (DH) or space heating (SH) was high enough to be used for electricity production (Kranz; Goldbrunner).

Table 1-2 Chosen geothermal projects in Europe

Country	**Location**	**Brine temperature (°C)**	**Brine flow rate (l/s)**	**Planned use**	**Power generation (MWe)**
Germany	Neustadt-Glewe	97	27,8	P, DH	0,2
	Gross Schönebeck	150	20,8	P	1,0
	Offenbach, Bellheim[1)]	150	100	P	3,5
	Bruchsal	120	23,9	P	0,5
	Speyer[2)]	150	125	P, DH	5,4
	Landau in der Pfalz	160	69,4	P, DH	2,5
	Unterhaching	122	150	P, DH	3,7
	Hannover	135	13,9	SH	
Austria	Altheim	105	22,8	P ,DH	0,5
	Blumau	110	8,3	P, DH	0,18

[1)]Project stopped; [2)]Oil was found

Figure 1.2 shows the distribution of boundary conditions: geothermal water temperature and total flow rate in binary power plants included in

Table 1-2. It can be noticed that brine flow rate peaks at two temperatures: 120°C and 150°C.

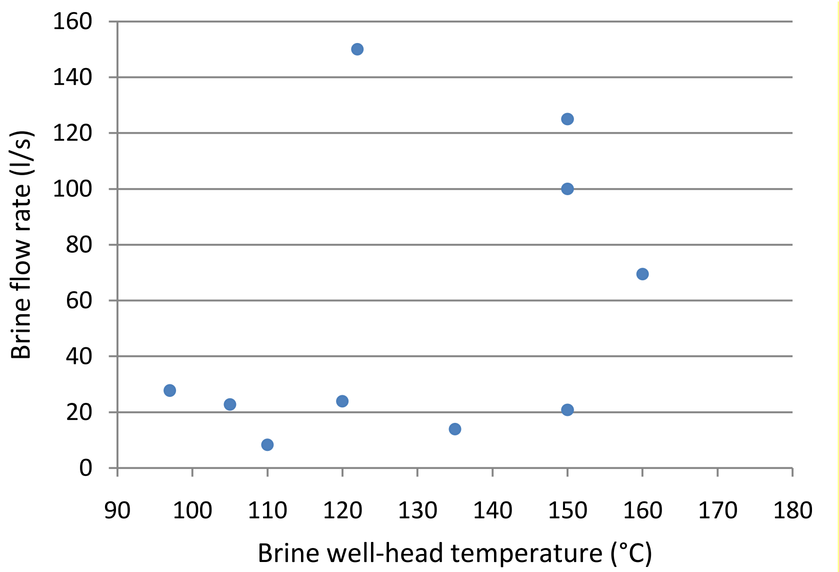

Figure 1.2 Distribution of boundary conditions in chosen geothermal projects in Europe

It has to be mentioned that only a few of the projects listed in

Table 1-2 are considered successful. In some cases where insufficient flow rate was obtained, it was because of failed assessments of the hydraulic parameters of the aquifer. According to Schulz, who performed studies of the probability of success for hydrothermal wells in Molasse Basin in Germany, production from one well should, for economic reasons, exceed 50 l/s. In Germany, for 3,5-4 km deep wells the probability of success (POS) is usually calculated by the investor for flow rates of 65 l/s and 100 l/s and drawdown of 150 m. If performance of the well after stimulation is worse than the minimal POS requirement, the well is considered to be unsuccessful. If only the lower limit is reached, drilling is regarded as partially successful. Projects such as Offenbach, Speyer and particularly Unterhaching, where a volumetric flow rate of 150 l/s was obtained from a single 3350 m deep production well (Knapek, Kittl), have achieved their goals. On the other hand, projects such as Bruchsal with a flow rate lower than 25 l/s are also present in the market. The design of a standardized binary power plant should be based rather on successful projects, because with gained experience their share will be growing. However, in the geothermal industry a high risk of failed drilling will always exist and even in unsuccessful projects, where productivity of a well is low, geothermal water usually has to be utilized in order to minimize financial losses. The Geothermal Energy Association estimated that during the confirmation phase of drilling in the new field, where a geophysical survey has been made, but no wells were drilled yet, the POS of a hydrothermal well is close to 0,6. For wells drilled in the site development phase in well known fields, POS is much higher and averages at 0,9. The designed unit is expected to use water from a single production well. It can be expected that in the future the percentage of successful wells in Europe will be between 60% and 90%.

1.2 Waste heat recovery applications

Heat supplied to an ORC Power Plant does not have to come from geothermal fluid. Such units can be successfully used for generating electric energy from rejected waste heat to the environment in industrial processes and by combustion engines. A survey made by the Energy Information Administration (EIA, Annual Energy Review 2006) shows that the quantity of waste heat available from U.S. industry is bigger than current energy production from all renewable sources combined.

Theoretically, each stream of heat, assuming the temperature of which is high enough and for which no other, more effective application is found, could be used as a source for an ORC unit. However, technical and economical limitations reduce the range of possible applications. A high temperature of rejected stream is usually an advantage, as it equates to greater cost effectiveness and higher potential for heat recovery. It is also required that the flow of fluid carrying waste heat from an industrial process should be concentrated, and fluid composition needs to allow for entry into the heat exchanger.

Most of the waste heat suitable for ORC power plants comes from ovens, furnaces or the exhaust gases of fossil fuel stationary engines.

Table 1-3 gives the temperature ranges of waste heat gases from process equipment (Kaltschmitt et al.).

Table 1-3 Temperature of waste heat gases from engines and industrial processes

Type of device	Temperature of exhaust gasses (°C)
Gas turbine	370-540
Reciprocating engine (naturally-aspirated)	315-600
Reciprocating engine (turbocharged)	230-370
Heat treatment furnace	425-650
Drying and baking ovens	230-600
Annealing furnace cooling systems	425-650

Heat produced in biomass boilers and solar concentrators is not considered a waste, but such applications can be effectively coupled with ORC power plants as well. (Turboden s.r.l.)

Although all foregoing devices are suitable for supplying waste heat to Organic Rankine Cycle power plants, only some of them favor an ORC unit over a traditional steam boiler. These are mainly reciprocating engines and gas turbines because their exhaust gas temperatures are relatively low. Systems working on variable load fall into that category as well.

During the last 4 years there has been significant growth in the gas turbines and diesel engines markets. Moreover, it is expected that in the next decade the share of natural gas-fired combustion engines, reciprocating engines and combined-cycle plants using these devices will capture over 47% of international new-generation market (Diesel & Gas Turbine Worldwide).

Gas turbines are known for their relatively low efficiency, especially under partial load. Simultaneously, open-cycle gas turbines exhaust heat of better quality (higher-grade heat) to the atmosphere than other types of engines. That creates circumstances for bigger efficiency improvement and boosting the power output by using a combined bottoming cycle unit.

Another characteristic feature of gas turbines and diesel engines is their short delivery and commissioning time. Such engines are also relatively easy to install and maintain. It is expected of ORC units working in bottoming cycle to posses the same features. Design of such binary units is not site-specific and, because series production is always more cost-effective, universal design is desired.

The amount of waste heat produced by stationary engines on the world scale is significant. Figure 1.3 shows the total power output of all reciprocating engines and gas turbines for power generation applications sold between June 2005 and May 2007 (Diesel & Gas Turbine Worldwide) as a function of their size. It can be noticed that the biggest share of the market is held by small diesel engines with power output lower than 3,5 MW. The combination of such small engines with bottoming Organic Rankine Cycle is possible and was described e.g. by Invernizzi et al.

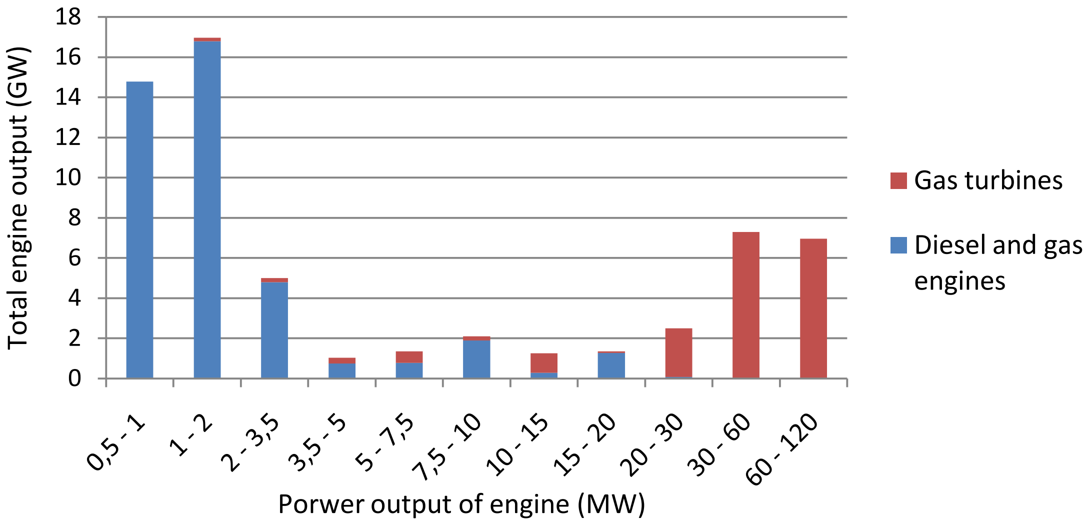

Figure 1.3 Diesel engine and gas turbine orders, June 2005-May 2007

The number of higher power rated units is low compared to small diesel engines. The share of units working on a continuous basis is higher for medium-sized gas turbines and large reciprocating engines compared to small diesel units. This fact however does not change the general picture (Diesel & Gas Turbine Worldwide).

Another application in which binary units can be successfully implemented (Bronicki; Citrin; Legmann) is cement factories. Production of cement is one of the most energy-consuming branches of industry. According to Legmann, even in an optimized process of cement production, significant heat loss occurs due to the rejection of gasses from the clinker cooler. Because of the existence of concentrated, high-temperature exhaust-gas stream of clinker-cooling air (up to 325 °C), from which heat can easily be recovered, many operators of cement plants have installed steam waste-heat boilers coupled with external burning sets. However, because of high variations in exhaust air temperature, which range from 160 to 330°C, such a design often resulted in instability and low efficiency of the unit in partial-load operation. Power plants with organic working fluid proved to operate with more stability in such conditions (Legmann).

2 BASIC CONCEPT OF ENERGY CONVERSION IN ORGANIC RANKINE CYCLE POWER PLANT

The simple Organic Rankine Cycle Power Plant follows the scheme shown in Figure 2.1. The working fluid operates in a sealed, closed-loop cycle. The thermodynamic process undergone by working fluid is shown in Figure 2.2 in a commonly used temperature-entropy diagram and in a pressure-enthalpy diagram.

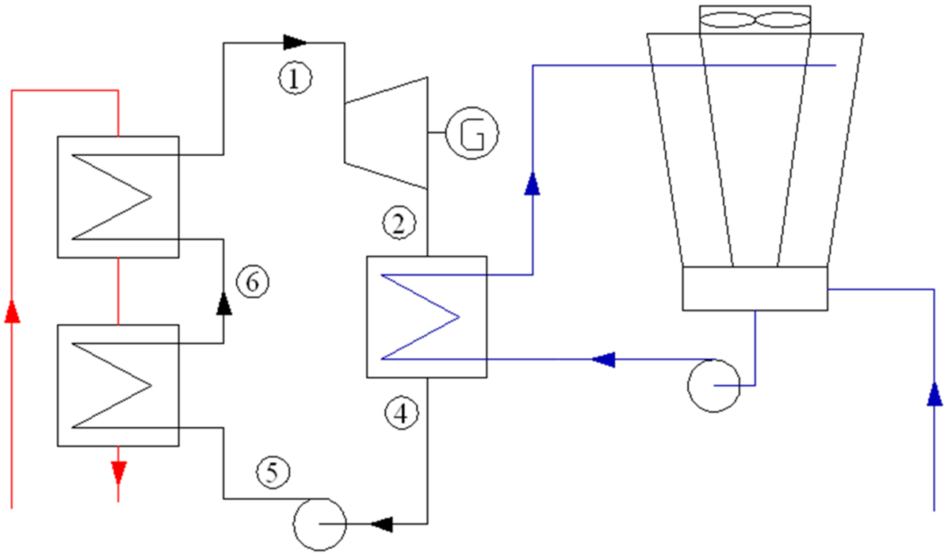

Figure 2.1 Simplified schematic of basic ORC power plant

The stream of geothermal brine - or any other fluid carrying source heat - enters the system through the network of heat exchangers, in which heat is transferred to the working fluid. Typically, there are two stages of heat exchange: one occurring in a preheater, where the temperature of the working fluid is raised to its bubble point, and the other in an evaporator, where the working fluid is vaporized. However, in cases when the fluid is to reach a superheated state, a third heat exchanger- superheater- is added. After isobaric heat addition, which occurs between states 5 and 1, high-pressure vapor is expanded in the turbine (1-2). The exhaust vapor of organic fluid from this process is superheated, which is a result of the characteristic retrograde shape of the working fluid saturation line. This behavior will be described in more detail in chapter 3.3, which discusses the selection of working fluid. The superheated stream of exhaust gasses may be sent directly to the condenser, where it is cooled to temperature T_4 and condensed. However, if economically feasible, exhaust from the turbine may lead to another heat exchanger- regenerator which recovers part of the sensible heat of superheated vapor and transfers it to the stream of liquid working fluid entering the preheater. After leaving the condenser, the working fluid enters the pump, where its pressure is increased to P_5 and returned directly, or through the regenerator, to the preheater.

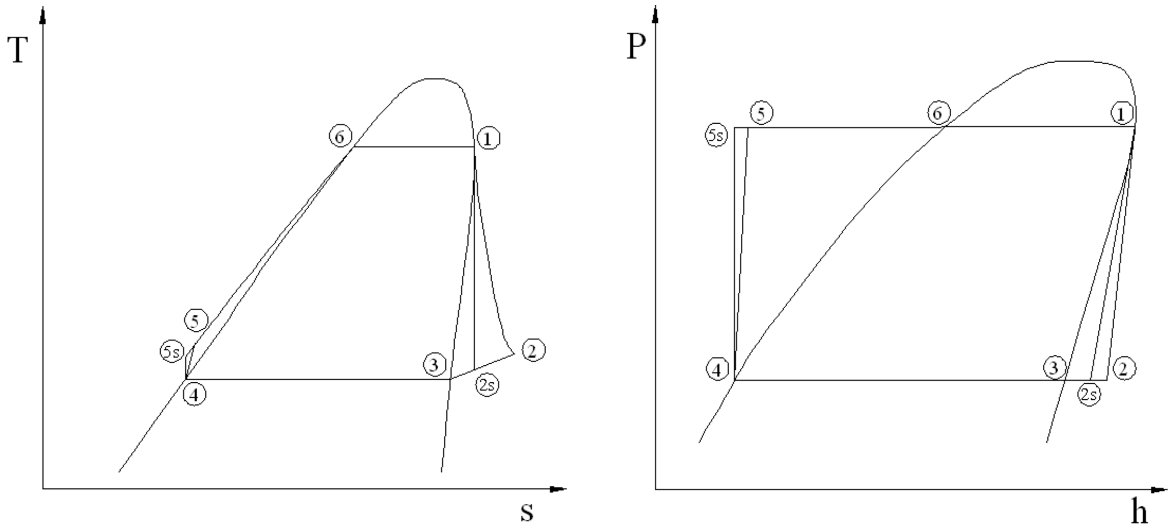

Figure 2.2 Temperature - entropy and pressure - enthalpy diagrams for binary plant

2.1 Preheater and evaporator

The analysis of the preheater and evaporator is straightforward and well described in many engineering books. Standard methodology assumes:

a) Steady state operating conditions
b) No heat losses from heat exchangers not connected with the transfer of mass
c) Pure countercurrent flow in heat exchangers
d) Constant overall heat transfer coefficient
e) Constant specific heat of heat source fluid
f) Changes of kinetic and potential energy of two streams of fluids are negligible

As heat losses in heat exchangers are neglected, the amount of the heat added to the working fluid in time is equal to the heat extracted from the heat source:

$$Q = \dot{m}_{hs}\left(h_{hs,ev,in} - h_{hs,ph,out}\right)$$

$$Q = \dot{m}_{wf}\left(h_{wf,ev,out} - h_{wf,ph,in}\right)$$

Where subscript 'hs' refers to heat source fluid, 'wf' to working fluid, 'ph' to preheater and 'ev' to evaporator.

Such an energy balance can also be made separately for the preheater and evaporator. Since constant heat capacity of heat source fluid is assumed, enthalpy difference may be replaced by temperature difference:

$$\dot{m}_{wf}\left(h_{wf,ph,out} - h_{wf,ph,in}\right) = m_{hs}c_{hs}\left(T_{hs,ph,in} - T_{hs,ph,out}\right)$$

$$\dot{m}_{wf}\left(h_{wf,ev,out} - h_{wf,ev,in}\right) = m_{hs}c_{hs}\left(T_{hs,ev,in} - T_{hs,ev,out}\right)$$

In the temperature-heat exchange diagram of a heat exchanger, the place where the minimum temperature difference between two fluids occurs is called the pinch. The location of the pinch point and value of the pinch point temperature difference is one of the major parameters influencing the performance of ORC Power Plants and will be investigated and optimized in the next chapters.

The amount of heat transferred in the heat exchanger is found from (DiPippo, 2008):

$$\dot{Q}_{HEx} = \overline{U}_{HEx} A_{HEx} LMTD$$

Where:

$\overline{U}_{HEx}$ is overall heat transfer coefficient in $\frac{W}{m^2K}$

A_{HEx} is heat transfer surface in m^2

$LMTD$ is logarithmic-mean-temperature-difference in K

Logarithmic-mean-temperature-difference for any heat exchanger can be calculated as follows:

$$LMTD = \frac{\left(T_{hot,in} - T_{cold,out}\right) - \left(T_{hot,out} - T_{cold,in}\right)}{\ln\left(\frac{T_{hot,in} - T_{cold,out}}{T_{hot,out} - T_{cold,in}}\right)}$$

Where subscript ‘hot’ refers to fluid which has a higher temperature and subscripts ‘in’ and ‘out’ suggest inlet and outlet of heat exchanger.

2.2 Turbine

The purpose of the turbine is to change the potential energy of pressurized gasses into rotational kinetic energy. The stream of high-pressure vapor of organic fluid expands in the turbine, causing its internal part to rotate. The rotor is connected by a shaft to the generator which changes rotational kinetic energy into electricity. The expansion process is considered adiabatic and a steady sate of operation is assumed.

Knowing the isentropic efficiency of the turbine (η_t), which is given by the manufacturer, generated power can be calculated as follows (DiPippo, 2008):

$$\dot{W}_{gen} = \dot{m}_{wf}(h_1 - h_2) = \dot{m}_{wf}\eta_t(h_1 - h_{2s})$$

Where state 2s corresponds to exhaust from ideal isentropic turbine.

2.3 Waste heat rejection system

The heat dissipation system is of great importance for binary power plants because of significantly bigger quantities of rejected heat per unit of electricity output, compared to fossil or nuclear power plants, as well as high sensitivity for temperature variations of the heat sink. The heat dissipated from the cycle is primarily a heat of condensation of working fluid and it can be defined in the terms of the thermal efficiency of the cycle as:

$$q_{rej} = q_{in}(1 - \eta_{th})$$

where q_{in} is heat input to the cycle – in our case the sum of heat rates of preheater and evaporator. η_{th} stands for thermal efficiency of the cycle.

The amount of heat that has to be rejected to the atmosphere per unit of work output is:

$$q_{rej} = w(\frac{1-\eta_{th}}{\eta_{th}})$$

Because of the low thermal efficiency of ORC power plants running on low-quality heat sources, the amount of waste heat per unit of work is approximately 5 to 7 times greater than from the average fossil fuel power plant (Kestin et al.).

On the other hand, the heat dissipation system in binary units has a tremendous effect on the cycle efficiency. The Carnot efficiency term $\eta_{Carnot} = \frac{T_1 - T_0}{T_1}$ shows that the lower the temperature of the heat source becomes, the bigger the effect on cycle performance from the change of heat sink temperature. Therefore, for the Organic Rankine Cycle, which does not reach high temperatures, assuring a low pressure of condensation for the working fluid is of crucial importance.

Several solutions for waste heat rejection systems in ORC power plants are known. Two of them are the most popular among commercially available systems: the wet mechanical-draft cooling tower and the air cooled condenser. Weather conditions and availability of water are crucial parameters determining the choice between these two systems. Water-cooled systems are generally considered less expensive to build and operate as long as makeup water is available and cheap. Although in some arid areas plants using air-cooled condensers may be more cost-effective, their power capacity is highly dependent on weather conditions and their net power output usually fluctuates between 20-25% (Geothermal Energy Association). Unfortunately, power plants equipped with air-cooled condensers reach a higher power output at night, when demand for electricity is lower. In moderate climates, wet cooling towers are preferred as long as a source of cooling water is available. Using a wet cooling tower, working fluid can be cooled down to lower temperatures, which improves the efficiency of the cycle significantly. In most of Europe the availability of water can be assumed, therefore the wet cooling tower was chosen to be the designed system.

A wet cooling tower is an evaporative heat dissipation device in which the cooling of the water is achieved by evaporating part of it. Hot water coming from the condenser is sprayed at the top of cooling tower and while falling, it mixes with a stream of air moving in the other direction. In mechanical-draft cooling towers the movement of the air is induced by one or more mechanically driven fan. Air enters the cooling tower through the inlets located at the sides of the tower and after mixing with water is removed by a fan located at the top of it. To compensate for water losses due to evaporation and blowdown, makeup water has to be supplied to the water cycle. Evaporation water losses are usually in the range of 1-2% of inflow to the cooling tower (Kestin et al.). The quantity of water lost due to blowdown is usually around 20% of the total evaporative loss.

Power consumed by the cooling tower fan is given by

$$\dot{W}_{fan} = \frac{\dot{V}_{air}\Delta p}{\eta_{fan}}$$

Where $\dot{V}_{air}$ stands for volumetric flow of air trough the fan, Δp is the pressure increase over the fan and η_{fan} is the efficiency of the fan and its electric motor.

Heat that is to be rejected to the environment has to change medium before entering the cooling tower, therefore another heat exchanger-condenser has to be used. In this component, heat from the desuperheating and condensing of the working fluid is rejected from the working fluid to the stream of water.

Because heat losses in exchangers are neglected, the amount of heat added to water in time is equal to that which is extracted from working fluid:

$$\dot{m}_{wf}\left(h_{wf,cond,in} - h_{wf,cond,out}\right) = \dot{m}_{cw}\bar{c}_{cw}\left(T_{cw,out} - T_{cw,in}\right)$$

2.4 Pump

Using the same assumptions that were used for the turbine, power consumed by the feedpump can be calculated as:

$$\dot{W}_p = \dot{m}_{wf}(h_5 - h_4) = \dot{m}_{wf}\frac{(h_{5s} - h_4)}{\eta_p}$$

Where η_p is is the isentropic pump efficiency and state 5s corresponds to ideal isentropic pump.

2.5 Net power output

Net power output of the power plant, also known as total power output, can be calculated by subtracting all auxiliary power requirements from gross power output produced by the turbine

$$\dot{W}_{net} = \dot{W}_t - (\dot{W}_p + \dot{W}_{fan} + \dot{W}_{p,cw} + \dot{W}_{p,well})$$

3 DESIGN OF ORC POWER PLANT

3.1 Design boundary conditions

Design boundary conditions for the model of the power plant should be carefully chosen in order to assure the best performance of the unit under its future operating conditions. Following are the factors which affect of performance of ORC power plant in the greatest way and have to be assessed before the design process:

3.1.1 Design temperature of heat source

The chosen design temperature of the heat source fluid at the inlet to heat exchanger is 150°C. It may be considered relatively high for an ORC power plant, but, as it was proved by market research, it is the most commonly found temperature of geofluid among already existing binary power plants in the European Union and it is suitable for waste heat recovery applications. In commercially available units designed for waste heat recovery from biomass-fueled boilers and heat treatment furnaces (Turboden s.r.l.), temperatures of thermal oil, which is used as a heat transporting fluid between the source of the waste heat and the working fluid, are much higher (265-290°C). Such power plants can, however, only operate effectively on a relatively high-quality heat. Their performance in systems working on variable load and exhausting at lower, often variable, temperatures (e.g. in cement plants) would be limited. Moreover due to very different boundary design conditions, it would be impossible to use this unit in geothermal applications. Lower temperatures of thermal oil yield in smaller sized recuperator and higher thermal power is recovered from the exhaust stream.

3.1.2 Mass flow and type of fluid used as a heat source

The mass flow of heat source fluid directly affects the power output of a plant. With all other boundary conditions fixed, optimal power capacity as well as the size of heat exchangers is almost proportional to the mass flow of the fuel. From an economic point of view, if the price of fuel is fixed, in almost all circumstances a high rated power plant is favored over a small unit. That is because the specific cost of each component (i.e. price per key characteristic unit, which for example for heat exchangers is heat transfer area) is dependent on its size. It is usually high for small units and decreases exponentially with the size. This relation is described in the economic analysis chapter by using a scaling exponent.

Because of economic considerations it is suggested that, as long as fuel can be supplied to the power plant with the same specific cost, the price of the product is constant and demand for the product exceeds production, binary power plants should be as big as possible. It is more sensible to build one big unit instead of several small units in order to have equal total power capacity.

If two separate standardized units were to be designed- one for geothermal applications and another for waste heat recovery plants- their optimal size would be much different. Although it is an objective of the investor to declare the minimal flow of geothermal fluid which makes a particular project economically feasible, it may be assumed that a successful hydrothermal production well in European conditions should be able to produce at least 65 l/s of geothermal fluid (Schulz). Theoretically this is the minimum value of flow for which geothermal universal binary power plants should be designed.

However, such an assumption causes two significant problems. First, demand for waste heat recovery binary units of this size is drastically lower than for smaller ones. Preliminary calculations showed that a unit utilizing 65 l/s of 150°C geothermal water should generate approximately 2,3 MW of net power. If exhaust gasses from a reciprocating piston engine or a gas turbine were used as a heat source instead, such engine would have to be rated at 8-10 MW if one want to avoid excessive cooling of exhaust. The order volume of stationary engines of this size is very low, especially if compared to small, up to 3,5 MW units. ORC bottoming power plants suitable for coupling with engine representing popular size category (2,5-3,5 MWe) would be able to generate approximately 600-800 kW net power. Preliminary calculations showed that geothermal plants rated at this power require approximately 18-22 l /s flow of 150°C water.

Another anticipated problem of big-size design is the mismatch between the flow of heat source fluid required by power plant and flow which is available to obtain from a geothermal well. Such a difference in waste heat applications is not harmful as long as the available quantity of low-quality heat is larger than the requirement of the binary unit. Since the stream of ‘waste’ heat was in the past dissipated to the environment and installation of the ORC unit does not increase costs of this stream, the efficiency of the heat recovery is, from the economic point of view , not so important in this case. However, such a discrepancy between supply and demand is very harmful for geothermal power plants, where fuel carries a significant share of the costs. That is why it should be utilized effectively. In geothermal applications, such a mismatch results in off-design operating conditions, which causes an increased cost of the final product – electricity.

A solution to these two problems exists, although it is not a perfect one. It takes advantage of an obvious feature of standardized units i.e. their similarity. Because of identical construction and performance, such units can work in a parallel network, where the flow of the heat source fluid (geothermal water or thermal oil) is distributed equally across several units. Such design provides a chance for a close fit of designed capacity to the available flow. The smaller the elementary unit –the power plant – is, the better the achievable match will be. However, compromise has to be found between the close fit of supply and demand and increased costs caused by the small size of the elementary unit, additional piping etc.

The unit is designed for the flow of 20 l/s of geothermal fluid (which for water at pressurized to 20 bara equals a mass flow of 18,35 kg/s). This size was estimated to optimal for waste heat recovery applications and, thanks to the possibility of in parallel networks, a whole variety of geothermal projects. It may also be a good alternative for individually designed units in geothermal projects where the productivity of the well is low. Among four plants, operating at the temperature of the span of 140-170°C, that are listed in Table 1-2, the flow of geothermal fluid was very low in only one case – it does not exceed 25 l/s. If the drilling phase is successful, coupling of larger number of small units can give a relatively good match between available fluid flow and the power plant´s design capacity.

3.1.3 Temperature of heat sink

If a wet cooling tower is used as a heat dissipation device, the heat sink for the plant is the environment. In many places where ORC power plants are introduced, a chance of using rejected heat for district heating purposes exists. However, if one is to design universal units, such a possibility cannot be assumed. The heat capacity of the environment is considered to be infinite. In order to find the temperature duration curve of the heat sink, ambient dry bulb temperature data from several European cities was collected and assessed (U.S. Department of Energy, Energy Efficiency and Renewable Energy). Figure 3.1 shows wet bulb temperature duration curves for several European cities located in geothermally attractive areas. Three out of the six chosen cities lay in Germany because the majority of new-build geothermal power plants are expected to be located in this country.

Wet bulb temperatures were calculated using EES software as a function of dew point temperature, relative humidity of air and atmospheric pressure. The temperature duration curve used as a design parameter in final EES model was created as an arithmetic average of these six series of data.

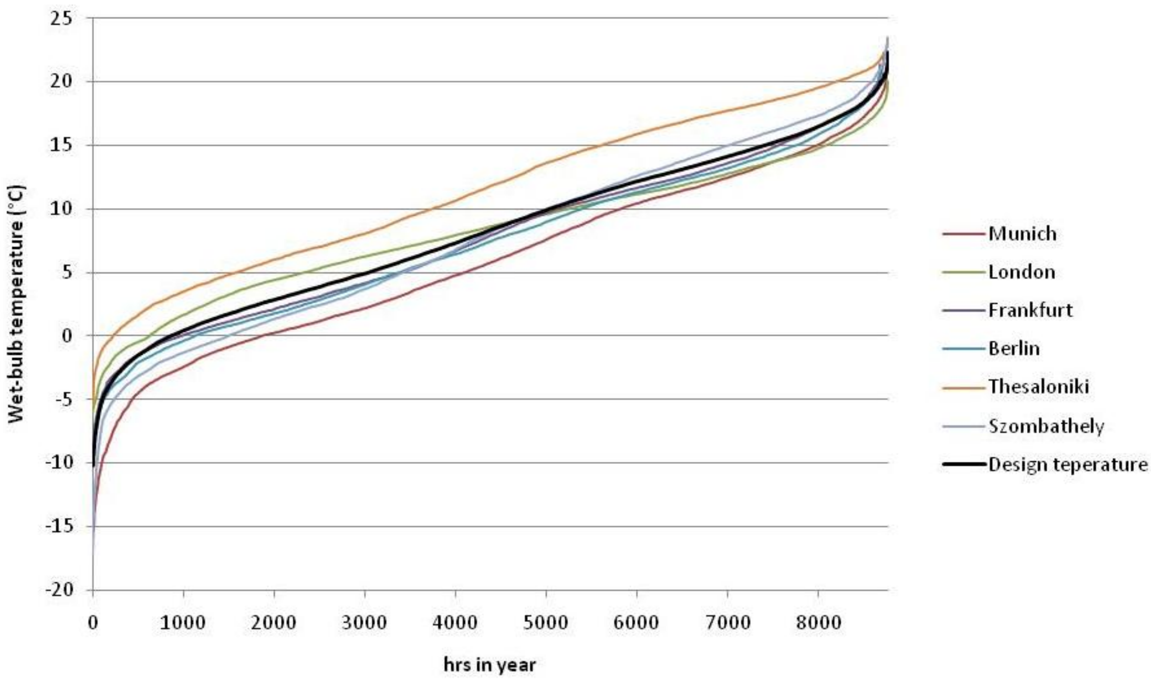

Figure 3.1 Duration curves of wet bulb ambient temperature for chosen European cities

3.2 Assumptions for the model

Remaining assumptions for the model include:

1) Efficiencies (Nughoro)

- Isentropic efficiency of turbine $\eta_t = 0{,}85$
- Isentropic efficiency of working fluid pump $\eta_p = 0{,}65$

- Isentropic efficiency of downhole pump in geothermal power plant $\eta_p = 0{,}65$
- Efficiency of fan in the cooling tower $\eta_{fan} = 0{,}65$
- Efficiency of electric generator $\eta_{gen} = 0{,}97$

2) Design temperatures

- $T_{d,in} = 400°C$ temperature of exhaust gases from diesel engine (Kaltschmitt et al.)
- $T_{c,in} = 275°C$ temperature of exhaust gases from clinker cooler (Citrin)
- $T_{cd,out} = 130°C$ temperature of exhaust gases leaving recuperator

3) Overall heat transfer coefficients (Valdimarsson, P., personal conversation).

- $U_{ev} = 2{,}0 \frac{kW}{m^2K}$ for evaporator
- $U_{ph} = 1{,}1 \frac{kW}{m^2K}$ for preheater
- $U_{reg} = 0{,}7 \frac{kW}{m^2K}$ for regenerator
- $U_{cond,d} = 0{,}7 \frac{kW}{m^2K}$ for condenser in part where working fluid is desuperheated
- $U_{cond,c} = 1{,}6 \frac{kW}{m^2K}$ for condensing process in condenser
- $U_{rec} = 0{,}06 \frac{kW}{m^2K}$ for recuperator in waste heat recovery plants

4) Load factor

- $\lambda_{F,d}$ =70% Load factor of a unit operating on exhaust gases from a diesel engine.
- $\lambda_{F,c} = 73\%$ Load factor of a unit utilizing waste heat from a cement plant (Citrin).
- $\lambda_{F,g}$ =90% Load factor of a geothermal power plant.

5) Pressure drops of working fluid in particular components were assumed to be between 0,1 and 0,4 bar. Additionally, a pressure drop of 0,7 bar occurs between the evaporator and the turbine and 0,8 bar between the regenerator and the preheater.

6) The water level in the borehole is assumed to be at ground level.

7) Rise of air pressure on the fan in the cooling tower $\Delta P_{fan} = 170\ Pa$ (Valdimarsson, P., personal conversation).

8) Rise of the cooling water temperature in the condenser $\Delta T_{cond} = T_{c2} - T_{c1} = 13°C$ (El-Wakil).

9) Cooling water approach of 3°C to ambient wet bulb temperature in the cooling tower (El-Wakil).

10) In order to avoid local freezing of water in cooling tower, the lowest obtainable temperature of cooling water was set as 7°C (Valdimarsson, P., personal conversation).

3.3 Characteristics of working fluids

The proper choice of working fluid in the Organic Rankine Cycle is of key importance as it has a major effect on the performance of the unit. Because of a low temperature of the heat source, irreversibilities occurring in heat exchangers are very harmful to the overall efficiency of the cycle. These inefficiencies are highly dependent on the thermodynamic properties of the working fluid.

A common feature of all organic working fluids used in power plant technologies is their low boiling point. They also have critical temperatures and pressures far lower than water. Judging by thermodynamic characteristics, the most practical candidates are hydrocarbons, refrigerants and fluorocarbons (Kestin et al.). However, when choosing working fluid, factors such as safety, health and environmental impact issues should be considered as well. Because of their strong effect on ozone layer depletion, fluorocarbons were forbidden to be used in such applications.

Thanks to a low critical temperature, some organic working fluids can operate under supercritical conditions in geothermal and waste heat recovery applications. This allows for a better match between the temperatures of the two fluids in heat exchangers, which in turn reduces exergy loss due to irreversibilities. However, because of software limitations, such a possibility will not be analyzed further in this design and only saturated vapor cycles will be taken into consideration.

Table 3-1 lists some of the possible candidate fluids classified as a function of their critical temperature (T_c) and characterized by their other relevant properties: critical pressure (P_c), acentric factor (ω), and molar weight (M). All working fluids included in Table 3-1 are hydrocarbons commonly used in ORC power plants. The range of their critical temperature covers temperatures both below and above the temperature of the heat source. It should be noted that for hydrocarbons, sorted by increasing critical temperature, critical pressure is decreasing and the acentric factor, which accounts for the molecular structure of the fluid, is rising.

Table 3-1 Properties of candidate organic working fluids

Fluid	**Formula**	**T_c (°C)**	**P_c (bar)**	**M (g/mol)**	**ω**
Isobutene	i-C_4H_{10}	134,66	36,29	58,122	0,184
n-butane	C_4H_{10}	151,98	37,96	58,122	0,201
Isopentane	i-C_5H_{12}	187,20	33,78	72,149	0,227
n-pentane	C_5H_{12}	196,55	33,70	72,149	0,251

All these hydrocarbons are of retrograde type, which means that over some temperature ranges the slope of the saturation line is positive. As proved by DiPippo (2008), fluids listed in Table 3-1 behave in that way over the whole range of temperatures typical for working fluids in geothermal binary cycles. This means that adiabatic expansion in the turbine will always have an effect in superheated vapor conditions at the turbine outlet. Invernizzi has shown that the extent of superheating is proportional to molecular complexity σ, defined as:

$$\sigma = \frac{T_C}{R}\left(\frac{\partial s}{\partial T}\right)_{SV,T_r=0,7}$$

Where:

T_C is critical temperature of working fluid (K)

R is gas constant (kJ/kg K)

s is specific entropy (kJ/kg K)

T is temperature

Subscript 'sv' refers to saturated vapor and 'r' to reduced temperature (T_r=T/T_{CR})

Molecular complexity increases with critical temperature and acentric factor and decreases with critical pressure. For homogenous fluids, σ rises with an increased number of atoms in the molecule. Therefore, in the range of considered fluids, the amount of superheat in the fluid at the turbine outlet is highest for n-pentane and lowest for isobutene. Ideally, working fluid should remain dry during the expansion process, but the amount of superheat should be minimal to reduce the cost of the heat dissipation system

All investigated fluids are non-corrosive and chemically stable within the considered temperature ranges.

3.4 Type of cycle

If the thermodynamic state of the fluid leaving the set of heat exchangers is to be considered, one can differentiate between saturated vapor, superheated vapor and supercritical vapor cycle.

The supercritical cycle will not be investigated in this thesis mainly because of software limitations. However, as proved by Kestin et al., such a cycle becomes advantageous compared to the other two when the temperature of the brine exceeds 200°C. It also causes some additional problems such as increased requirements for heat exchangers and piping or increased sensitivity for operating conditions.

The superheated vapor cycle, which is advantageous and commonly implemented in fossil power plants where water is used as a working fluid, also will not be investigated in this thesis. It can be noticed, that in the case of the investigated hydrocarbons, the minor divergence of the lines of constant entropy on the P-h diagram exist. Therefore a significant amount of superheat added to the hydrocarbon working fluid has the effect of a relatively small increase of power output from the turbine. However, minor superheat of the working fluid leaving the evaporator is preferred. It does not improve the thermodynamic performance of a plant, but it lowers the risk of wet expansion at the turbine and eliminates the need of using moisture removers. For the purpose of this thesis 1°C superheat is assumed (Valdimarsson, P., personal conversation).

3.5 Heat exchangers

The type and number of heat exchangers is determined by the type of binary power cycle. For a saturated vapor cycle, components should include a preheater, evaporator, condenser and in some cases also a regenerator. Various types of heat exchangers have been suggested for geothermal power plants, but most of them do not fulfill the requirements of size or resistance for the needed pressure. Plate and panel heat exchangers, although very compact and efficient, are suitable for lower pressures than those found in binary power plants. That makes shell-and-tube type heat exchangers the most viable for use in ORC units (Kestin et al.)

In heat exchangers using geothermal brine, scaling may occur. Thus, for ease of cleaning, geothermal fluid should be on the inner side of the tube.

The type of shell-and-tube heat exchanger is most easily determined by the type of flow (concurrent or countercurrent) and the number of passes, which refers to the number of times the fluid in the tubes passes through the fluid in the shell. In binary units, because of higher efficiency, only countercurrent type heat exchangers are used. From a thermodynamic point of view, a one pass design is preferred because it is the only one that gives pure countercurrent flow. Although it is more expensive to build than the two or four pass types, this type of heat exchanger is assumed in calculations.

In the design of heat exchangers, not only the construction and size, but also the material being used is of a high importance. The most basic and the cheapest kind of steel used for heat exchangers is carbon steel. However, it cannot be used in some cases. In geothermal systems, where the pH of a brine is often highly acidic, carbon steel, with its corrosive nature, is not the right choice. In individually designed units, special corrosion analysis should be made before the material for heat exchangers is chosen. In universal power plants, in order to avoid problems with corrosion, to ensure longer service life and low operating costs, it is advised to choose a more expensive but fouling-resistant material.

Nickel alloys, such as Inconel-625 and Hastelloy C-256, are considered to be highly corrosion-resistant materials (Rafferty). However, because of their high price (Couper et al.), they are used only with very corrosive, high-temperature fluids.

For heat exchangers which handle chemically aggressive geothermal fluid, stainless steel is a reasonable material selection. Stainless steels are much more resistant to uniform corrosion than carbon steel, but some of them have a high potential for pit corrosion and cracking corrosion.

According to Kaya and Hoshan, AISI 300 series stainless steels perform well in systems using geothermal condensate which does not contain dissolved oxygen. Another stainless steel that combines resistance to chloride stress corrosion cracking, crevice corrosion and pitting with good mechanical parameters is 254 SMO. The big advantage of this type of steel is its high yield strength, which is nearly double that of 300 series stainless steels (Metal Suppliers Online). However, it is also much more expensive than AISI 300 series steels and it has increased processing requirements.

Because of economic reasons, the choice of heat exchanger material is practically limited to three types of stainless steels: 304, 316 and 254 SMO. According to Rafferty, steels 304 and 316 are the most suitable choice for heat exchangers used in geothermal district heating systems. The selection between 304 and 316 is then usually based on the temperature and chloride content of the geothermal fluid. In the range of considered sizes, heat exchangers made of this steel are approximately 30% cheaper than those manufactured from steel 316 (Couper et al.). Nonetheless, mainly due to increased molybdenum content, steel 316 has increased resistance to corrosion when compared to type 304.

However, in the considered range of temperatures, even type 316 stainless steel may be not resistant enough to corrosion. Rafferty shows that the concentration of chloride required to produce localized corrosion decreases logarithmically with temperature and, even at 100°C, this figure is lower than in some of the geothermal fields in Europe. Therefore, the use of 254 SMO steel is suggested. According to the manufacturer´s data (Outokumpu Stainless, Inc.) this steel has significantly higher resistance to pitting and crevice corrosion than AISI 300 series steels and is often used as a technically adequate but less costly substitute for nickel-base alloys such as Inconel-625 and Hastelloy C-256.

The best material selection for heat exchangers which do not operate on corrosive fluid is the cheapest one. It was calculated that in range of sizes and pressures considered in this thesis, heat exchangers made of carbon steel are approximately 2,5 times cheaper than those made of AISI 316 stainless steel and over three times cheaper than if they were manufactured from 254 SMO (Couper et al.)

3.6 Thermodynamic optimization of system

Using assumptions listed in this chapter, the model of the ORC Power Plant was created in EES. Its scheme is shown below.

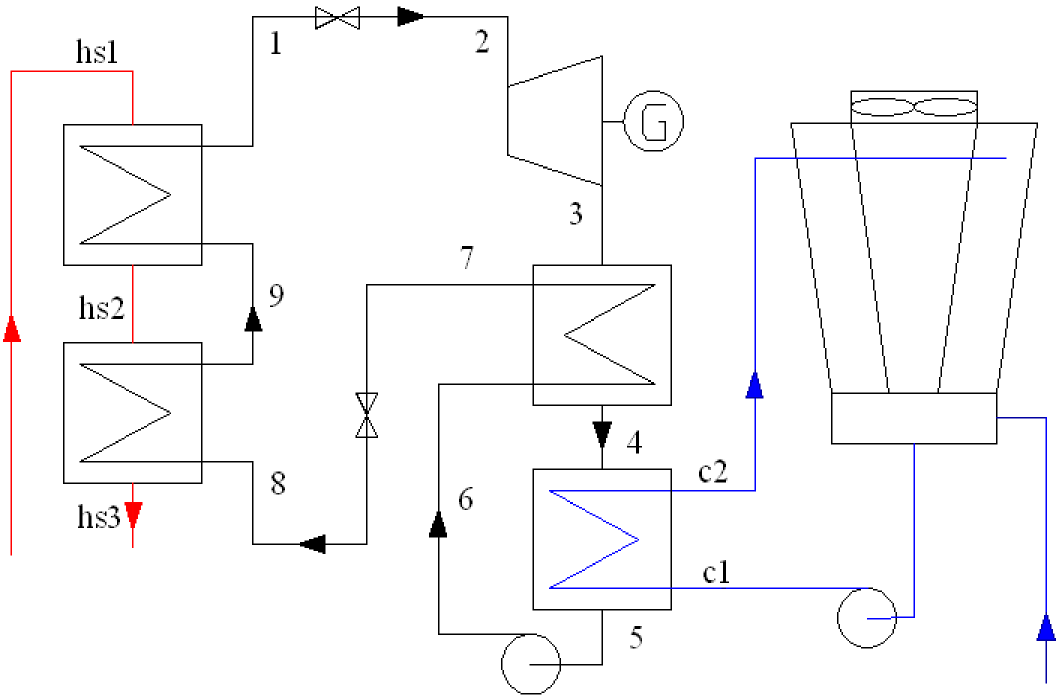

Figure 3.2 Schematic of designed ORC power plant

For each of the working fluids with different thermodynamic properties, different sizes of heat exchangers were chosen. A pinch point temperature difference of 3°C served as the criterion of the design. That was assumed to be a reasonable approximate value for the preliminary design, which would be further optimized by exergy-aided costing.

The evaporating temperature of the working fluid has a crucial effect on the performance of the Organic Rankine Cycle. The term temperature is used here interchangeably with pressure because for saturated liquid, saturated vapor and wet vapor it is equivalent. In this step, the evaporating pressure was chosen in order to maximize the amount of net energy produced from unitary flow of heat source fluid. It was proved by Invernizzi et al. that for working fluids, the critical temperature of which is lower than the temperature of a heat source, the optimal temperature of evaporation is relatively high - close to the critical temperature of the working fluid. That is true in the case of this experiment: as shown in Figure 3.3 isobutene, which has the lowest critical temperature, has a considerably higher (by approximately 5°C) optimal evaporation temperature than the rest of the investigated fluids. Isobutane cools the heat source most effectively, which results in the most efficient heat recovery. However, the amount of heat recovered during the evaporating phase of isobutane is the lowest of all fluids. The share of latent heat rate to total heat exchanged in the preheater and evaporator increases with the critical temperature and molecular complexity and reaches the highest value for pentane.

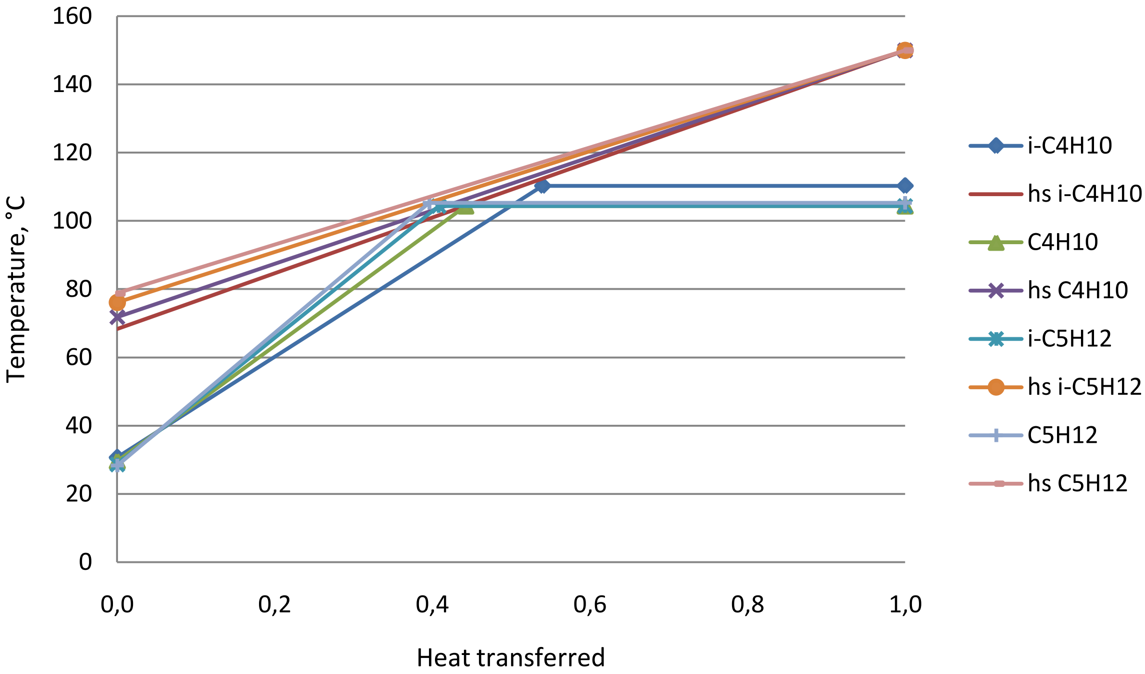

Figure 3.3 Temperature – heat transfer diagram for preheater and evaporator

Table 3-2 lists results of thermodynamic optimization of the Organic Rankine Cycle Power Plant for four working fluids. In this phase of design the lack of a regenerator is assumed. The design's wet bulb temperature is set at 8,4°C.

Table 3-2 Performance of unit without regenerator

working fluid	P_1	T_1	T_{hs3}	$\dot{m}_{wf}$	$\Delta h_{turbine}$	$\dot{W}_{net}$	$\dot{W}_{gen}$	$\dot{W}_{parasitic}$
isobutane	23,5	110,3	68,3	15,18	58,31	708,7	858,9	150,2
n – butane	16,3	104,4	71,8	13,18	61,45	674,5	785,5	111,0
isopentane	7,75	104,4	76,1	12,37	58,12	617,9	697,5	79,6
n – pentane	6,5	105,3	78,8	11,24	61,21	595,9	667,6	71,7

working fluid	$\dot{Q}_{ph+ev}$	$\dot{Q}_{ph}/\dot{Q}_{ph+ev}$	$\dot{Q}_{cond}$	$\dot{Q}_{cond,sh}$	A_{ph+ev}	A_{cond}	$A_{cond,sh}$	VFR
isobutane	6273	0,54	5482	546,8	320	500	83,5	7,05
n − butane	6008	0,44	5257	545,9	270	500	80,2	6,15
isopentane	5679	0,41	4989	751,9	240	450	81,9	6,53
n − pentane	5468	0,40	4801	705,5	230	450	77,7	6,81

The highest power generated by the turbine is obtained when isobutene is used as the working fluid and it decreases for working fluids with higher molecular complexity. The higher requirement for parasitic power of systems using isobutene and n-butane is a result of bigger heat input to the cycle, which affects the power demand of the pump and fan in the cooling tower. Even with these drawbacks, the net power produced by the unit in these two cases is still significantly higher than for isopentane or n-pentane.

The pressure of evaporation is higher for fluids with lower critical temperatures. The pressure that the heat exchanger has to resist usually significantly affects its cost. However, as shown in Kestin et al., higher pressure on the shell side of the heat exchanger does not increase the cost of this component too much unless it exceeds 40-50 bar.

The advantage of cycles using working fluids with higher critical temperature lies mainly in the size of heat exchangers and the effect of regeneration. In the design, the area of the preheater, evaporator and condenser is inversely proportional to LMTD, which in their cases is higher. Therefore, assuming equal values of pinch point temperature difference, the total cost of the heat exchangers will be lower for power plants using e.g. n-pentane compared to those which operate at isobutane.

Foregoing analysis disregarded one component of the cycle – the regenerator – which, although not obligatory, may have a positive effect on the performance of the plant. The regenerator is the heat exchanger located in the cycle between the turbine, condenser, pump and preheater as shown in Figure 3.2. The purpose of using the regenerator is to recover heat form the superheated vapor before it reaches the condenser. This reduces the heat duty of the condenser and at the same time raises the enthalpy of the working fluid leaving the pump. By increasing the temperature of the working fluid at this point, it decreases the heat duty of the preheater, improving the thermodynamic efficiency of the cycle. However, this also leads to a considerable decrease in the recovery efficiency of the cycle since the heat source fluid is not cooled to as low a temperature as in the system without the regenerator.

In the described model isopentane and n-pentane have significantly higher amounts of superheat in fluid exhausted from the turbine compared to isobutane and n-butane. The rate of heat rejected in the condenser from superheated fluid before it reaches the saturated vapor state is higher for these two fluids, even though the total heat duty of the condenser is lower than for isobutane or n-butane. Therefore, as was also described by Invernizzi, the regenerator is more important in cycles using fluids with higher critical temperatures.

To test the effect of regeneration in systems using different working fluids, an equal regenerator with a heat transfer area of 150 m^2 was added to each of them. The area of the remaining heat exchangers was reduced to obtain the same 3°C minimal pinch point temperature difference in each of them. Table 3-3 lists the results of the thermodynamic optimization of this system. The wet bulb temperature is set as 8,4°C.

In each case, the temperature of the heat source fluid at the preheater outlet has raised. The highest increase occurred for n-pentane (9,1°C) and isopentane (8,7°C), compared to 5,4°C for isobutene and 6,3°C for n-butane. Fluids with low critical temperature also encounter the highest rise in total area of heat exchangers after installing a regenerator. For isobutene it is a relatively high value of 12,2%, while in the case of n-pentane this number is only 6%. It can be explained by the different amount of superheat in the fluid entering the regenerator. The addition of an equal regenerator to a system working on isobutene or n-butane cannot result in as much of a reduction of the size of the preheater and condenser as for isopentane because of the smaller heat duty of the regenerator.

Apart from the effect on the cost of the system, the regenerator has one more important drawback. It reduces the efficiency of heat recovery from the heat source fluid by a bigger factor than it can increase thermal efficiency of the cycle. This results in reduced overall efficiency of the plant. Among all four investigated fluids, the smallest reduction of net power output of the unit after the addition of the regenerator occurs for isobutane (1%) and it increases with the critical temperature and molecular complexity of the fluid to reach 3,8% for n-pentane.

For the designed power plant, with isobutene circulating as the working fluid, the limit for the temperature of reinjected geothermal brine is 84°C which corresponds to the temperature of vaporization of isobutene equal to the critical temperature. If geothermal brine cannot be cooled to this temperature because of expected problems with scaling, a different working fluid has to be used.

Table 3-3 Performance of unit with regenerator

working fluid	$\mathbf{P_1}$	$\mathbf{T_1}$	$\mathbf{T_{hs3}}$	$\dot{\mathbf{m}}_{\mathbf{wf}}$	$\mathbf{\Delta h_{turbine}}$	$\dot{\mathbf{W}}_{\mathbf{net}}$	$\dot{\mathbf{W}}_{\mathbf{gen}}$	$\dot{\mathbf{W}}_{\mathbf{parasitic}}$
isobutane	23,5	110,3	73,8	15,22	57,57	705,5	850,1	144,6
n − butane	16,3	104,4	78,1	13,16	60,06	662,1	766,4	104,3
isopentane	7,75	104,4	84,8	12,34	56,57	606,8	677,0	70,2
n − pentane	6,5	105,3	87,9	11,17	59,26	577,8	642,0	64,2

working fluid	$\dot{\mathbf{Q}}_{\mathbf{ph+ev}}$	$\dot{\mathbf{Q}}_{\mathbf{ph}}/\dot{\mathbf{Q}}_{\mathbf{ph+ev}}$	$\dot{\mathbf{Q}}_{\mathbf{cond}}$	$\dot{\mathbf{Q}}_{\mathbf{cond,sh}}$	$\mathbf{A_{ph+ev}}$	$\mathbf{A_{cond}}$	$\mathbf{A_{cond,sh}}$	**VFR**
isobutane	5856	0,51	5067	133,6	320	450	39,8	6,90
n − butane	5525	0,39	4783	97,5	260	420	29,7	5,95
isopentane	5004	0,33	4305	95,1	225	370	28,0	6,22
n − pentane	4768	0,31	4128	76,6	210	350	23,2	6,41

working fluid	**Change in $\dot{\mathbf{W}}_{\mathbf{net}}$ due to regeneration**	**Change in $\mathbf{A_{total}}$**
isobutane	-1,0%	12,2%
n − butane	-2,4%	9,2%
isopentane	-2,9%	8,0%
n − pentane	-3,8%	6,0%

All of the above-mentioned reasons make a regenerator useful, especially in cases where the temperature to which the heat source fluid can be dropped is limited. This restriction, although not important in waste heat applications, is crucial in geothermal power plants. Temperature is the main factor, much more important than pressure, governing water-mineral equilibriums in geothermal fluids. By reducing the temperature of the solution, these equilibriums change, which may result in the scaling of some minerals. Because the chemical composition of geothermal fluid is different in each field, and sometimes even varies significantly between wells located in the same field, temperature limitations for reinjected water should be estimated individually for each project.

Figure 3.4 and Figure 3.5 show the net power produced by the designed power plant as a function of the temperature of the geothermal water at the outlet of the preheater. Variation in the temperature of the reinjected geothermal water was obtained by changing the evaporation pressure of the working fluid. Figure 3.4 presents results for a system with no regenerator while Figure 3.5 shows performance curves for a design in which a regenerator with a heat transfer area of $150m^2$ was added. For the designed power plant, in which isobutene is used as a working fluid, the limit for temperature of reinjected geothermal brine is 81°C without regenerator and 84°C when regenerator is added.

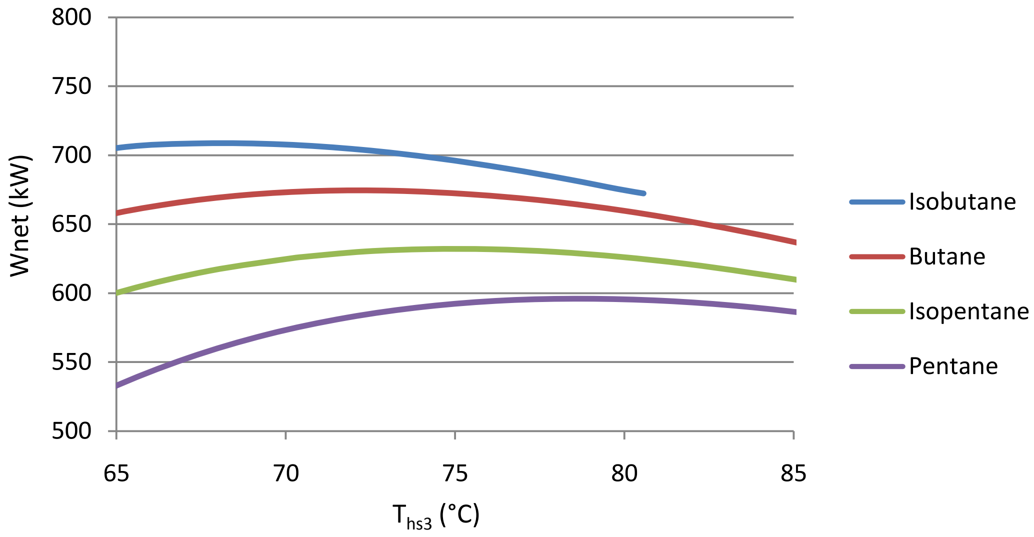

Figure 3.4 Net power output versus temperature of heat source fluid leaving preheater for the unit without regenerator

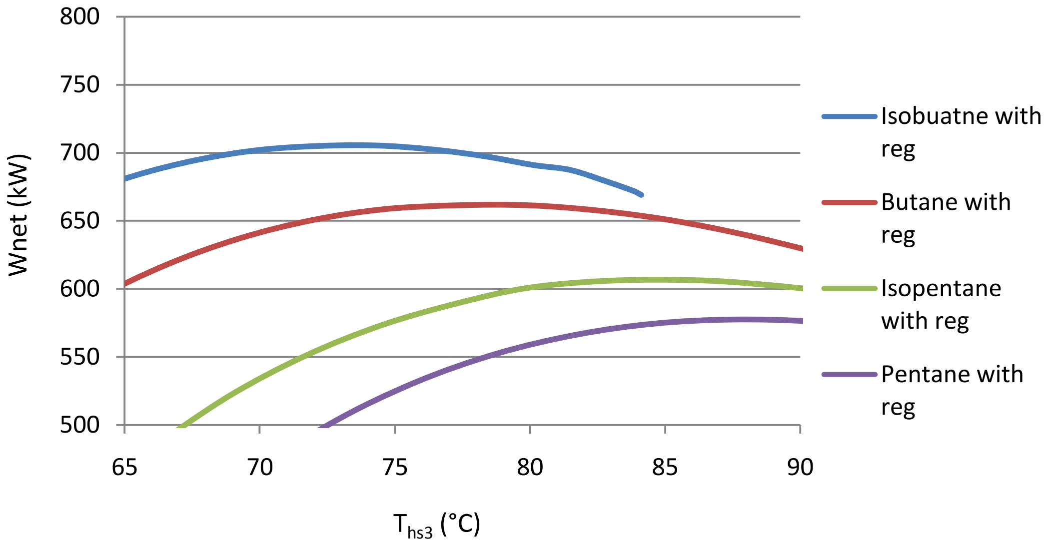

Figure 3.5 Net power output versus temperature of heat source fluid leaving preheater for the unit with regenerator

One of the basic requirements for organic working fluids used in ORC power plants is that the pressure of the working fluid in each phase of the cycle should be higher than the atmospheric pressure. It eliminates the risk of air leakages into the cycle. Such inflows are difficult to notice and very dangerous for power plants. Obviously, the lowest pressure in the Organic Rankine Cycle exists in the condenser.

Efficient utilization of the heat sink is crucial for efficient operation of the power plant. From a thermodynamic point of view, it can be easily shown that, for a Rankine Cycle unit, the reduction of heat sink temperature gives a higher rise in the thermal efficiency of the cycle than an equivalent increase in heat source temperature. Thus, the working fluid should be chosen in a way that, even for the lowest annual temperatures, its condensation pressure exceeds atmospheric pressure. Figure 3.6 shows the pressure in the condenser as a function of wet-bulb ambient temperature. Isobutane and n-butane fulfill the given condition for the whole range of the design's ambient temperatures, but systems using isopentane and n-pentane cannot effectively use low heat sink temperatures. If the condensing pressure is to be kept above atmospheric pressure, annual net electricity generation would be reduced by 5,8% in the case of the isopentane and by 12,7 % for n-pentane without any significant decrease in the cost of the power plant. Such degradation of performance is unacceptable. Therefore these two fluids are not included in further considerations.

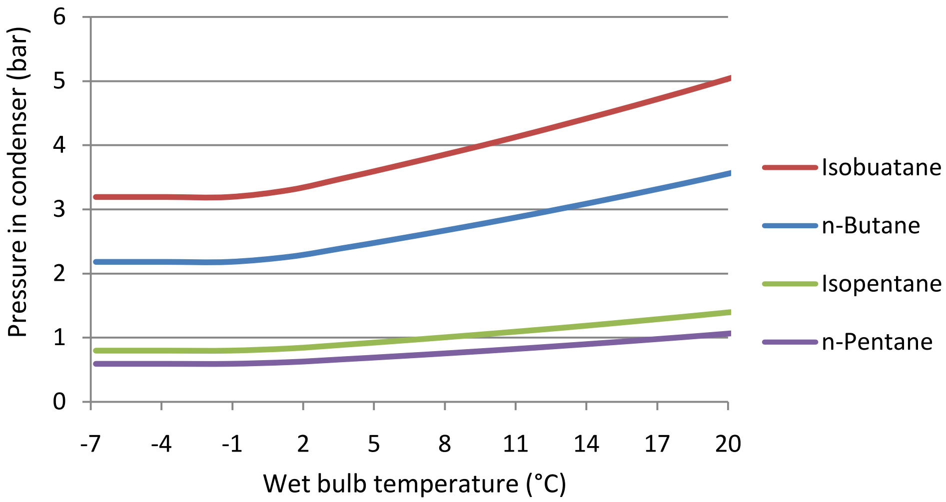

Figure 3.6 Condensing temperature of working fluids for various wet bulb ambient temperatures

Simulation in EES has also proven that at least some of the pre-design assumptions were correct. Values of the isentropic volume flow ratio (VFR) of the turbine, which basically is volumetric flow at the turbine's exhaust divided by the volumetric flow at its inlet, range between 6 and 7 for all four fluids listed in Table 3-1. According to Invernizzi, these are considerably low numbers and the lower the VFR is, the easier it is to design a high-efficiency turbine. Although isentropic efficiency exceeding 0,8 is possible to obtain in a single stage axial turbine with VFR up to about 50, VFR as low as obtained in this case indicates that it is also possible for very small turbines (Invernizzi).

4 EXERGY ANALYSIS

Each form of energy can be described not only by its quantity, but also quality. Quality of energy is defined as the capacity to cause some desired change. To compare the quality of different forms of energy, an universal standard is required. According to Kotas, the most natural and convenient standard is exergy. It is the maximum work obtainable from a given form of energy, when it is brought from its initial state to a dead state by a process in which the stream of energy interacts only with the environment. Therefore, any stream of matter outside the environment, which has at least one of its physical parameters (pressure, temperature or chemical potential) different from those corresponding in the environment, is able to perform work. A state in which the exergy of a system is equal to zero is called a dead state.

Exergy of any stream of matter may be similar to energy divided into components (Kotas):

$$\dot{E} = \dot{E}_k + \dot{E}_p + \dot{E}_{ph} + \dot{E}_0$$

where $\dot{E}_k$ is the rate of kinetic exergy, $\dot{E}_p$ is potential exergy rate, $\dot{E}_{ph}$ stands for the flow of physical exergy and $\dot{E}_0$ is the time rate of chemical exergy

In this analysis it can be assumed, that all streams have negligible kinetic and potential exergy. Moreover, as the stream does not undergo significant change in its chemical composition, chemical exergy at the inlet and outlet of each component is the same. Therefore a change in the exergy of a stream is equal to a change of its physical exergy that is given by (Kotas):

$$e_{ph} = h_1 - h_0 - T_0(s_1 - s_0)$$

Where T_0 is the temperature of environment (dead state temperature) measured in K.

As does the energy balance, an exergy balance also exists for thermal systems. However, there is a major difference between these two. Balance of energy is based on the law of conservation of energy. The equivalent for exergy is a law of degradation of the potential to cause change. Degradation of exergy is caused by irreversibilities of the real thermodynamic processes. Exergy balance for control volume in a steady state is expressed in terms of time rates of exergy transfer and destruction (Bejan):

$$0 = \sum_j \dot{E}_{q,j} - \dot{W}_{cv} + \sum_i \dot{E}_i - \sum_e \dot{E}_e - \dot{E}_D$$

$\dot{E}_i$ accounts for time rate of exergy transfer at the inlet of component and equals

$$\dot{E}_i = \dot{m}_i e_i$$

Similarly, the rate of exergy transfer at the outlet is denoted by $\dot{E}_e$. $\dot{E}_{q,j}$ accounts for exergy transfer due to heat transfer at the boundary of the control element. $\dot{W}_{cv}$ represents the time rate of exergy transfer by work other than flow work, and finally $\dot{E}_D$ stands for exergy destruction rate.

Based on the exergy balance equation, it can be easily proven that exergy flow into some control region is always greater than flow from a control region. Difference, called irreversibility rate, occurs during each thermodynamic process in a power plant.

Irreversibility in the heat exchanger is caused principally by the heat transfer over a finite temperature difference. Although pressure losses, thermal interaction with environment and steamwise conduction in the walls of the heat exchanger also decrease the efficiency of the heat exchanger, only the first cause is taken into consideration in further calculations.

Exergy in the system is lost when a stream carrying exergy leaves the boundary of the system. In a geothermal binary power plant, exergy loss occurs in a preheater, when geothermal fluid is rejected from the plant. A significant amount of exergy is also lost in a cooling tower in the dissipative process.

Table 4-1 shows time rates of exergy in a system with isobutene used as a working fluid, and assumed sizes of heat exchangers equal to:

$A_{cond}=500m^2$

$A_{vap}+A_{ph}=320m^2$

$A_{reg}=50m^2$

Table 4-1 Exergy time rates for a preliminary designed system using isobutane as a working fluid

Type of exergy time rate	**Time rates of exergy (kW)**
Heat source	
Exergy inflow to vaporizer	2073,0
Exergy outflow from vaporizer	1194,0
Exergy loss in preheater	484,0
Working fluid	
Preheater	
Exergy destruction in preheater	116,3
Exergy at inlet of preheater	542,8
Exergy at outlet of preheater	1137,0
Vaporizer	
Exergy destruction in vaporizer	139,7
Exergy at inlet of vaporizer	1137
Exergy at outlet of vaporizer	1876,0
Piping between evaporator and turbine	
Exergy destruction	10,9
Turbine	
Exergy drop in turbine	1027,0
Exergy destruction in turbine	165,6
Work done by turbine	861,2
Exergy at inlet to turbine	1865,0
Exergy at outlet of turbine	837,8

Regenerator at high temperature side	
Exergy drop in regenerator	35,6
Exergy destruction in regenerator	16,8
Condenser	
Exergy drop in condenser	337,3
Exergy destruction in condenser	146,8
Exergy at inlet of condenser	802,2
Exergy at outlet of condenser	464,9
Pump	
Exergy raise in pump	61,1
Exergy destruction in pump	20,4
Work done on pump	81,5
Exergy at inlet of pump	464,9
Exergy at outlet of pump	526,0
Regenerator at low temperature side	
Exergy raise at regenerator	18,9
Exergy at outlet of regenerator	544,9
Piping between regenerator and preheater	
Exergy destruction	2,0
Cooling water	
Condenser	
Exergy raise at condenser	190,7
Exergy at inlet of condenser	13,6
Exergy at outlet of condenser	204,1
Cooling tower	
Work done on cooling tower	25,6
Exergy loss in cooling tower	216,1

Exrgy time rates in the system are also shown in Figure 4.1 in a popular Grassman diagram.

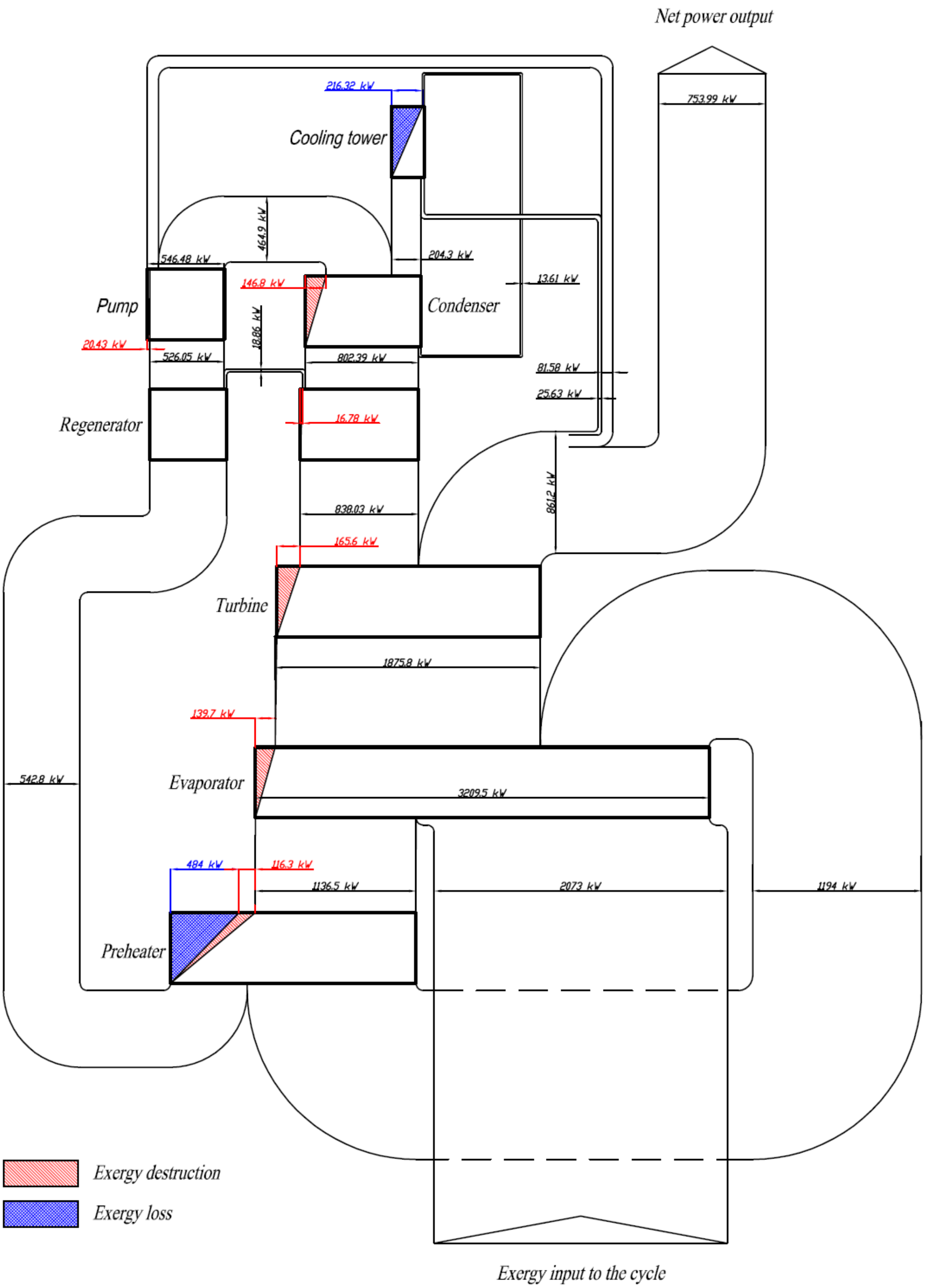

Figure 4.1 Grassman diagram

4.1 Criteria of performance for power plant

4.1.1 Exergetic efficiency

Exergetic efficiency (also referred to as rational efficiency) is a criterion for thermodynamic perfection of a process and can be defined as the ratio of the product of a process to the required input of fuel, both of which are expressed in exergy units. The term ‘product’ represents the desired output of a process and ‘fuel’ refers to the resource that is used to generate this output, which only in some cases is an actual fuel. Rational efficiency shows the percentage of the fuel exergy that is transferred to product exergy. Exergetic efficiency, if calculated not for a single component but for the whole plant, is equal to the Second Law functional efficiency.

$$\varepsilon = \frac{\dot{\mathrm{E}}_{\mathrm{P}}}{\dot{\mathrm{E}}_{\mathrm{F}}}$$

Methods for calculating exergetic efficiency for each component are well described in the literature (Bejan, Kotas), thus showing them in this thesis is not considered necessary. However, one exception has to be made. It was suggested by Bejan, that for heat exchangers for which the purpose is to dissipate waste heat to the environment, the exergy gain of the colder stream should be discarded. That way, the condenser would have no useful output expressible in rates of exergy. Nonetheless, it is important to distinguish between destruction of exergy occurring in the condenser and loss of exergy due to the final destination of heat transferred to the cooling water. Exergy ‘lost’ in the condenser is in fact transferred to the cooling tower and the exergy rate of stream c2 affects the design, performance and economics of this final dissipative device. Exergetic efficiency is not calculated for the cooling tower since it does not have an output expressible in exergy terms.

4.1.2 Efficiency defect

The efficiency defect is a fraction of the exergy input to the process, which is lost trough irreversibility. For k-th process it is denoted by:

$$\delta_k = \frac{\dot{I}}{\sum \Delta \dot{E}_{IN}}$$

For the process occurring in the cooling tower, the rational efficiency cannot be calculated for the same reasons as for the exergetic efficiency (Kotas).

Table 4-2 Values of exergetic efficiency and efficiency defect for particular components

Component	**Exergetic efficiency**	**Efficiency defect**
Preheater	83,8%	0,290
Evaporator	84,1%	0,067
Turbine	83,9%	0,080
Regenerator	53,5%	0,008
Condenser	56,4%	0,071
Pump	69,4%	0,013

Piping (1-2)	99,4%	0,005
Piping (7-8)	99,6%	0,001

4.1.3 First Law assessment of Power Plant

Although less useful than the Second Law assessment, thermal efficiency is still often used for the purpose of comparing the performance of different binary power plants. The standard ideal cycle used in First Law assessment of power plants in general is the Carnot cycle. While still often used for describing the ideal performance of binary plants, it has been proven (Valdimarsson; DiPippo, 2007) to be an unrealistically high limit for these units. The Carnot cycle operates between two ideal heat reservoirs with an infinite heat capacity and, as every ideal cycle, assumes processes to be reversible. Thus, it can be a relatively good reference for fossil fuel power plants working in the Rankine cycle, but not for binary power plants in which the heating medium has a finite heat capacity and it cools down as the heat is transported to the working fluid. DiPippo suggested that appropriate ideal cycle for binary power plants is the triangular cycle, for which the efficiency term is given by:

$$\eta_{th}^{TRI} = \frac{T_H - T_L}{T_H + T_L}$$

Where T_H stands for the temperature of the heat source and T_L represents the condensing temperature of the working fluid.

The maximum efficiency for the triangular cycle operating on a high-temperature heat reservoir can be calculated using dead-state temperature as a reference (DiPippo, 2007):

$$\eta_{th,max}^{TRI} = \frac{T_H - T_0}{T_H + T_0}$$

According to Valdimarsson, the only valid reference for an electric binary power plant which uses the environment as a heat sink is maximum First Law efficiency which can be calculated as:

$$\eta_{I,max} = \frac{\dot{W}_{rev}}{\dot{Q}_{in}} = \frac{\Delta\dot{\mathrm{E}}_{\mathrm{hs}} - \Delta\dot{\mathrm{E}}_{\mathrm{c}}}{\dot{Q}_{in}}$$

Where $\dot{W}_{rev}$ accounts for power produced in a reversible process. It is the difference between the time rate of exergy drop in the heat source fluid occurring in the preheater and evaporator and the time rate of the gain in exergy of the cooling fluid.

4.1.4 Second Law assessment of Power Plant

Since exergy is a true measure of the ability to perform a work, it is very useful to define efficiencies based on this property. These indicators, also referred as Second Law efficiencies, are well described in relevant literature (DiPippo, 2004 and 2008; Valdimarsson; Kanoglu). According to DiPippo, two different approaches for evaluating Second Law efficiency in binary power plants exist.

First, so-called brute-force efficiency, is the ratio of all exergy output streams to all exergy input streams. This is a straightforward attempt but such efficiency does not differentiate between desired streams and streams of exergy losses dissipated to the environment

$$\eta_{II,bf} = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \frac{\dot{E}_{hs3} + \dot{E}_{c2} + \dot{W}_{net}}{\dot{E}_{hs1} + \dot{E}_{c1}}$$

Another type of Second Law efficiency is functional efficiency, which can be described as the ratio of desired output, which, for the designed unit is net electric power generated, to exergy input to a system; which basically is the sum of the time rates of exergy drop of the heat source in the preheater and evaporator. For the saturated vapor cycle:

$$\eta_{II,f} = \frac{\dot{W}_{net}}{\dot{E}_{hs1} - \dot{E}_{hs3}}$$

Where $\dot{E}_{hs1}$ is the exergy rate of the heat source fluid at the inlet of the evaporator and $\dot{E}_{hs3}$ is the exergy rate of heat source fluid at the outlet of preheater.

Table 4-3 shows calculations of First and Second Law efficiencies for the unit described in this chapter. Assumed dead state temperature is 9,5°C

Table 4-3 First and Second Law efficiencies of preliminary designed unit

Efficiency of triangular cycle ($\boldsymbol{\eta_t^{TRI}}$)	16,7%
Maximum efficiency of triangular cycle ($\boldsymbol{\eta_{t,max}^{TRI}}$)	19,9%
Maximum first law efficiency ($\boldsymbol{\eta_{I,max}}$)	19,1%
Brute force efficiency ($\boldsymbol{\eta_{II,bf}}$)	67,4%
Functional efficiency ($\boldsymbol{\eta_{II,f}}$)	45,1%

5 ECONOMIC ANALYSIS

The primary objective of every project is to be profitable. Therefore a proper design for any cost-effective thermal system requires evaluation of the project's cost. Four different analyses were performed for the purpose of this thesis: two for a unit using geothermal water as the source of heat and another two for waste heat recovery scenarios. In terms of geothermal applications it has to be differentiated between the ones using new-drilled wells (in thesis referred to as scenario A) and those utilizing water from abandoned CH-wells (option B). For the waste heat recovery plant, all calculations are performed for the unit coupled with a diesel engine (C) and the unit operating on exhaust gases from the clinker cooler in a cement plant (D).

As it was noticed by Stefansson, analyses of the capital investment as well as operation and maintenance costs (O&M) for geothermal power plants are not frequently published. Moreover, because of the uncertainty connected with the costs of drilling, in some cases the costs of a project can hardly be predicted. However, authorities suggest (Bejan) that exergoeconomic analysis is viable even if the costs estimated in the economic analysis vary slightly from the real ones.

The total cost of production in a geothermal power plant consists of capital investment, which is a one-time cost, and operation and maintenance costs (O&M), which are continuous in nature.

5.1 Estimation of the Total Capital Investment (TCI)

The total Capital Investment of an ORC Power Plant can be shown as (Bejan):

$$TCI = FCI + SUC + WC + LRD + AFUDC$$

Where:

FCI- fixed-capital investment

SUC-startup costs

WC- working capital

LRD- costs of licensing, research and development

AFUDC-allowance for funds used during construction

5.2 Fixed-capital investment (FCI) of on-ground part of power plant

Fixed-capital investment cost is basically the capital needed to purchase and install all needed equipment and build all necessary facilities. It consists of direct and indirect costs. The first type, direct cost, represents all equipment, materials and labor involved in the creation of permanent facilities. Indirect costs, although needed for completion of the project, do not become permanent part of the facilities.

5.2.1 Purchased-equipment cost and effect of size on equipment cost

Type and size of components were roughly estimated in the phase of thermodynamic optimization. A material survey helped to find the best alloys for all heat exchangers. Costs of the heat exchangers were calculated using correlations from Couper et al. Due to the fact that such cost equations, given for AISI 316 steel and Hastelloy are missing for less popular steel 254 SMO, cost calculations are performed for steel 316 and the final cost of components is multiplied by 1,2. A price difference of 20% was estimated based on literature (Rafferty, Couper et al.) and current market prices of products manufactured from these two steels (Metal Suppliers Online). The cost of the cooling tower was calculated for design boundary conditions using the on-line cost estimator available at the Cooling Tower Depot website.

Costs of single components are often given by non-linear equations in which a key characteristic unit of a component is used as a variable. Such an equation is used in this thesis for the estimation of capital investment in heat exchangers (Couper et al.)

$$C_{b,k} = \exp\,[8{,}821 - 0{,}30863(\ln A) + 0{,}0681(\ln A)^2$$

$$150 < A < 12000$$

Where A is heat exchange area in ft^2

Such an equation gives the cost $(C_{b,k})$ of the heat exchanger of a specific design, manufactured from a base material which for this type of component is carbon steel and designed to withstand a defined working pressure. Differences in price between a base design and a specific one are taken into consideration trough cost factors (Bejan):

$$C_k = C_{b,k} f_m f_d f_p$$

Where C_k stands for the cost of component, f_m is the material factor, f_d is the design-type factor and f_p represents the pressure factor.

A scaling exponent was used in order to compensate for differences in the size of the turbine and pump affecting the price of these two components. The meaning of the scaling exponent is shown by the relation (Bejan):

$$C_{PE;Y} = C_{PE;W}\left(\frac{X_Y}{X_W}\right)^{\alpha}$$

Where:

$C_{PE;Y}$ - cost of equipment of size X_Y

$C_{PE;W}$ -cost of equipment of size X_W

α - scaling exponent

Values for the scaling exponent were obtained from Bejan. They are assumed constant in the range in which the component size is optimized.

All prices were adjusted to the reference year 2008 using cost indices (Bureau of Labor Statistics, U.S. Department of Labor). Cost indices are inflation indicators which are crucial in each economic analysis. During the last few years, costs of power plant components increased considerably, mainly due to growing demand for steel. According to the U.S. Department of Labor, if the costs of heat exchangers obtained by using foregoing equations were true in the middle of 2005, these values should be increased by 25,3% in order to get the current prices.

Table 5-1 lists the costs of particular components of the geothermal power plant. It should be noted that among all heat exchangers used in the system, the most expensive are those operating on geothermal fluid. This is caused by the very high price of steel 254 SMO from which they are manufactured. Although necessary in geothermal power plants, the use of such high-quality steel is not required in waste heat recovery systems. In these applications, the preheater and evaporator are not in contact with any corrosive fluid. In order to reduce costs of waste heat recovery units, one may use heat exchangers similar in dimensions and performance, but manufactured from carbon steel. This solution does not require any change in structure or parameters of the other components and results in a significant reduction in PEC.

Table 5-1 PEC of geothermal power plant

Component	**Price (in $2008)**	**Key characteristic (unit)**	**Price per unit of key characteristic (in $2008)**	**Scaling exponent**
Preheater	159.887	Area(m^2)	726,8	-
Evaporator	75.135	Area(m^2)	767,9	-
Turbine	644.205	W_{gen}(kW)	750,0	0,6
Regenerator	11.375	Area(m^2)	227,5	-
Condenser	78.404	Area(m^2)	156,8	-
Pump	44.482	W_p(kW)	500,0	0,48
Cooling tower	154.176	Q_{ct} (kW)	30,0	0,6
Total PEC	**1.167.664**			

Table *5-2* contains the costs of components used in a waste heat recovery ORC power plant. It can be noticed, that due to the change of steel used for the preheater and evaporator, prices of these components dropped significantly. If manufactured from cheap carbon steel, they cost only 32% of equivalent heat exchangers made from the stainless steel 254 SMO. Due to this change, the PEC of the heat recovery plant decreased by approximately 17% compared to the geothermal unit.

Table 5-2 PEC of waste heat recovery plant

Component	**Price (in $2008)**	**Key characteristic (unit)**	**Price per unit of key characteristic (in $2008)**	**Scaling exponent**
Preheater	49.859	Area(m^2)	226,6	-
Evaporator	25.235	Area(m^2)	257,9	-
Turbine	644.205	W_{gen}(kW)	750,0	0,6
Regenerator	11.375	Area(m^2)	227,5	-
Condenser	78.404	Area(m^2)	156,8	-

Pump	44.482	W_p(kW)	500,0	0,48
Cooling tower	154.176	Q_{ct} (kW)	30,0	0,6
Recuperator (C)	124.430	Area(m^2)	158,6	-
Recuperator (D)	194.905	Area(m^2)	164,1	-
Total PEC (C)	**1.132.166**			
Total PEC (D)	**1.202.641**			

5.2.2 Remaining FCI costs

Other FCI costs are estimated using a factor method (Bejan). It calculates the expenditures for TCI as a linear function of the purchased-equipment-cost. A breakdown of total capital investment (TCI) is shown in Table 5-3.

Piping

The cost of piping in the power plants is usually in the range of 10-70% of the purchased-equipment cost (PEC). Although it is suggested that for plants handling fluid and with heat regenerators the higher numbers in this range apply, for geothermal power plants the relative cost of piping is much lower. It is caused by lower piping diameters and the high cost of other components in binary units. The cost of pipes in this range (5-75cm) is almost linearly dependent on their diameter (Bejan), therefore the total cost of piping is assumed to be equal to 7% of PEC for geothermal applications and 9% of PEC for waste heat recovery systems.

Installation of equipment

This cost accounts for transportation of equipment from the factory, insurance, costs of labor, foundations, insulation, cost of working fluid, thermal oil for the waste heat recovery loop and all other expenses related to the erection of a power plant. Installation costs of fossil fuel power plants account usually for 20-90% of the purchased equipment costs. However, the designed ORC power plant is assembled into one unit when it is manufactured and its transportation is relatively cheap and easy because of its small size. On average, the transportation cost alone accounts only for 5% of PEC for power plants in general. For these calculations, a value of 6% of PEC is assumed for this category.

Instrumentation, controls and electrical equipment

For instrumentation and controls a cost of 5% of PEC is assumed. The cost of electrical equipment, which for power plants usually includes distribution lines, emergency power supplies and switch gears is relatively low for the designed plant. It is however variable depending on the place of installation. Assuming the close proximity of the electric grid this cost is set at 4% of PEC.

Cost of land

Direct costs also include so-called offsite costs, i.e. cost of land, civil and structural work. The cost of land is highly site-dependent. It is suggested (Bejan) that if land is to be purchased, cost may contribute up to 10% of PEC. However, specific land use for a binary power plant is significantly lower than for any other type of power plant. A medium-sized geothermal binary power plant needs on average 1415m^2/MW of peak power while higher-rated nuclear power needs about 10000m^2/MW (DiPippo, lectures). Therefore the cost of land may be assumed negligible compared to other costs.

Civil and structural work

This category includes the costs of all needed roads, buildings etc. These costs are site specific and vary significantly between different projects. For waste heat recovery systems, where part of the required infrastructure already exists, these costs are relatively low and assumed to be 3% of PEC. For geothermal projects, although in reality they are still highly variable, these costs are fixed in this thesis and equal to 7% of PEC.

Engineering and supervision

This category of costs includes the cost of planning and the design of the power plant as well as the manufacturer's profit, the engineering supervisor, inspection and administration. Bejan suggests that for power plants in general, these costs range between 25 and 75% of PEC. Data for German geothermal binary power plants (Kranz) gives an average value of design and planning costs of about 20% of the power plant's cost. Such an expensive design phase of traditionally designed binary power plants is the biggest advantage of a standardized unit. Since the ORC power plant described in this thesis is preassembled by manufacturer it can be assumed that the only cost of design and planning is the manufacturer's profit. Costs of supervision should also be lower as the time of construction is much shorter compared to traditionally designed plants and it does not require such a highly specialized staff. All these factors make the final cost of engineering and supervision to be set at the relatively low level of 6% of PEC.

Construction

Expenses for construction include all the costs of temporary facilities and contractor's profits. This is yet another phase in which universally designed units hold an advantage over those which are individually designed. Usually for power plants these costs account for about 15% of direct costs (Bejan). However, the placement of an already assembled and frame-mounted unit into its place of operation is much cheaper than assembly on the spot. In this thesis construction is assumed to account for 3% of direct costs.

Contingencies

This cost should compensate for all unpredictable events which may occur during transportation, construction and erection of the power plant. The contingency factor usually ranges between 5 and 20% of FCI and is dependent on the level of complexity and uniqueness of the power plant. Since the risk of unpredictable events is low due to universal design, the contingency factor is smaller than for an average power plant but it still has to be taken into consideration. It is assumed to equal 3% of FCI.

5.2.3 Other outlays

Other outlays consist of the startup costs, working capital and allowance for funds during construction (Bejan).

Startup costs

Startup costs are the expenses that have to be spent after the construction of the power plant but before the unit can operate at a full load. They have to cover not only the cost of equipment and work during startup time, but mainly the difference in income which is the result of a partial load during this time. Some sources show a comprehensive and detailed approach to estimating these costs on the basis of the cost of fuel (Bejan). However, such methodology does not apply in the geothermal industry since the investment in fuel is done before the erection of the power plant. Finally, a contribution of 1% of FCI was assumed for geothermal and waste heat applications.

Working capital

Working capital is the amount of money needed to cover the costs of power plant operation before receiving payment for electricity sold to the grid. According to Bejan, working capital for power plants is calculated as the sum of costs representing two months of fuel at full load and three months of labor. The first of these costs may obviously be neglected in a heat recovery system as well as a geothermal power plant, where the investment in fuel is made before the plant starts to operate. The unit should also work without permanent supervision; therefore labor costs are relatively low. Finally, working capital is assumed to be very low and set at 3% of PEC.

Allowance for funds used during construction (AFUDC)

With this category of capital investment costs comes another advantage of universally designed power plants. In site-specific design, the construction period is relatively long. During this time various amounts of money have to be invested in design studies, on-site engineering work, construction of the power plant, etc. An allowance of funds used during construction time compensates for different time values of money and is based on the interest rate. In the investigated unit, foregoing costs are relatively low since construction time is short. Thus AFUDC is ignored in this thesis.

Table 5-3 Total capital investment (TCI), values in US$(2008)

Fixed capital investment (FCI)		**Geothermal (A,B)**	**Waste heat C (Diesel engine)**	**Waste heat D (Cement plant)**
I.	***Direct costs***			
1.	Onsite costs			
•	Purchased-equipment cost	1.168.000	1.132.000	1.203.000
•	Piping	81.738	101.896	108.238
•	Installation of equipment	70.062	67.930	72.159
•	Instrumentation and controls	58.385	56.609	60.132
•	Electrical equipment	46.708	45.287	48.106
•	Total onsite costs	1.424.893	1.403.722	1.491.635
2.	Offsite costs			
•	Civil and structural work	81.738	33.965	36.079
	Total direct costs	1.506.631	1.437.687	1.527.714
II.	***Indirect costs***			
•	Engineering and supervision	70.062	67.930	72.159

•	Construction costs	45.190	43.136	45.821
•	Contingency	48.647	46.468	49.360
	Total indirect cost	163.899	157.534	167.340
	Total fixed capital investment (FCI)	1.670.530	1.595.221	1.695.054
III.	**Other outlays**			
•	Startup costs	16.702	15.954	16.947
•	Working capital	35.031	33.965	36.079
TCI of power plant		**1.722.263**	**1.645.140**	**1.748.080**

5.3 Cost of geothermal wells

5.3.1 New drilled wells

For ORC power plants running on waste heat, foregoing estimation gives the final value of the total capital investment. However, the amount of money which has to be invested in a geothermal power plant before its operation starts is considerably higher. For a geothermal binary plant utilizing low-temperature water as a heat source it is sometimes not surface equipment, but drilling that has the highest share in total investment cost. It was estimated by Stefansson that for high temperature fields, the drilling cost is typically in the range of 20-50% of TCI. For low temperature fields such estimations found in literature vary significantly. The majority of reports say that 17-47% of the total project cost is allocated to drilling (Geothermal Energy Association). However, some research projects estimate that for binary power plants installed in Europe this share is higher. Kranz suggests that in such low-temperature projects even 70% percent of TCI can be allocated to drilling and site development.

Even though, as it was mentioned in beginning of this chapter, the cost of drilling is site-specific and varies drastically for different projects, some average values can be estimated.

Remaining on-site investments in a geothermal power plant can be split into three phases: exploration, confirmation and development of the site. Exploration is the initial development phase and includes prospecting and field analyses aiming to locate a productive geothermal reservoir. Cost estimates for this stage of development found in the literature are inconsistent. GeothermEx (2004) assesses it to range between \$14/kWe and \$263/kWe. The cost reported by ORMAT (Geothermal Energy Association), which is equal to \$250/kWe, may be considered as the best estimate for the purpose of this thesis.

The confirmation phase consists of the drilling of production wells until approximately 25% of the needed resource capacity is confirmed. Cost of the confirmation phase for commercially viable projects should average around \$150/kWe.

The most expensive of these three phases is the last one: site development, in which, for large-scale projects, approximately 75% of required brine flow is obtained. Sensible cost estimates for drilling are very difficult to provide. The EPRI suggests (Geothermal Energy Association) that average drilling costs for binary power plants account for \$996/kWe. Based on interviews with geothermal developers, GEA assessed total drilling costs which include confirmation phase to range between $600 (2006)/kW and $1200 (2006)/kW with the average close to $1000 (2006)/kWe.

Based on these reports, the total cost of exploration, confirmation and site development in this thesis is assumed to be equal to $47250 (2006) per l/s of geothermal fluid. This is equivalent to $1100 (2006)/kW of energy generated by power plant described in exergy analysis chapter.

In order to bring this cost into perspective for the year 2008, the producer cost index for the drilling of oil and gas wells was used (Bureau of Labor Statistics, U.S. Department of Labor). That raised the cost of drilling by 39%, to $65693 per l/s of geothermal fluid equivalent to $1529 per kW of energy generated by the preliminary designed plant.

According to the Geothermal Energy Association it takes a minimum of 3 to 5 years to put a geothermal power plant on line, therefore it is assumed that capital investment in a geothermal power plant occurs during the period of 3 years, where in the first two years 20% of the overall cost is invested in exploration and confirmation of resources and 80% of the total site development cost is invested in the third year

5.3.2 Abandoned CH-wells

All sources agree that reopening some old oil and gas wells is more cost-effective than drilling new ones (Árpási, Holl et al.) The largest uncertainty in such projects is connected to non-documented materials that remain in the well. However, the cost of recompletion of existing CH wells is estimated to be 10-15% of a new well. In this thesis the higher value (15%) is assumed.

5.4 Operation and maintenance (O&M) costs

Operation and maintenance costs consist of all expenses accrued during the operation phase of the power plant. They encompass expenses related to labor, chemicals, spare parts, etc. In geothermal power plants these costs are usually very low compared to the initial investment. If waste heat recovery applications are considered, this figure may be even lower. ORMAT claims that the costs of operation and maintenance of one of their units working in a cement plant were $0,2/MWh on average during 30 months of operation (Legmann).

Using the methodology presented by Kaltschmitt et al., annual maintenance costs are estimated for each component individually as a fraction of the component purchase cost:

Table 5-4 Annual operation and maintenance costs

Component	Annual O&M cost (fraction of initial cost)
Recuperator in waste heat recovery unit	10%
Preheater and evaporator in geothermal power plant	9%
Other heat exchangers	4%
Cooling tower	5%
Turbine and working fluid pump	2%

Operation and maintenance costs of the preheater and evaporator in geothermal applications are significantly higher because of the risk of scaling and corrosion. Maintenance requirements of the recuperator are also higher than for the remaining heat exchangers due to deposition of particulate matter carried by streams of exhaust gasses.

5.5 Time value of money

It is commonly known that the value of money changes in time and that typically the same amount of money now is worth more than it will be in the future. Because economic analysis of the project requires comparison of many flows of money occurring in different points in time, a method of converting these flows into an equivalent constant quantity has to be used. Such a concept is called levelization and is used in this thesis to calculate costs of operation, maintenance and levelized total cost of electricity. Costs of fuel and capital investment are also expressed in the form of annuities.

To account for change in the value of money over time, the effective rate of return (i_{eff}) is used. If computing occurs only once a year it is defined as:

$$F = P\left(1 + i_{eff}\right)^{n}$$

Where P and F represent present and future value of money and n is the number of years between these two.

Annuity is a series of equal cash flows occurring during some period of time. To determine annuity (A) taking place once a year during n years from the present value P, the capital recovery factor is used (Bejan):

$$A = CRF \cdot P = \frac{i_{eff}(1 + i_{eff})^{n}}{(1 + i_{eff})^{n} - 1} P$$

Without any doubt the biggest investment in an Organic Rankine Cycle power plant occurs before its operation starts. In a geothermal power plant, if the flow from wells is maintained during the lifetime of the unit and no additional wells have to be drilled fuel costs can be simply calculated as an annuity of the initial cost of drilling. However, this methodology does not apply to O&M costs rates. Due to inflation, changes in the market situation, technological advances etc., these costs vary in time. This phenomenon is known as escalation and in engineering economics it can be expressed using the nominal escalation rate (r_n). In order to transform a nonlinear series of O&M costs into an equivalent series of annuities called levelized values, the constant-escalation levelization factor (CELF) is used (Bejan):

$$A = CELF \cdot P_0 = \frac{k(1 - k^{n})}{1 - k} CRF \cdot P_0$$

$$k = \frac{1 + r_n}{1 + i_{eff}}$$

Where P_0 represents expenditure for O&M in the first year.

5.6 Financing mechanisms

The cost of electricity, which is the final product of economic analysis, is affected not only by capital investment, but also by the origin of these funds. The cost of borrowing the money may vary significantly depending on the type of the project, the situation in financial markets and other conditions.

The majority of projects are financed from two sources: debt and equity. The cost of debt is lower, however in the case of project failure the debt provider is usually the first one to get its money back. Moreover it is usually expected by the bank to secure the debt with some share of money from equity. This one, however, is more expensive. According to the Geothermal Energy Association, in geothermal projects usually 30% of financing comes from equity and 70% from debt. Typical interest rates in the geothermal industry are around 6-8% for debt and approximately 17% for equity.

Both of these interest rates are highly influenced by the estimated risk of failure of the project. Investors usually compensate for increased danger by raising interest rates. Geothermal plants, because of the uncertainty of well productivity and chemistry of geothermal fluid, are considered by banks more risky investments than waste heat recovery plants. Therefore, it is expected that the effective rate of return is also higher for geothermal projects.

Based on the report of the Geothermal Energy Association, the effective rate of return was estimated to equal 10% for geothermal plants and 8,5% for waste heat recovery projects.

The lifetime of a binary power plant is relatively long. In various sources it is estimated to average between 30 and 35 years. However, one has to differentiate between this actual expected time of operation and its financial lifetime, which is much shorter. Investors' expectations concerning the money return period are also different for each type of project and originate in the different nature of developers. Corporations investing in geothermal power plants are usually energy utilities which do not need a very fast return of their investment. On the other hand, companies interested in ORC waste heat recovery systems require much faster repay. Electricity is simply a by-product for them, and an ORC power plant is one of few options for improving their financial statistics. Consequently, the lifetime of the designed plant is assumed to be 15 years for the geothermal version and 10 years for the waste heat recovery system. Table 5-5 summarizes economic assumptions made in this chapter.

Table 5-5 Economic variables

	Geothermal power plant	**Waste heat recovery plant**
Effective rate of return (i_{eff})	10%	8,5%
Nominal escalation rate (r_n)	4%	4%
Lifetime of power plant	15 years	10 years

5.7 Results of economic analysis

The results of the economic analysis are shown in Table 5-6 and Table 5-7. Table 5-6 contains levelized annual costs in US $(2008). It is shown that the biggest share in the annual levelized cost has the capital investment in the on-ground part of the power plant. Although TCI of the unit was almost the same in all four analyzed cases, the levelized value of this cost is considerably higher for waste heat scenarios. That is caused by the potentially different economic lifetimes of the power plant (15 years for geothermal compared to 10 years for waste heat recovery plant), which determine the capital recovery factor and by that, the levelized cost of capital investment. Levelized costs of operation and maintenance are the lowest for units coupled with a diesel engine. This is an effect of the relatively small recuperator and lower maintenance requirements compared to geothermal system due to a lower risk of scaling and corrosion in the preheater and evaporator.

Table 5-6 Levelized annual cost

	Geothermal A ($/yr)	**Geothermal B ($/yr)**	**Waste heat C ($/yr)**	**Waste heat D ($/yr)**
Levelized TCI	226.478	226.478	250.719	266.407
Levelized cost of fuel	178.094	26.714	0	0
Levelized O&M costs	59.928	59.928	49.285	57.856
Levelized cost of product	464.500	313.120	300.004	324.263

Table 5-7 contains the final levelized cost in U.S. ¢(2008) per kWh of electricity produced for each of the four different applications. Values of annual cash flows shown in Table 5-6 could suggest that, on average, the electricity produced from waste heat is much cheaper than from a geothermal plant. This would be true if all units had the same load factor. However, the lower load factor of waste heat recovery plants increases the relative cost of electricity produced by them. Judging by the cost of the final product it places the waste heat recovery plant between the most expensive geothermal plant using new-drilled well (A) and the cheapest option - scenario B.

Table 5-7 Levelized cost per unit of net power output

	Geothermal A (¢/kWh)	**Geothermal B (¢/kWh)**	**Waste heat C (¢/kWh)**	**Waste heat D (¢/kWh)**
Levelized TCI	3,67	3,67	5,14	5,23
Levelized cost of fuel	2,92	0,46	0,00	0,00
Levelized O&M costs	0,97	0,97	1,01	1,14
Levelized cost of product	7,56	5,09	6,16	6,37

Values presented in Table 5-7 are calculated for the preliminary designed unit and will be used as a reference for results obtained through thermoeconomic optimization.

6 THERMOECONOMIC ANALYSIS AND EVALUATION

6.1 Exergy costing

The purpose of thermoeconomics is to combine exergy analysis with economic considerations in order to find the most cost-effective design for the system. Exergy costing is an approach basing on exergy as the only meaningful thermodynamic value to which costs may be assigned. In the chapter dealing with exergy analysis, all exergy flow rates in the system were calculated - including losses of exergy to the environment and exergy destruction rates. In this chapter, costs are assigned to these streams in order to identify cost-ineffective processes and show the way the system may and should be changed to assure the best economic performance.

Thermoeconomic evaluation requires streams of costs to be calculated as levelized values. Costs per unit of exergy are denoted with c and have unit of U.S. \$(2008)/kJ. Exergy time rates are represented by $\dot{E}$ and calculated in U.S. \$(2008)/s. The cost rate of a stream j is calculated by multiplying the exergy rate of this stream by the cost per unit of exergy of this stream.

$$C_j = c_j \dot{E}_j = c_j e_j \dot{m}_j$$

In engineering economics the cost balance of the system, as well as for each component separately, may be formulated (Bejan):

$$\dot{C}_P = \dot{C}_F + \dot{Z}$$

$$\dot{Z} = \dot{Z}^{CI} + \dot{Z}^{OM}$$

Where $\dot{C}_P$ and $\dot{C}_F$ are the cost rates of product and fuel. Cost rates associated with the capital investment and operation and maintenance of the power plant are denoted by $\dot{Z}^{CI}$ and $\dot{Z}^{OM}$.

Exergy costing usually involves cost balances for each component. Such a balance for k-th component with n entering streams and m exiting streams, operating in steady state conditions, can be represented as:

$$\sum_m \dot{C}_{e,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_n \dot{C}_{i,k} + \dot{Z}_k$$

Where $\dot{C}_{w,k}$ is the cost rate of work and $\dot{C}_{q,k}$ equals cost rate of heat transfer from the component. Cost rates are expressed in U.S. \$(2008) per second.

In this paper, cost balances are applied separately to each component. According to Bejan it is a better solution than considering group of components together, even if cost assumptions for some of individual components are slightly different than the real values.

In the following analysis, costs per exergy unit for entering stream are known. The only one different from zero is the cost of geothermal brine i.e. the levelized value of drilling and site development costs calculated per second of operation time. Based on exergy flow rates, distribution of costs in the system can be easily calculated. However, for components which have more than one exiting exergy stream, auxiliary relations are required in order to solve the cost rate balance. For components with n exiting streams, (n-1) of such relations are needed.

In the case of the evaporator it is assumed that the cost per unit of exergy carried by the entering stream of heat source fluid is equal to the cost of the same stream leaving this component:

$$c_{hs1} = c_{hs2}$$

It is reasonable to charge the product, and not the fuel, with the costs of operation and maintenance of the component. Moreover, it is the exiting stream of organic vapor which should be penalized for the destruction of exergy occurring in evaporator. By this relation we assume that exergy losses in this component are covered by an additional supply of heat source fluid of which the cost in the geothermal plant is constant.

The cost rate of each stream of exergy dissipated to the environment is equal to zero. This relation applies to both the preheater ($c_{hs3} = 0$) and the cooling tower. It is assumed that these two streams leave components without requiring any additional treatment and do not cause any damage, which could be expressed in terms of money, to the environment. If they did, their cost rates would have negative values. In the case of the preheater in the geothermal power plant, an assumption of zero cost of exiting stream hs3 allows all costs of fuel, purchase and operation to be transferred to the other exiting stream (9) which is further used by other components.

In the waste heat recovery plant, the specific cost per unit of exergy carried by thermal oil is equal for all three streams (hs1, hs2 and hs3). Since the heat carried by stream hs3 is not dissipated to the environment, but enters another heat exchanger – the recuperator - exergy carried by this stream is not lost as it is in the geothermal power plant. Because of the fact that in both the preheater and the evaporator exergy is removed from thermal oil, the specific cost of exergy remains constant.

Another component which has two exiting streams is the turbine. The auxiliary relation used for this component is the equivalence of cost per unit of exergy for two streams of working fluid (streams 2 and 3). The main reason for this is that costs of exergy destruction, capital investment and operation of the component should be transferred to the cost of power, as it is the primary output of a turbine.

Similar methodology applies to the regenerator. Although it has in fact two purposes, preheating of pressurized working fluid leaving the pump and reducing the heat duty of the condenser, it is the specific cost of the hotter stream that remains constant.

Since the purpose of the condenser and cooling tower in the designed system is to dissipate heat from the working fluid, all the costs associated with the purchase and operation of these two components should be charged to the stream of condensed working fluid. The cost rate of stream c2 is taken as 0, which causes all costs of the cooling tower to be transferred to the stream c1, which in turn makes it negative. An equal rise in the exergy cost rate between streams 4 and 5 would be obtained if we assumed the cost for exergy carried by the stream c1 to equal zero (then cost rate of stream c2 would become positive). Cost balance of the condenser shows that it is stream 5 that, in the end, bears the full burden of the costs associated with the cooling tower and condenser:

$$\dot{C}_5 + \dot{C}_{c2} = \dot{C}_4 + \dot{C}_{c1} + \dot{Z}_{cond}^{CI} + \dot{Z}_{cond}^{OM}$$

Table 6-1 shows the cost rates and costs per unit of exergy of all streams shown in Figure 6.1 for each of the four systems. Taking into consideration that exergy flows of working fluid and cooling water are equal in systems A through D, variations in cost time rates are caused by different specific costs of exergy in each application. These specific costs of exergy are highly affected by the expenses for purchasing and operation of the components as well as the cost of fuel (which is easily visible when comparing cases A and B). The cost per unit of exergy increases for streams 1-8, as a part of the exergy is destroyed, and the cost rates associated with the components are added to the cost rate of the stream of organic fluid.

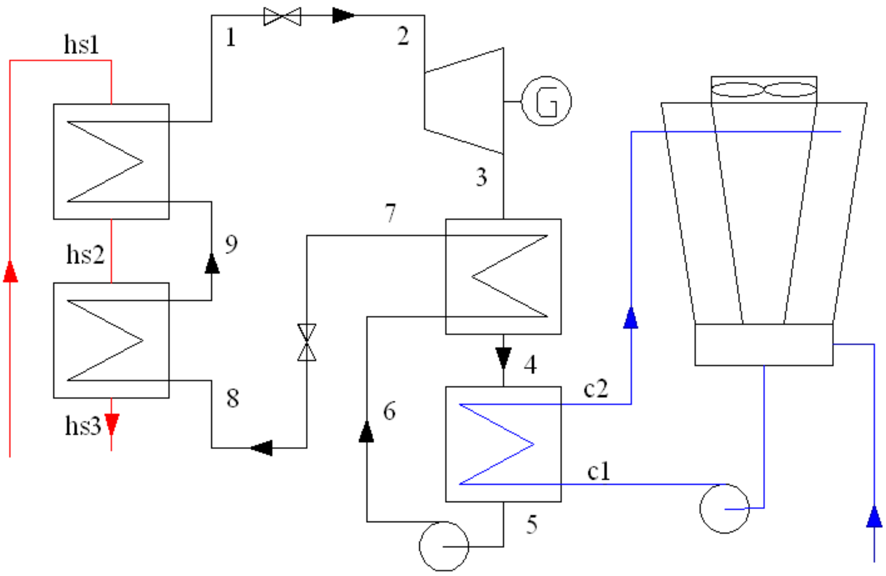

Figure 6.1 Schematic of ORC power plant

Table 6-1 Levelized cost rates and cost rates per unit of exergy

	Geothermal A	**Geothermal B**	**Waste heat C**	**Waste heat D**
	Cost rates $\dot{C}$ [10^{-3} U.S. $(2008)/s]			
hs_1	6,28	0,94	2,01	3,02
hs_2	3,62	0,54	1,18	1,77
hs_3	0,00	0,00	0,47	0,70
1	22,25	11,15	14,04	16,54
2	22,12	11,08	13,96	16,44
3	9,97	5,00	6,30	7,42
4	9,54	4,78	6,02	7,09
5	11,58	6,64	8,61	9,62
6	13,45	7,89	10,29	11,41
7	13,97	8,18	10,67	11,84
8	13,92	8,15	10,63	11,79
9	18,93	10,09	11,80	13,30
c_1	0,00	0,00	0,00	0,00
c_2	-1,54	-1,36	-1,87	-1,85

$\mathbf{w_{gen}}$	15,71	9,66	12,82	13,97
$\mathbf{w_{pump,wf}}$	1,63	1,00	1,33	1,45
$\mathbf{w_{fan}}$	0,47	0,29	0,38	0,42
$\mathbf{W_{pump,hs}}$	0,21	0,13	-	-
Cost per unit of exergy c [U.S. $(2008)/GJ]				
$\mathbf{hs_1}$	3,03	0,45	0,99	1,48
$\mathbf{hs_2}$	3,03	0,45	0,99	1,48
$\mathbf{hs_3}$	0,00	0,00	0,99	1,48
1	11,86	5,94	7,49	8,82
2	11,86	5,94	7,49	8,82
3	11,86	5,94	7,49	8,82
4	11,86	5,94	7,49	8,82
5	24,90	14,28	18,51	20,69
6	25,55	14,98	19,54	21,67
7	25,56	14,97	19,53	21,66
8	25,56	14,97	19,53	21,66
9	16,65	8,88	10,38	11,70
$\mathbf{c_1}$	0,00	0,00	0,00	0,00
$\mathbf{c_2}$	-7,49	-6,61	-9,13	-9,00
w	18,30	11,25	14,93	16,27

6.2 Cost rates of fuel and product

Concepts of fuel and product were defined previously in the exergy analysis chapter. The cost rates associated with fuel and product are defined in the same way as exergy rates and can be easily calculated using the methodology given by Bejan or Kotas.

However, one exception has to be made. In the majority of components, positive cost is transferred in the same direction as exergy. This indicates that the stream of product has to be charged with the costs of purchasing and operating the component. It is reasonable for heat exchangers whose purpose is to provide heating for the colder stream. However, this is not the case for the condenser. The main purpose of this component is to take away the heat from the hotter stream. Exergy, just like heat, can be transferred only in one direction and this rule applies to the condenser in the same way as it does to any other heat exchanger. Yet, as it was described before, its direction of cost transfer is different than in other heat exchangers. Thus the cost rates of the fuel and product of the condenser have to be redefined:

$$\dot{C}_{F,cond} = \dot{C}_{c1} - \dot{C}_{c2}$$

$$\dot{C}_{p,cond} = \dot{C}_5 - \dot{C}_4$$

Cost rates obtained this way are used for calculating the average costs per unit of fuel and product for each component, which for component k are defined as follows (Bejan):

$$c_{F,k} = \frac{\dot{C}_{F,k}}{\dot{E}_{F,k}}$$

$$c_{P,k} = \frac{\dot{C}_{P,k}}{\dot{E}_{P,k}}$$

Table 6-2 Average costs per unit of fuel and product for particular components (in U.S $(2008)/GJ)

	Geothermal A		**Geothermal B**		**Waste heat C**		**Waste heat D**	
	c_F	c_P	c_F	c_P	c_F	c_P	c_F	c_P
Preheater	5,11	8,46	0,77	3,28	1,01	1,97	1,52	2,55
Evaporator	3,03	4,49	0,45	1,429	0,95	3,03	1,42	4,38
Turbine	11,86	18,30	5,94	11,25	7,49	14,93	8,82	16,27
Regenerator	11,86	25,86	5,94	14,81	7,49	19,21	8,82	21,48
Condenser	4,53	10,68	4,00	9,73	5,52	13,52	5,45	13,23
Pump	18,30	30,43	11,25	20,24	14,93	27,34	16,27	29,04

6.3 Cost of exergy destruction and exergy loss

The cost rates of exergy destruction and exergy loss are of crucial importance in the optimization of thermal systems. The cost rate of exergy destruction is not present in the cost balance for the particular component - therefore is it sometimes referred to as the hidden cost. However, simple equations exist for calculating the costs associated with exergy destruction and loss (Bejan):

$$\dot{C}_{D,k} = c_{F,k}\dot{E}_{D,k}$$

$$\dot{C}_{L,k} = c_{F,k}\dot{E}_{D,k}$$

In the equations above it is assumed that the average cost of fuel remains constant with varying quantity of exergy destruction or loss. On the other hand, it could be assumed that it is the specific cost of exergy of the product, and not fuel, which remains constant. In fact, as described by Bejan, the true value of the cost rate of exergy destruction is between those calculated using specific cost of fuel and product. However, the second approach is not recommended as it overestimates the calculated cost rates due to irreversibilities.

Table 6-3 presents cost rates associated with exergy destruction and loss in particular components

Table 6-3 Cost rates of exergy destruction and loss

	Geothermal A	Geothermal B	Waste heat C	Waste heat D
	Cost rate of exergy destruction (10^{-3} U.S. $(2008)/s)			
Preheater	0,59	0,09	0,12	0,17
Evaporator	0,42	0,06	0,13	0,20
Turbine	1,96	0,98	1,24	1,46
Regenerator	0,20	0,10	0,13	0,15
Condenser	0,67	0,59	0,82	0,80
Pump	0,50	0,31	0,41	0,45
	Cost rate of exergy loss (10^{-3} U.S. $(2008)/s)			
Preheater	2,49	0,376	0	0

6.4 Relative cost difference and exergoeconomic factor

The relative cost difference for component k represents the increase in the unit cost of exergy between product and fuel expressed in relation to the unit cost of fuel (Bejan):

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}}$$

Relative cost difference is a very good way of pointing out ineffective elements in the system, which may have serious influence on the formation of costs in the system. Together with the exergoeconomic factor it is one of the most useful parameters in thermoeconomic optimization. The exergoeconomic factor defines what portion of the cost rate increase in considered components is caused by the destruction and loss of exergy, and what is due to a purchase and maintenance cost. By comparing values obtained in calculations to typical ones for the particular type of the component, which can be found in literature, it is possible to find the optimum balance between component efficiency and investment cost. The exergoeconomic factor for component k is defined as:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{F,k}\left(\dot{E}_{D,k} + \dot{E}_{L,k}\right)}$$

Table 6-4 Relative cost difference (r_k)

	Geothermal A	Geothermal B	Waste heat C	Waste heat D
Preheater	0,66	3,27	0,95	0,68
Evaporator	0,48	2,15	2,21	2,09
Turbine	0,54	0,89	1,00	0,85
Regenerator	1,18	1,49	1,57	1,44

Condenser	1,36	1,43	1,45	1,43
Pump	0,66	0,80	0,83	0,79

Table 6-5 Exergoeconomic factor (f_k)

	Geothermal A	Geothermal B	Waste heat C	Waste heat D
Preheater	0,31	0,75	0,80	0,71
Evaporator	0,61	0,91	0,64	0,53
Turbine	0,65	0,78	0,81	0,77
Regenerator	0,26	0,42	0,45	0,40
Condenser	0,43	0,46	0,47	0,46
Pump	0,33	0,45	0,47	0,43

Additionally, Table 6-6 shows the sum of levelized cost rates of capital investment, operation and maintenance and exergy destruction and loss. It depicts the cost importance of each of the components in the system and their potential for influencing the final cost of the product.

Table 6-6 Sum of levelized costs of TCI, O&M and exergy destruction and loss (in 10^{-3} U.S. \$(2008)/s)

	Geothermal A	Geothermal B	Waste heat C	Waste heat D
Preheater	4,48	1,86	0,57	0,61
Evaporator	1,08	0,72	0,36	0,42
Turbine	5,53	4,55	6,39	6,40
Regenerator	0,28	0,18	0,23	0,25
Condenser	1,18	1,10	1,53	1,49
Pump	0,75	0,55	0,77	0,79

6.5 Conclusions and recommendations from thermoeconomic analysis

The thermoeconomic evaluation was made based on tables included in this chapter as well as on Table 4-2. The highest overall cost rate of initial investment, O&M and exergy destruction is held by the turbine. It indicates that this component has a high potential for improvements and should be optimized in the first order. The calculated exergetic efficiency of the turbine is relatively high (83,9%) and the efficiency defect has a low, at least for this type of the component, value of 8%. The evaluation of costs confirms what the exergy analysis indicated. Since the relative cost difference of turbine in systems B,C and D is high, we should consider using less efficient but cheaper turbines in order to reduce the cost of the final product. This becomes even more evident when we consider the value of exergoeconomic factor calculated for this component. In the conventional geothermal system (A), where fuel entering the system is the most expensive, the value of variable f is approximated at 0,65. According to Bejan, this value should be considered high yet it is still acceptable as for this type of component. However, the exergoeconomic factor for the turbine in systems B,C and D is close to 0,8, which means that only 20% of the total cost increase is caused by exergy destruction. In order to improve the overall economics of the unit, we should consider redesigning the turbine. However, the cost function, describing the effect of organic fluid turbine efficiency on its purchase price, is not available and therefore this part of the optimization is not pursued.

As indicated by the sum of $\dot{Z}$ and $\dot{C}_d$, the next components with the greatest influence on the cost of the final product are the preheater and evaporator. A big discrepancy exists between units utilizing cheaper fuel (waste heat applications, where the cost of the fuel is basically equal to the levelized cost of purchase and maintenance of recuperator) and those using more expensive energy obtained from geothermal fluid. Relative cost differences in all cases, except scenario A, are high. In waste heat applications it is caused mainly by the low cost rate of the fuel, which allows even cheap heat exchangers, manufactured from carbon steel, to raise the cost of exergy product by a high rate. The highest values of relative cost difference are obtained for the preheater and evaporator in the geothermal plant using former CH wells. It is an effect of using the least expensive fuel among all other considered applications and the most costly heat exchangers manufactured from corrosion-resistant steel 254 SMO. In this case, the values of exergoeconomic factor are enormously high. They are typically lower than 55% (Bejan) for heat exchangers, but here reach over 78% for the preheater and 91% for the evaporator. This is why it may be expected that reduction in the size of these two components will be beneficial for system B.

Values of the relative cost difference and the exergoeconomic factor suggest that the reduction in size of the preheater and evaporator might also provide the most favorable results in applications C and D. However, it has to be remembered that, if manufactured from carbon steel, these heat exchangers generate a very low (approximately 6%) contribution to the total purchased-equipment-cost. Yet, they still have a crucial effect on the total exergy destruction and efficiency of the power plant. Therefore, it is expected that the size increase of the preheater and evaporator will positively affect the cost of electricity.

The regenerator is characterized by the lowest exergetic efficiency among all components in the designed system ($\varepsilon_{reg} = 53{,}9\%$). Because of the fact that using a highly-efficient regenerator decreases heat recovery efficiency, it is obvious that, in a cost-effective system, the regenerator will have lower exergetic efficiency than other heat exchangers. Among the four considered applications exergy destroyed in the regenerator is the most expensive in system A. Therefore, the chance of reducing the cost of power by increasing size of this component is the greatest in this system. Based on the data available, it is difficult to determine the possible optimum sizes of the regenerator in cases B,C and D, but the results from unit A may be used as indicator.

7 THERMOECONOMIC OPTIMIZATION

Thermoeconomic optimization is the final stage of the design procedure which, for defined boundary conditions, makes it possible to find the optimal values of independent variables. Values which minimize or maximize the chosen optimization criteria are considered to be optimal in this case. It may be the annual levelized net profit, time of return of investment or any other economic profitability criterion. In this thesis, the levelized cost per unit of net electric power generated by the plant was chosen as the primary measure of the performance of the system. It is considered to be the most universal optimization criterion since the other two mentioned are dependent on price at which electricity can be sold. This price may vary depending on the country and the type of application.

As a result, the procedure of thermoeconomic optimization gives proper values of independent variables, which, in our case, are the heat transfer areas of the preheater, evaporator, regenerator and condenser. It is important to emphasize that this process does not aim at reducing irreversibilities, but allows the finding of irreversibility rates that are most reasonable from an economic point of view.

Although various sources (Kotas, Bejan) provide a developed methodology for optimizing a single component, this procedure is not performed in this paper. Results obtained by single-component optimization cannot be used for optimizing the whole plant because they do not correspond to overall optimum of the system. Changes in the exergetic efficiency of one element in a cycle always affect the performance and economics of all other components. This fact is neglected when components are optimized in isolation.

The mathematical model created in EES was used in order to find the optimum size of heat exchangers. Calculations were performed for combinations of heat exchangers ranging in size as follows:

Preheater and evaporator	$290 \leq A_{ph+ev} \leq 390$
Regenerator	$0 \leq A_{reg} \leq 100$
Condenser	$460 \leq A_{reg} \leq 550$

Where A_k represents area of heat exchange in the heat exchanger in square meters.

The unit step for design variables was set at 5 m2. Although higher accuracy could be obtained, it is not considered necessary since cost curves close to the optimal point are usually flat. Cost rates obtained with such precision are very close to ideal optimal values.

7.1 Individual optimization of units for different applications

As a first step, a specific design that gives the lowest levelized cost of power was found for each application. Final results of this part of the procedure are shown in Table 7-1.

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Table 7-1 Results of individual thermoeconomic optimization of units for different applications

Application	Levelized cost of electricity	Annual electricity production	A_{reg}	A_{cond}	A_{vap+ph}	Change in lev. cost of electricity	Change in annual production rate
	U.S. ¢/kWh	MWh/yr	m2	m2	m2		
A	7,55	6091,0	70	505	300	-0,09%	-0,9%
B	5,04	5994,6	25	465	290	-1,00%	-2,5%
C	6,15	4922,7	25	480	355	-0,09%	1,0%
D	6,36	5165,9	45	505	355	-0,18%	1,6%

It can be noticed that optimal configuration of component sizes for each application is different. Although individual optimizations of the system gave more cost-effective designs for each option, obtained improvements were not tremendous. This discovery confirms that, as long as boundary conditions in different applications of ORC plant are similar, the design of a standardized unit is sensible. Cost reduction achievable by the individual design of a unit for a particular use is minor compared to savings which may be seen from big series production.

The highest reduction of levelized cost of electricity occurred for option B. Because of the lowest cost rate of fuel and expensive heat exchangers manufactured from stainless steel, the reduced size of the preheater and evaporator resulted in a 1% improvement of the final product cost compared to initial design. However, the annual production rate decreased by 2,5% as an effect of lower exergetic efficiency of heat exchangers.

Exactly as expected, in a conventional geothermal power plant (case A), the biggest change in optimal configuration compared to initial design is the increased size of the regenerator. This change allowed the reduction of the levelized cost of electricity by 0,11%, but also resulted in a considerably lower annual production rate. Such a change in design also has another advantage. It is the lower risk of scaling due to the increased temperature of reinjected brine. Nonetheless, if the price at which electricity is sold is much higher than the production cost, calculations should be repeated using a different optimization criterion - one which would also take into account the value of the production rate. Levelized net profit may be used, for example, as an objective function.

As it was indicated by thermoeconomic analysis, the optimization of waste heat recovery plants resulted in a significant increase in the heat exchange area of the preheater and evaporator. The cost of electricity produced by the plant did not drop much, yet, due to the improved overall efficiency, the annual production rate was raised by 1% and 1,6% for scenarios A and B respectively.

It is worth noticing that the optimal design of the waste heat recovery plant includes the regenerator. The majority of commercially available ORC heat recovery plants do not use a regenerator and the process of desuperheating occurs in the condenser. One might think that if the stream of waste heat exhausted from the engine or furnace is free of charge, a regenerator would be a costly and senseless option. However, due to the high capital investment cost of a gas/thermal oil recuperator and its high maintenance requirements, exergy delivered to the working fluid is not cheap. In fact it is more costly than in a power plant using geofluid from a former CH well. The optimal configuration of the unit without a heat regenerator proved this thesis. The results for both the diesel bottoming power plant (C) and the unit using exhaust gasses from a clinker cooler (D) are shown in Table 7-2.

Table 7-2 Results of individual thermoeconomic optimization of waste heat recovery plants without regenerator

Application	Levelized cost of electricity	Annual electricity production	A_{cond}	A_{vap+ph}	Change in levelized cost of electricity	Change of annual production rate
	U.S. ¢/kWh	MWh/yr	m2	m2		
C	6,19	4941,3	525	360	0,61%	1,3%
D	6,41	5215,1	550	380	0,55%	2,6%

Without the regenerator, the levelized cost of electricity increased by 0,61% and 0,55% in cases C and D respectively. That is a significant decrease in performance, especially when compared to the minor benefits obtained by thermoeconomic optimization.

7.2 Optimization of a standardized ORC power plant

Despite the fact that optimal values of design variables differ from one application to another, the purpose of this thesis is to design one universal power plant. Thus, a design has to be found which ensures the level of performance close to optimal in each of the considered applications. The EES model was used for this purpose. The optimal configuration of components was chosen based on the weighted average of levelized costs of electricity for cases A-D. Because of expected non-equal demand in the market, a higher weighting factor is assumed for applications A and C.

As a result, it was found that the lowest average levelized cost of net energy is obtained for the system as follows:

A_{ph+vap}=310 m^2

A_{cond}=495 m^2

A_{reg}=45 m^2

These optimal values are very close to those which were assumed in the beginning, which minimize the significance of exergy-aided cost optimization.

7.2.1 Thermodynamic performance

The performance of the designed plant is shown in Figure 7.1. Net electric power output, total power generated and the temperature of the heat source fluid at the outlet of the preheater are calculated as functions of wet bulb temperature. For typical European weather conditions, net power output of the plant ranges between 580 and 800 kW, and the temperature of water reinjected from the geothermal plant varies from 70 to 76,7°C.

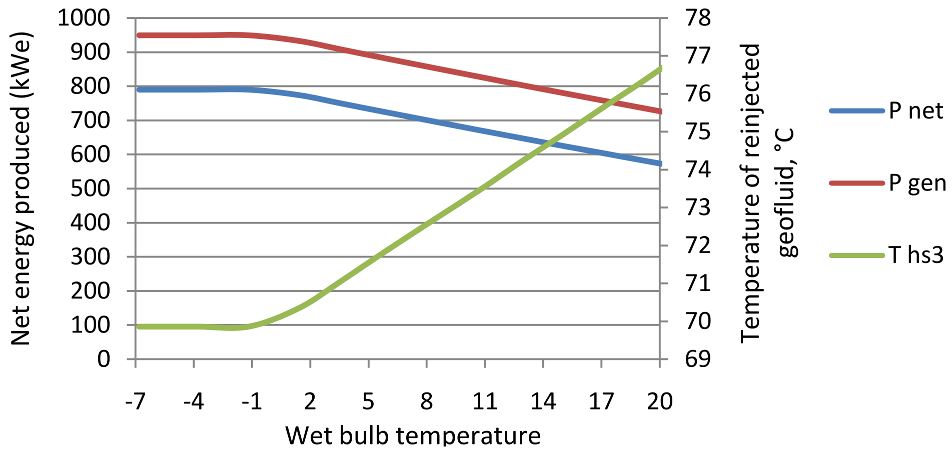

Figure 7.1 Performance of a power plant for variable ambient wet bulb temperature

The power duration curve was obtained by combining the data from Figure 7.1 with the temperature duration curve shown in Figure 3.1. This curve, shown in Figure 7.2, assumes that the load factor equals one. It can be noticed that, with a constant load, the unit generates at least 75% of its maximum power output during 97% of the year.

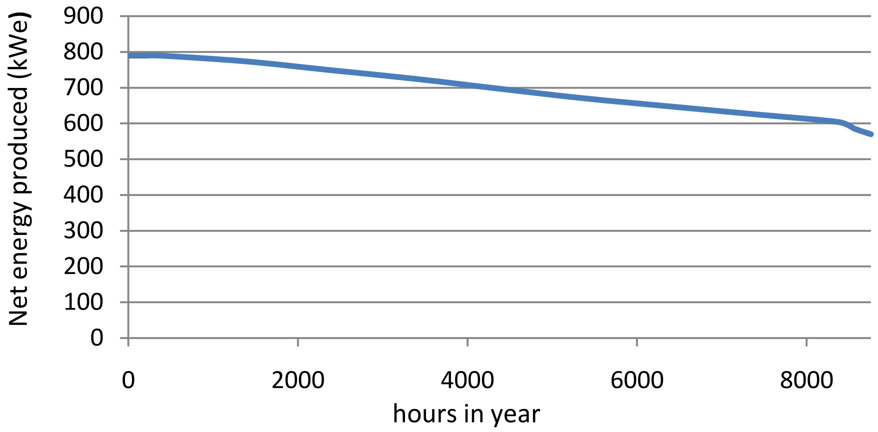

Figure 7.2 Power duration curve for a plant working on maximum load

The annual production rate, which is basically the integral over this power duration curve multiplied by the load factor, for applications A and B is equal to 6106 MWh and for cases C and D, due to a lower load factor, equals 4843 and 5051 MWh respectively.

The efficiency of the unit decreased as an effect of the decreased total heat exchange area. The Second Law utilization efficiency of the plant is 1% lower than for the design described in the fourth chapter and now equals 44,1% for an assumed outdoor wet bulb temperature of 8,4°C. Values of the Second Law utilization efficiency and the maximum First Law efficiency are plotted below as functions of outdoor temperature.

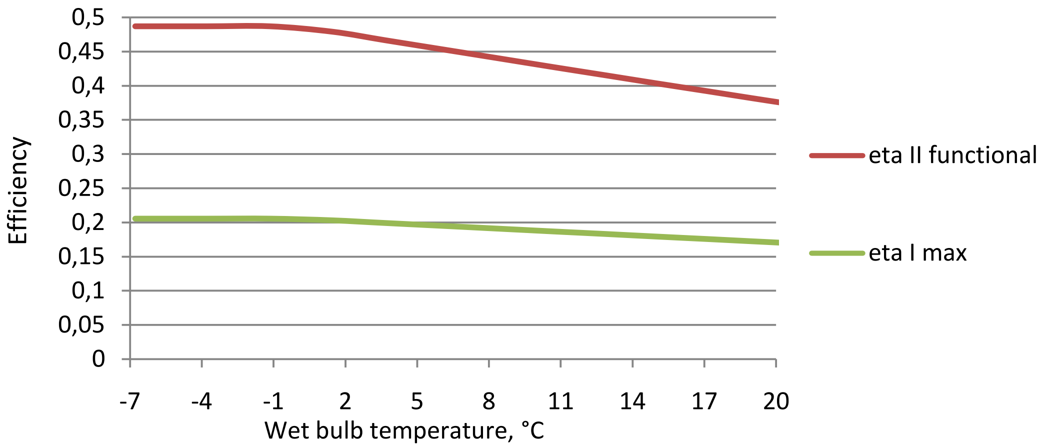

Figure 7.3 Second Law functional efficiency and maximum First Law efficiency as a function of ambient wet bulb temperature

7.2.2 Economic performance

Table 7-3 shows final results for optimized universal unit.

Table 7-3 Final results of optimization of standardized ORC power plant

Application	Levelized cost of electricity (U.S. ¢/kWh)	Annual electricity production (MWh/yr)	Change in lev. cost of electricity	Change in annual production rate
Geothermal A	7,55	6107,8	-0,02%	-0,7%
Geothermal B	5,07	6107,8	-0,35%	-0,7%
Waste heat C	6,16	4845,1	0,07%	-0,6%
Waste heat D	6,38	5052,7	0,10%	-0,6%

The shift of the final optimal design towards low efficiency is caused by the influence of scenario B in which the cost rates of exergy destruction are very low. The only reasonable benefit (0,35% reduction in cost of electricity) was achieved in this particular application. Changes in economic performance in all other cases are negligible.

The annual production rate decreased as an effect of this optimization but this drawback can be accepted since the only basis of the evaluation was the levelized cost of unitary product. If the price at which electricity could be sold was known however, the primary optimization criterion should be changed to e.g. levelized net profit.

Table 7-4 shows the breakdown of total pre-operation investment. Despite design differences, the cost of the power plant used for each application is almost equal. However, due to the high cost of site exploration and drilling, which for scenario A contributes to almost 45% of TCI, the value of initial investment in geothermal units and waste heat recovery plants differs significantly.

The capital cost of power plants is often presented in relation to installed power. It was estimated by GEA, that in 2005 the minimal TCI of a new binary power plant was approximately $2800/kW, and for the majority of projects under development it ranged between $3000 and $4000/kW. Unfortunately, over the last three years the cost of geothermal power projects increased considerably, mainly due to the rise of drilling costs and prices of raw materials. Therefore the value of capital investment calculated for scenario A, which is equal to $3872 per kW of net power output (corresponding to $3224 per kW of generated power) is considered relatively low. For a geothermal plant using a former CH well (case B) this value equals to $2415/kW of net electric output and it drops to approximately $2100/kW for waste heat recovery applications.

The levelized cost of electricity produced by the designed unit, shown in Table 7-3, is also moderate. Some reports give slightly lower numbers for the cost of electricity produced in binary plants. The Geothermal Energy Association estimated in 2005, that the minimal power production cost from binary plants is around 5,7 ¢/kW. However, other sources (California Energy Commission, Glitnir Bank) give much higher numbers of 7-7,5 ¢/kW. Moreover, it has to be emphasized, that all of these reports were prepared between the years 2003 and 2007, when costs in the geothermal industry were much lower.

If the considered power plant was to be erected in the country where geothermal energy is subsidized, the payback time could be very short. In Germany, since the new Renewable Energy Sources Act was passed in 2004, energy produced from small (under 5MW) geothermal power plants is sold for guaranteed price of 0,15€/kWh. This is more than double the production cost for the designed power plant.

Even in the most expensive of investigated scenarios, electric power is supplied with the cost of 7,55 ¢/kWh which makes it cheaper than the majority of other renewable sources. According to Glitnir Bank and the California Energy Commission, only wind and biomass plants can achieve similar or even lower costs of production. However, they cannot be used as a base load, as geothermal power is an ideal source for this.

The second type of geothermal application, where brine is obtained from a former oil well, gives the lowest levelized cost of electricity, equal to 5,05 ¢/kWh. According to the literature, this price is below the range typical for other kinds of renewable energy. Moreover, the levelized cost of electricity in this case was calculated with the assumption of the power plant lifetime of 15 years, while the true operation period may even exceed 30 years.

The biggest disadvantage of using old CH wells for geothermal purposes is the shortage of productive wells which can supply high-temperature fluid. Besides, as it was noticed by Holl et al., one has to expect additional problems with drilling and scaling as well as other unpredictable situations since reports from abandoned wells usually do not include all the necessary information.

The designed Organic Rankine Cycle recovery unit can supply electricity in both described waste heat recovery applications at a cost slightly exceeding 6¢/kWh. For a plant coupled with a diesel engine this cost is lower due to higher temperature of the heat source. The calculated levelized cost is higher than for fossil fueled power plants, but still lower than for other renewable energy sources. However, due to variable heat supply to the recovery plant, it loses the big advantage of the geothermal power plant – high load factor. In order to improve both economics and reliability of the power supply, the recovery plant should use a heat source with a constant power output. The magnitude of the effect of load factor on a levelized cost of electricity is shown in the next chapter.

Table 7-4 Breakdown of total pre-operation investment

	Geothermal A	**Geothermal B**	**Waste heat C**	**Waste heat D**
Evaporator	73.447	73.447	24.724	24.724
Preheater	154.654	154.654	48.369	48.369
Turbine	641.890	641.890	641.890	641.890
Regenerator	10.592	10.592	10.592	10.592
Condenser	77.653	77.653	77.653	77.653
Pump	44.402	44.402	44.402	44.402
Cooling tower	153.872	153.872	153.872	153.872
Recuperator	-	-	124.217	194.615
Total PEC	1.156.510	1.156.510	1.125.719	1.196.117
Piping	80.956	80.956	101.315	107.650
Installation of equipment	69.391	69.391	67.543	71.767
Instrumentation and controls	57.825	57.825	56.286	59.806
Electrical equipment	46.260	46.260	45.029	47.845
Civil and structural work	80.956	80.956	33.772	35.883
Total direct costs	1.491.898	1.491.898	1.429.664	1.519.068
Engineering and supervision	69.391	69.391	67.543	71.767
Construction costs	44.757	44.757	42.890	45.572
Contingency	48.181	48.181	46.203	49.092
Total indirect cost	162.329	162.329	156.636	166.431
Fixed Capital Investment	1.654.717	1.654.717	1.586.581	1.685.499
Startup costs	16.542	16.542	15.863	16.855
Working capital	34.695	34.695	33.772	35.883
Total Capital Investment	1.705.954	1.705.954	1.636.216	1.738.237
Drilling and site development	1.355.000	203.189	-	-
Total pre-operation investment	**3.060.954**	**1.909.143**	**1.636.216**	**1.738.237**

8 SENSITIVITY ANALYSIS

The levelized cost of energy is dependent on many factors, including the characteristics of the heat source, heat sink, macroeconomic climate and project-specific considerations. In the previous analysis for all of these parameters constant, typical values were assumed. However, since each project is different it would be useful to know how sensitive the levelized cost of power is to the change of each of these variables. This chapter presents an analysis of the sensitivity of the cost of geothermal power to capital investment, the cost of providing heat input to the cycle, O&M costs and the cost of borrowing of money, and the escalation rate. Special consideration is given to the effects of variable temperature and flow rate of geothermal fluid in applications A and B since these parameters are very site-specific and influence the cost of the final product to a great degree.

8.1 Geothermal applications

Figure 8.1 and Figure 8.2 show the relative cost of power against the percentage of deviation of a given variable for applications A and B respectively. High sensitivity of the cost of power is in this type of graph implied by a steeper curve.

A relative levelized cost of power was used on ordinate instead of cost in ¢/kWh. It not only allows the comparison of the sensitivity of plants in different applications, but also helps to estimate the cost when more than one variable is different from the base design (by multiplying relative costs for all independent variables by base case levelized cost). However, these variables are not fully independent i.e. change in one of them affects the significance of the others. Thus, such estimation provides only approximate results.

The cost of power is quite sensitive for the effective rate of return in both scenarios. An increase of i_{eff} to 12% results in a 10% rise in the levelized cost of electricity. On the other hand, both units are almost insensitive to a nominal escalation rate since it affects only O&M expenditures. The presented analysis assumes that the price at which electricity is sold is constant - unaffected by inflation during the whole lifetime of the plant. If it was not, an increased escalation rate would have a positive effect on the cost of final product.

Sensitivity to change in capital investment, the cost of drilling and O&M costs is dependent on the share of these costs in total expenditures. Thus, as it might be expected, a geothermal plant using a newly drilled well is highly affected by the change in capital investment and the cost of drilling. On the other hand, the cost of power obtained in a plant working with a second-hand well is highly influenced by capital investment but relatively insensitive to drilling cost.

The influence of the length of the power plant's lifetime clearly shows the meaning of the time value of money: the shortening of economic life has a highly negative influence on the levelized cost of electricity but its lengthening gives much smaller benefit.

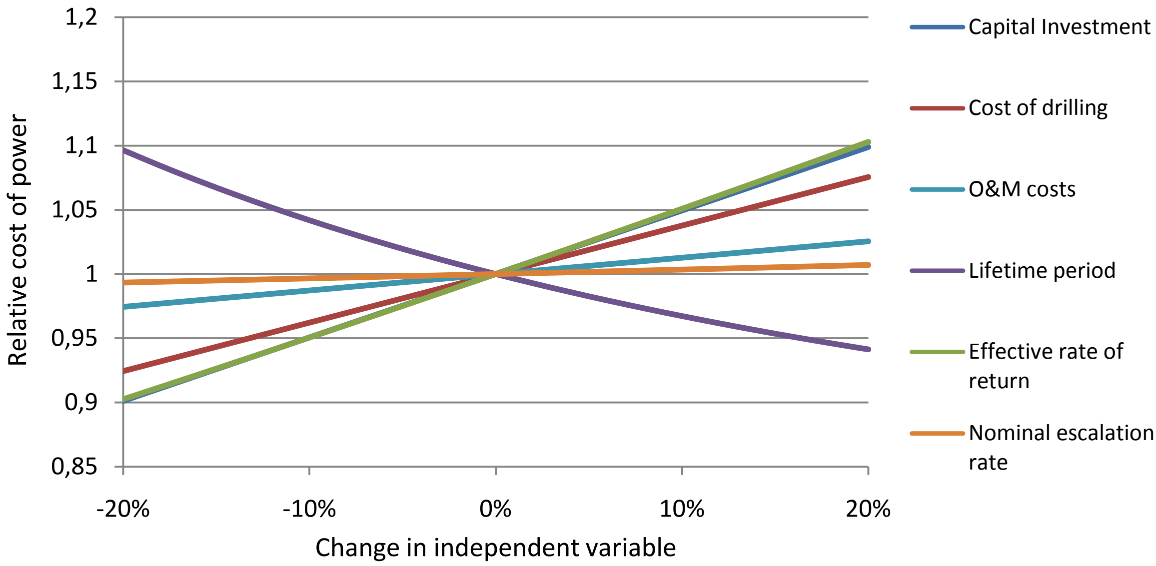

Figure 8.1 Sensitivity of base case power cost for geothermal power plant using new drilled well (A) to changes in independent variables

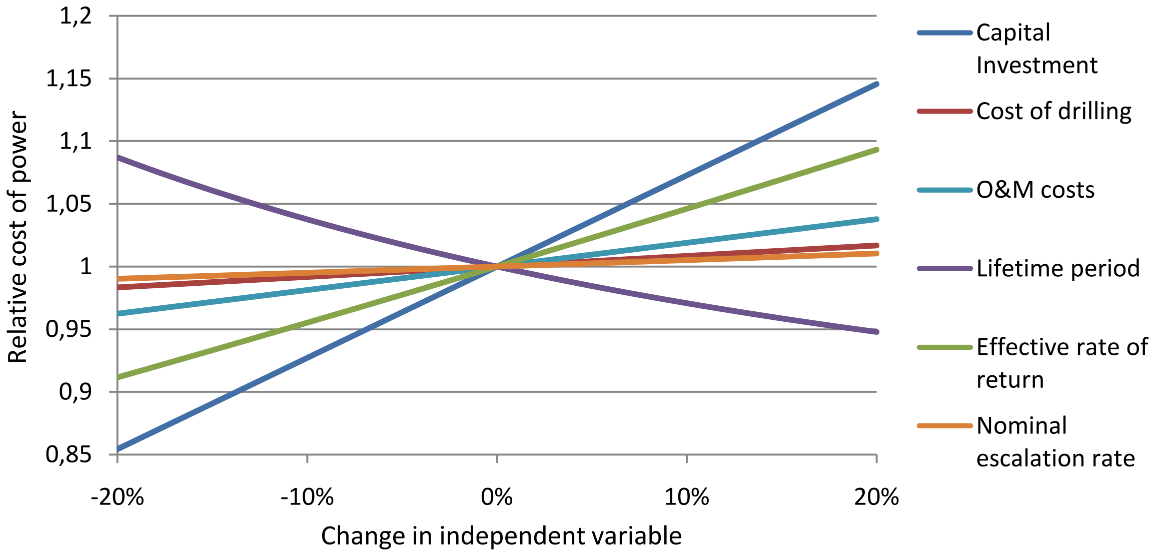

Figure 8.2 Sensitivity of base case power cost for geothermal power plant using old CH well (B) to changes in independent variables

The relative cost of produced power is also dependent on non-economic boundary conditions, which are assumed to be fixed in the foregoing analysis. Although the number of such parameters was high, only a few of them a have significant impact on the power plant´s performance. Without any doubt the parameters characterizing the stream of geothermal fluid delivered to the plant are: its temperature and mass flow rate.

Figure 8.3 and Figure 8.4 show the sensitivity of levelized cost of power to wellhead temperature and volumetric flow rate of brine for applications A and B respectively. For each configuration of boundary conditions, the evaporation temperature was chosen to maximize the net power output of the plant. Moreover, it was assumed, that the turbine and generator are not oversized i.e. power output of the plant cannot exceed coldest-day power production from base case design. The effect of off-design working conditions on isentropic efficiency of the turbine is neglected

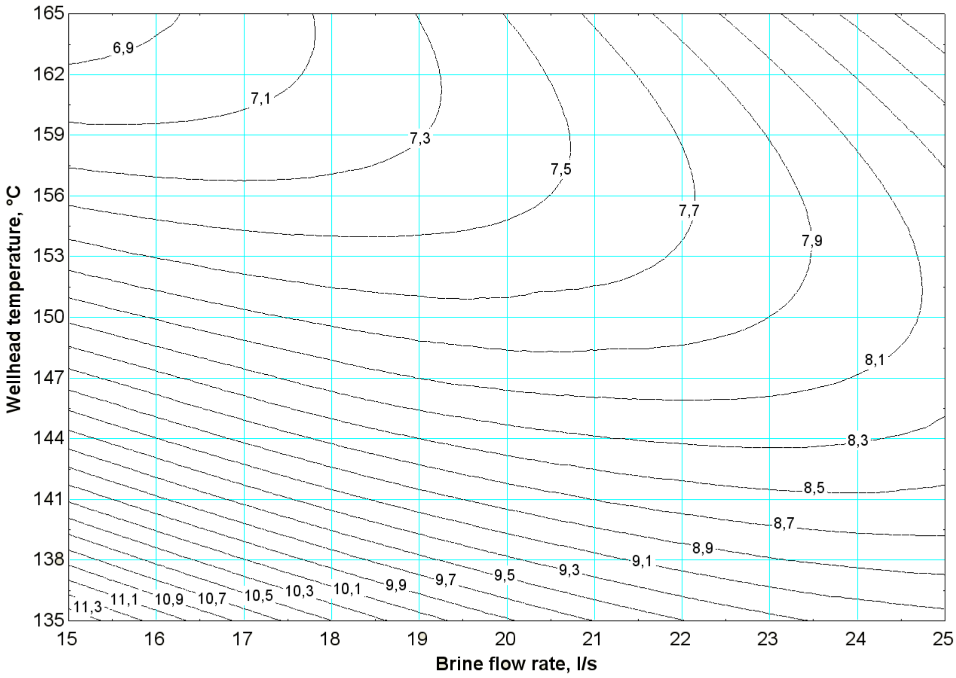

Figure 8.3 Sensitivity of base case power cost to resource characteristics for geothermal plant using water from new drilled well (A)

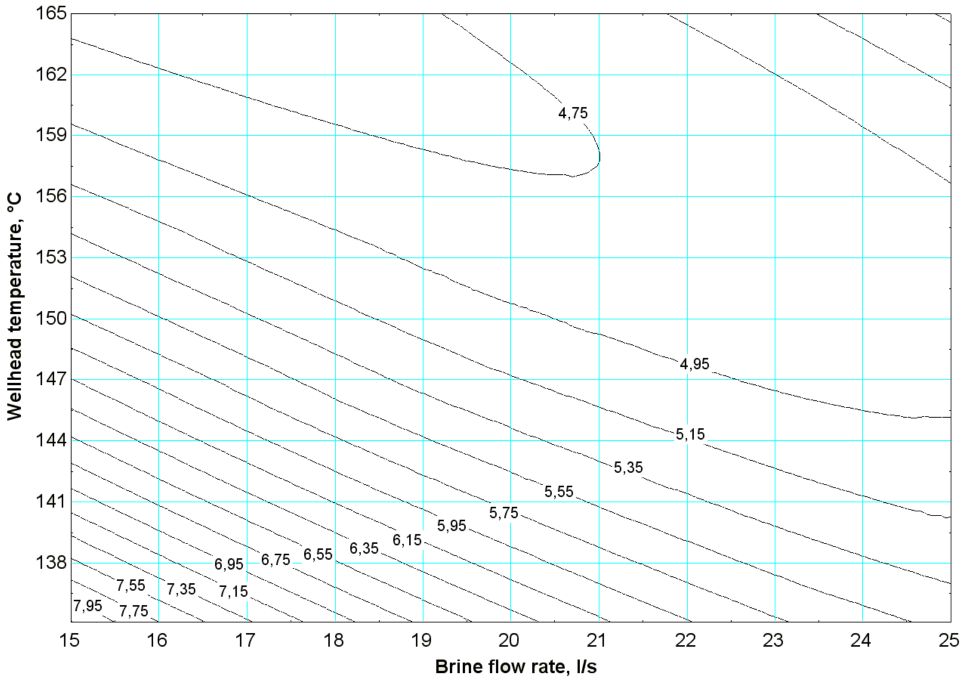

Figure 8.4 Sensitivity of base case power cost to resource characteristics for geothermal plant using fluid obtained from old CH well (B)

The economic performance of both units is highly sensitive to the temperature and flow rate of geothermal water. The lowest levelized cost of power is obtained for high wellhead temperature and values of flow lower than in the base design. It is because the rise in geothermal fluid temperature already makes the power plant work at its full capacity and therefore a lower flow rate than in the base case is required. Since the cost of drilling was assumed to be a linear function of brine flow rate, lower brine requirement has a positive effect on the levelized cost of electricity.

For low-temperature wells the cost of power becomes very high. For wellhead temperature of 140° and base design flow rate, the cost of power increases by approximately 1,5 and 1¢/kWh for applications A and B respectively. In a plant using a second-hand well this increase can be reduced if a flow rate higher than 20 l/s can be supplied. An increased flow rate does not have such a dramatic effect on the economics of a unit using a conventional geothermal well. The curves of constant levelized cost of energy in the lower- left part of Figure 8.3 are almost horizontal. This indicates that losses caused by the lower temperature of geofluid cannot be covered by an increased flow of fuel.

The base case analysis for geothermal applications assumed that water level in the well is at the surface of the ground. This is a very simplified assumption, which for most wells gives an optimistic estimate of the price of the final product. Due to the fact that in the base case the downhole pump is used only to maintain liquid state of high-temperature fluid, its power requirement is very low. Yet, parasitic energy consumption of the pump becomes very influential with an increased depth of water level. Figure 8.5 shows the effect of the height of pumping on relative cost of power. Additionally, work done by the downhole pump affects the cost rate of exergy of geothermal fluid. Thus, if the drawdown in the well is high, an optimally designed plant will have higher exergetic efficiency than the base design.

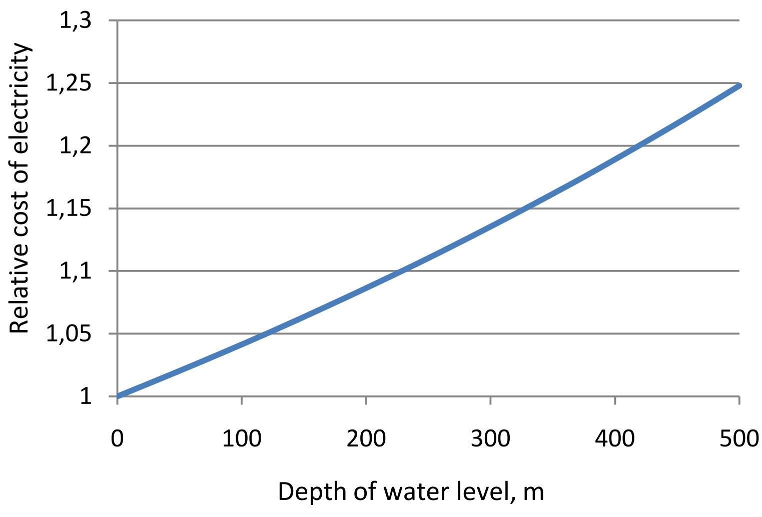

Figure 8.5 Relative cost of power in geothermal power plant versus depth of water level in well

From both a thermodynamic and an economic point of view, the optimal temperature of geothermal fluid leaving the preheater in the base case design is 71,5°C. It is known that the temperature is the main factor governing water-mineral equilibriums and its extensive lowering may cause scaling. While in a majority of geothermal fields in Europe this temperature will not induce any significant scaling, in some of them – the ones with very unfavorable chemistry - this may be the case. In such projects the trade-off has to be made between the potential of scaling and the performance of the plant. The reinjection temperature may be increased through changes in the pressure of evaporation of the working fluid, but it reduces the thermal efficiency of the plant. Impact of such an action on the levelized cost of power in scenario A is shown in Figure 8.6. Sensitivity is very high in this case due to the large contribution of drilling to total initial cost. In scenario B, where drilling cost was assumed to be only 15% of the equivalent of the new well, the impact of recovery efficiency on the levelized cost of power is very low.

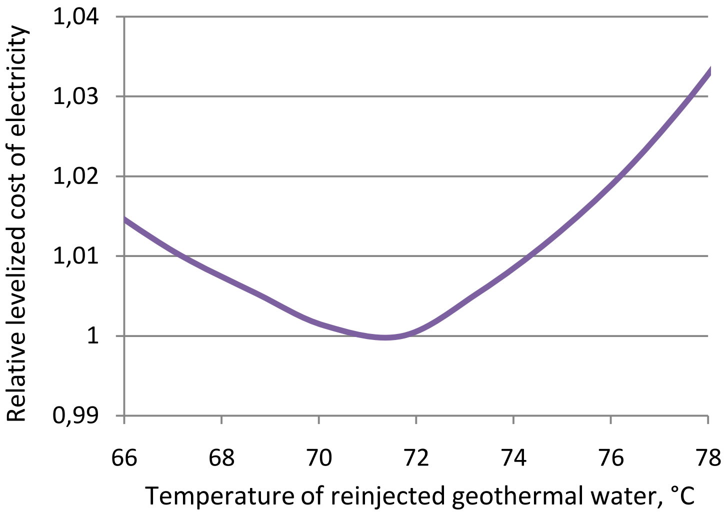

Figure 8.6 Sensitivity of levelized cost of power to temperature of reinjected geofluid

8.2 Waste heat recovery applications

Similarly to geothermal plants, plots of the levelized cost of power versus deviation of economic variables from their base case have been made for waste heat recovery. Since the distribution of costs in both plants C and D is very similar, results are presented on one chart. The cost of delivering exergy to the preheater and evaporator is in this case included in the capital investment, which means that the impact of this category of costs is higher.

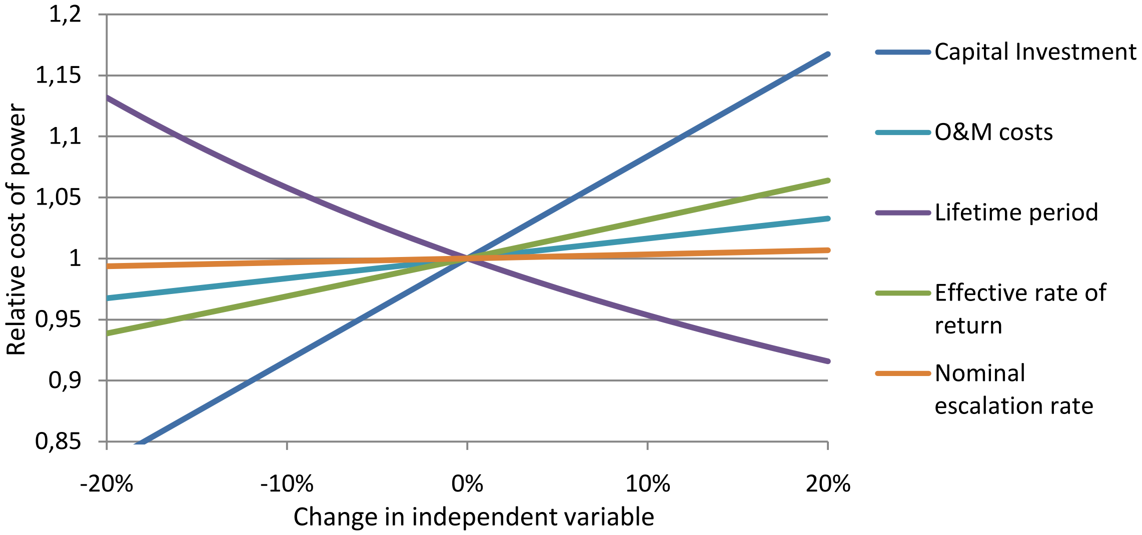

Figure 8.7 Sensitivity of base case power cost for waste heat recovery plant to changes in independent variables

9 ADDITIONAL CONSIDERATIONS

9.1 Effect of size

The effect of size on the prices of components was described in the fifth chapter of the thesis. The designed unit is of typical size for waste heat recovery plants and geothermal applications using old CH wells, but relatively small when compared to typical solutions found in the geothermal industry. This choice was made based on the assumption that the coupling of many small units can give a better fit between the flow rate required by the plant and the productivity of the well.

Figure 9.1 compares the levelized cost of power and the total capital investment for the two different cases. Values for the universally designed unit with flow the rate specified on abscissa are presented with continuous lines. Calculations assume that each component of the system can be manufactured as a single piece i.e. there are no size restrictions for any of the heat exchangers. Additional costs related to transportation, construction and erection of the power plant are constant per unit of power output. It has to be emphasized that these results do not relate to typical, individually designed power plants as they include all price reductions due to serial production. Results for the final-design plant working either alone or in a parallel network are marked with separate points.

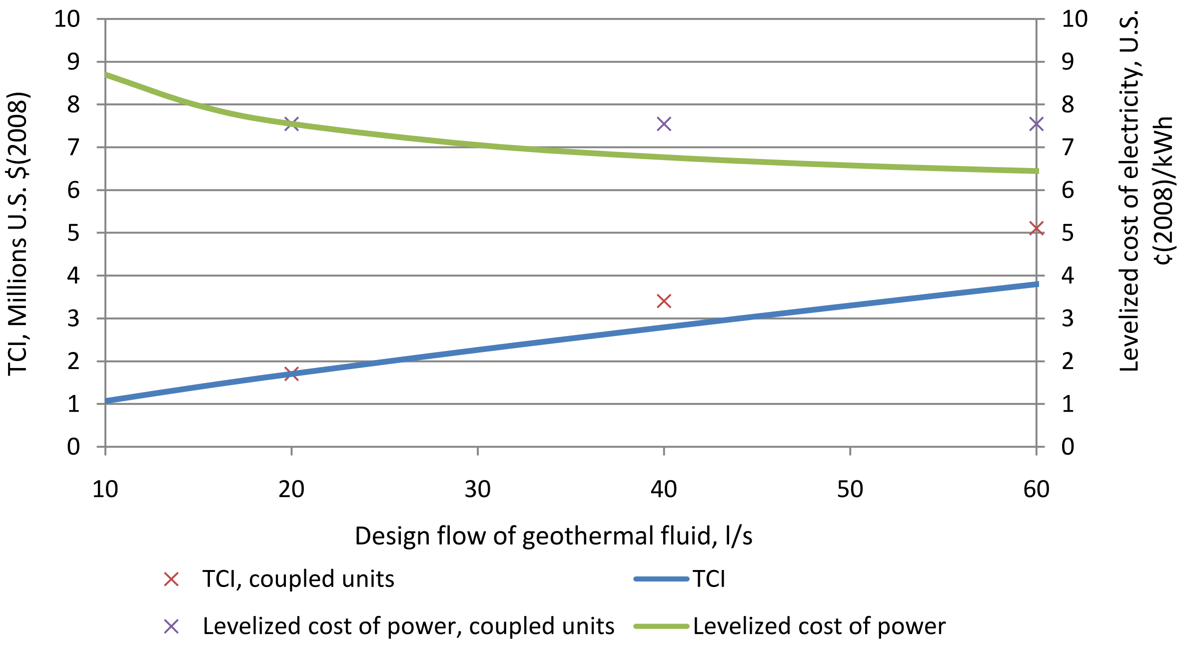

Figure 9.1 Effect of the size of unit on levelized cost of power

Cost change caused by the size is higher for units considered to use less than 30 l/s of geofluid. Moreover, designing standardized units for flows higher than 30 l/s would reduce its applicability - especially in waste heat recovery application. On the other hand, the levelized cost of power produced by plants smaller than in our base design is too high to be competitive on the market of renewable energy. Therefore, since a trade-off has to be made between high cost-efficiency of the unit and its applicability for wide range of applications, its optimum size is equivalent to the heat source fluid flow of 20 to 30 l/s.

9.2 Performance of parallel networks

As it has been proved already, off-design working conditions are very harmful for ORC power plants. The productivity of geothermal wells in already existing and ongoing projects listed in

Table 1-2 is very random and ranges from 13,3 to 150 l/s. Even though the designed plant is relatively small and the coupling of units should result in a good fit between the designed and available exergy input, almost each project unit will operate in more or less off-design conditions. Figure 9.2 shows the levelized cost of power as a function of the productivity of a typical geothermal well for a heat source fluid temperature of 150°C. The number of units working in the network was chosen to minimize the final cost of electricity.

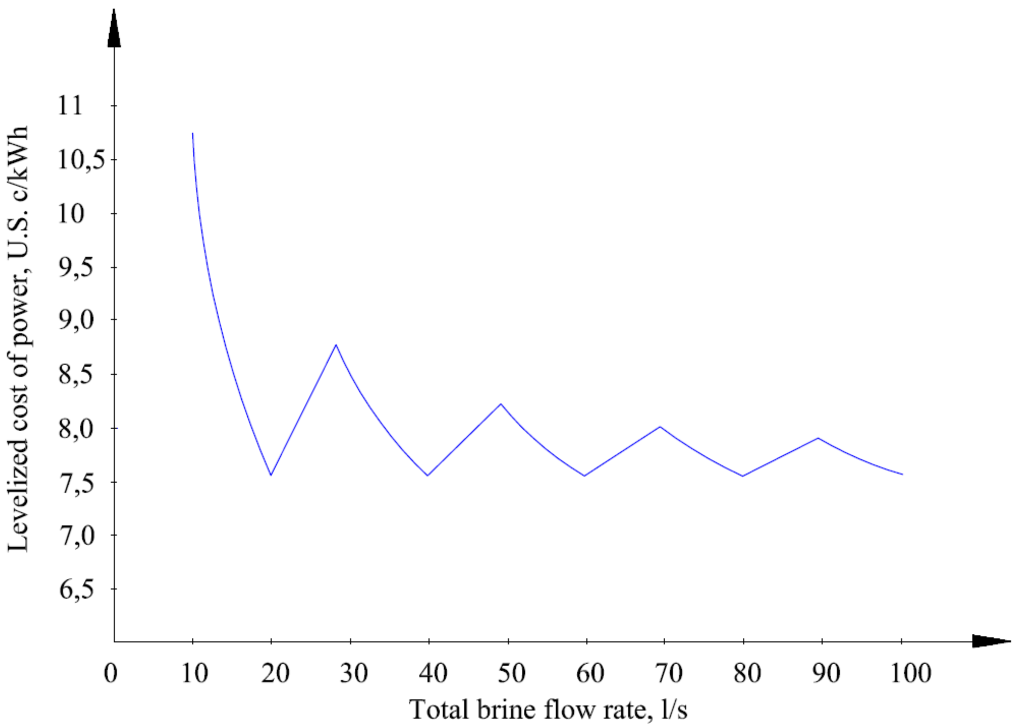

Figure 9.2 Levelized cost of power for parallel networks of units using water from conventional geothermal well

For very low-productivity wells with outflow lower than 15 l/s, the cost-effectiveness of the designed unit decreases drastically. The cost of power produced by the system operating on brine flow rates of 25-32 l/s is also relatively high (8,4 to 8,8 U.S. ¢/kWh). However, for more productive wells, the curve of levelized cost flattens as the mismatch between the design and project-specific boundary conditions becomes lower. The levelized cost of power does not go above 8 U.S. ¢/kWh for well head outflows exceeding 52 l/s.

9.3 Disadvantages of universal design and expected problems

The foregoing analysis includes many assumptions and simplifications. Some of them could not have been avoided due to time limitations and the overall complexity of the problem.

The method used for estimating the Total Capital Investment is probably the biggest drawback. The factor method introduced by Bejan gives only a rough approximation of initial costs. Allocation of capital investment in Organic Rankine Cycle power plants is much different than for fossil fuel plants. Due to the low popularity of binary units, the literature containing estimations of initial expenditures is insufficient, and available data for fossil fuel plants does not apply. This is why the values presented in this thesis should be treated as approximations. More accurate numbers can be obtained with the help of a cost engineer experienced in ORC power plants.

Another disadvantage of the presented conceptual model is the lack of any connection between the reservoir and the power plant. A true analysis of the economic life should include the changes of well productivity and downhole water pressure and their impact on the cost of the final product. However, the influence of the drawdown was neglected in this paper because if its very site-specific nature.

It was proved that, due to the differences of thermodynamic characteristics of the heat source, standardized power plants used in the same type of applications can have very different economic performance. However, introducing identical power plants to various types of applications is even more challenging and may cause additional problems.

The presented analysis assumes that the cooling of combustion gasses to 130°C does not cause condensation of vapor in the recuperator. Since this heat exchanger is manufactured from corrosive carbon steel, excessive cooling of gaseous stream should be avoided. Various sources give different estimations of the dew point temperature of water-acid mixtures. Invernizzi suggests that exhaust from a natural gas turbine can be cooled to 70°C. On the other hand, studies by Pejuro show that although the dew point of the mixture HNO_3/H_2O is only 56 °C, H_2SO_4 may condensate even if cooled only to 115 °C. If the condensation temperature of any of these two mixtures is lower than the temperature of the thermal oil entering the recuperator, its flow velocity in the loop has to be increased.

10 CONCLUSIONS

It has been shown that the design of a standardized ORC power plant is sensible. It has to be noticed that such a unit performs well only as long as the thermodynamic characteristic of the heat source is similar to the one it was designed for. Therefore, binary plants with geothermal purposes should be used only for those heat recovery applications where the temperature of the exhaust gases is relatively low or unstable. If used as a bottoming cycle for naturally aspirated engines exhausting at high temperature or heat treatment furnaces, the unit described in this thesis will not be competitive against a conventional steam boiler or other ORC power plant using higher temperatures of thermal oil.

However, the designed unit can be used in various types of applications and is capable of generating power at a fairly low cost. The analysis has shown that, depending on the type of application, this cost ranges from 5,07 to 7,55 U.S. ¢/kWh for unit operating in optimal conditions. Taking into consideration the increase in price of raw materials and the drilling of geothermal wells that occurred in the last years, these values are fairly low. Unfortunately, the ORC power plant has also been found to be quite sensitive to the thermodynamic properties of the heat source. In geothermal applications, decreased temperature, flow rate or high water level drawdown can easily defeat the economic viability of the project. This problem may be partially solved by introducing a series of standardized power plants designed for various heat source temperatures and brine mass flow rates.

Among all the investigated applications, the lowest levelized cost of electricity was obtained for a geothermal plant utilizing water from a former oil and gas well. Unfortunately, the number of productive CH wells which are capable of providing fluids at the required temperature is very low. As shown in the sensitivity analysis, in cases where the well is not able to provide the required exergy input to the cycle, the levelized cost of electricity drastically rises.

The levelized cost of power produced by a waste heat recovery plant was found to be 6,16 and 6,38 U.S. ¢/kWh for units coupled with a diesel engine and a cement plant respectively. Although the electricity produced by the unit operating in the cement plant is more expensive, the designed power plant seems to be especially suitable for this application. Low exergy of exhausted waste heat and unstable heat supply favor the ORC power plant with low temperatures of thermal oil over other solutions. On the other hand, the bottoming unit designed to operate on exhaust gasses from a diesel engine, if designed only for this purpose, would operate on higher temperatures of thermal oil and achieve higher thermal efficiency. Problem with the sensitivity to heat source characteristics can be solved for a recovery plant by sacrificing the recovery efficiency. In order to minimize the cost of energy output, the power plant may be slightly undersized since the prime objective of each project is to be profitable and the stream of waste heat is considered free of charge.

Thermodynamic optimization of the units gave meaningful results as well. It was shown that, for the heat source temperature of 150 °C, isopentane assures the best performance among all four investigated fluids. Isopentane and n-pentane were proven to be unable to fully utilize low outdoor temperatures, which resulted in a significant decrease in annual energy output. Use of n-butane reduces the size and cost of heat exchangers when compared to isobutene, but this effect is not big enough to compensate for the decrease in net power output.

The choice of suitable material for heat exchangers was also found to be the crucial part of the analysis as it has a significant impact on the cost of the system. In waste heat recovery applications, the use of cheaper heat exchangers, identical in dimensions, but manufactured from carbon steel allows the improvement of plant economics.

The optimal design for each particular application included the regenerator as a part of the system. Despite the fact that the majority of commercially available waste heat recovery units do not incorporate this component, the performed analysis has shown its usefulness. In geothermal power plants the use of a regenerator allows a reduction in the size of the preheater and evaporator. Since these two heat exchangers are manufactured from expensive stainless steel, the effect of such a change on the cost of the unit is significant. Despite reducing the net power output of the geothermal plant, the regenerator allows a reduced cost of electricity. Thus, it may be concluded that as long as the preheater and evaporator have to be manufactured from more costly material than the regenerator, the main reason for its use is economics. For waste heat recovery applications the levelized cost of power achieved by an optimally designed unit is lower by over 0,5% than for the best design without the regenerator. In the same way as in the geothermal unit, regeneration allowed the increase of the thermal efficiency of the cycle. Moreover, in the recovery plant the negative feature of the regenerator - decreased efficiency of heat recovery - has a limited impact. Since thermal oil circulates in a closed loop, exergy of the stream hs3 leaving preheater is not lost and recovery efficiency has a minor effect on the cost rate of fuel supplied to the preheater and evaporator.

Although it was shown that the designed standardized unit is capable of producing electricity at a low cost, its usefulness is limited due to the high sensitivity to boundary conditions. However, if binary power plants keep on growing in popularity as fast as they have over the last few years, high market demand will promote standardized units. With a few units designed for different heat source characteristics, the impact of the off- design working conditions will be minimized but the cost reduction due to series production will remain.

Not only has this thesis reached meaningful results, it also proved the usefulness of exergy-aided cost optimization in the analysis of geothermal power plants. The fact that its positive impact was minimized by a very good initial guess as to the optimal size of heat exchangers does not change the overall picture. Thermoeconomic analysis did not improve the efficiency of the system, yet, taking profitability as a measure of performance, it allowed a minimization of the cost of the final product of the system.

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