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Design of a geothermal binary system for Operation in Remote Areas.

A Study Case in Hveravellir (Kili), Iceland, and comparison with Conditions in Southern Poland.

Marcela Bagierek

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A 30 ECTS credit units Master´s thesis

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ABSTRACT

In this thesis, a feasibility study of power generation from a geothermal source at Hveravellir, Iceland, was conducted applying a custom designed Organic Rankine Cycle (ORC) binary system that can utilize geothermal fluid from a newly drilled well in the area.

Natural water cooling was applied, due to the presence of a cold well on the site.

Working fluid selection was made from six various choices. The best solution appeared for R245fa and isopentane, with the latter rejected due to its bad influence on the environment.

There were three main factors, which were searched during optimization process, in order to fulfill the estimated demand of 40 kW, at the lowest specific cost. Finally, optimal values of vaporizer pressure, vaporizer and condenser heat exchange areas were estimated.

1^{st} and 2^{nd} law assessment of the proposed power plant was performed.

For the designed binary geothermal power plant, exergy and economic analysis were investigated. The economic analysis was very basic, but it showed that the binary system will start to pay off in six years.

Finally, performance of the unit, designed for operation at Hveravellir, but operating in Southern Poland conditions was investigated. A comparison between the different reservoir features was made and indicated what should be done with the Hveravellir design so that it fits the selected geothermal conditions in Poland.

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1 INTRODUCTION

1.1 Binary power market

Low-temperature geothermal reservoirs, with temperature ranges between 100 – 200 ^{o}C, are the most abundant in the world. Therefore, a proper utilization of these resources is a huge opportunity for future power production and heating systems. Binary systems allow exploitation of those fields as well as industrial waste heat. Their main function is to produce electricity, but additionally there is the possibility of using rejected water for heating (i.e. co-generation or combined heat and power processes). Besides, binary plants have no dangerous emissions to the environment, apart from water vapour (if using wet cooling towers) compared to high temperature coal and oil fired systems (greenhouse gases e.g. CO_2, CH_4). Finally, the geothermal fluid (brine) does not have a direct contact with power plant components other than the heat exchangers, which allows binary units to last longer than most geothermal flash turbines.

So far, binary systems are common systems in the world (44% of installed units).Table 1-1 presents binary units per country prepared in 2010 (Bertani, Geothermal Power Generation in the World 2005-2010 Update Report, 2010).

Table 1-1 Binary unit per country in 2010

Country	MW	Unit
Australia	1	2
Austria	1	3
Costa Rica	21	2
El Salvador	9	1
Ethiopia	7	2
France	2	1
Germany	7	3
Guatemala	52	8
Iceland	10	8
Japan	2	2
Kenya	14	3
México	3	2
New Zealand	137	24
Nicaragua	8	1
Philippines	209	18
Portugal	29	5

Turkey	14	2
USA	653	149
TOTAL	1178	236

Total production from binary systems in 2010 was 1178 MW from 236 units. USA has the biggest number of units (149) with about 653 MW installed capacity. The majority consists of units with low capacity (up to 10 MW). A binary unit is mainly classified as a small power plant, due to only 5 MW/per unit on average and energy of 27 GWh/per unit, on average. Therefore, such plants account for only 11% of total installed capacity in geothermal and about 9% of produced energy, respectively. Flash steam systems are also utilized in projects with low power capacity, which is mainly from low and medium enthalpy reservoirs. There are about 257 units of both flash and binary types with capacity less than 10 MW in operation. They have got an average capacity of 3.2 MW. Most of them are binary type (196 units), 22 are back pressure, 22 are single flash and 17 double flash (Bertani, Geothermal Power Generation in the World 2005-2010 Update Report, 2010). Comparing 236 units in 2010 to 162 binary running units in 2007, it can be noted that a huge expansion is taking place in this field. Since 2005 there were 111 new binary systems installed out of a total of 146 additional systems installed up to 2010, which constitutes about 76%. Figure 1-1 and 1-2 present changes in shares of various types of power plant units between 2005 and 2010 (Bertani 2005, 2010).

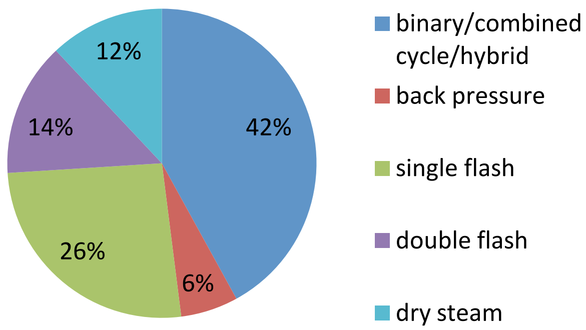

Figure 1.1 Number of units in 200

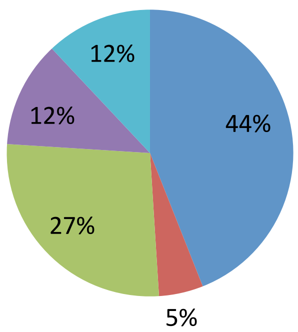

Figure 1.2 Number of units in 2010

Between 2005 and 2010 there were 465.8 MW of new capacity in binary systems installed. It corresponds to about 26 % of the total growth in geothermal power production worldwide over this period, which was 1,782 MW. The most pronounced installation of binary plants, 183 MW was in 2009 (Bertani, Geothermal Power Generation in the World 2005-2010 Update Report, 2010).

New plants list of binary system between 2007 and 2009 is shown in Table 1-2.

Table 1-2 Binary new plants list

Country	Plant	Unit	Year	Capacity[MW]
USA	Faulkner	1	2009	50
USA	Stillwater	1-2	2009	48
USA	Salt Wells	1	2009	24
USA	North Brawely	1-7	2009	49
USA	Thermo Hot Spring	1-50	2009	10
USA	Galena III	1	2008	30
New Zealand	Ngawha 2	1	2008	15
USA	Raft River	1	2008	13
USA	Heber South	1	2008	10
El Salvador	Berlin	4	2008	9.4
New Zealand	KA24	1	2008	8.3
Turkey	Kizildere Binary	1	2008	6.8
Germany	Unterhaching	1	2008	3.4
Germany	Landau	1	2008	3
France	Soultz-sous-Forệts	1	2008	1.5
Guatemala	Amatitlả n	1	2007	24
New Zealand	Mokai 1A	1	2007	17
USA	Galena II	1	2007	13
USA	Blundell I	2	2007	11
USA	Desert Peak II	1	2006	23
Portugal	Pico Vermelho	1	2006	13
Turkey	Dora	1	2006	7.4
USA	Gould	1-2	2006	10
Japan	Hatchobaru	3	2006	2
USA	Richard Burdett	1-2	2005	30
New Zealand	Mokai 2	1-5	2005	20
New Zealand	Wairakei Binary	15-17	2005	14

Resources with temperatures lower than 130 ^{o}C comprise 68% of total geothermal energy. They have potential of about 4400 GW_{th} which is equivalent to 139 EJ per year (Stefansson, 2005). Nowadays, the development of a geothermal is focus mainly on the binary systems. The development over the last few years, show that this field is currently very active. The biggest installations for the other types of systems took place a long time ago, the largest single flash unit was installed in 1997, the largest double flash unit in 1986 and the largest dry steam unit in 1985. The short term forecasting for 2015 expects 18 GW of world geothermal electricity. It is assumed mainly due to an increase in the medium-low temperature development projects through binary utilization and realizing all the economically viable projects worldwide (Bertani, Geothermal Power Generation in the World 2005-2010 Update Report, 2010). There is a plan to carry out 7 GW of new plant installations based on currently existing paper-projects during next five years. The ambitious target of 70 GW before 2050 has been put forward (Bertani, Geothermal Power Generation in the World 2005-2010 Update Report, 2010).

The main geothermal ORC power plant manufacturers in the potential small scale geothermal market are from the United States, Japan, Iceland and New Zealand. Japanese Mitsubishi and Toshiba manufacture turbines for back pressure systems. The biggest leader in the binary market is ORMAT Company from Israel (about 92 % of produced turbines for binary). They are operating mainly for United States, but also for the other countries. Currently they have got projects in Guatemala, Kenya and Nicaragua. Important brands which were providing geothermal equipment during last years were ORMAT, Mafi Trench (US), UTC/Turboden (US/Italy) and Enex (Iceland) (Bertani, Geothermal Power Generation in the World 2005-2010 Update Report, 2010) (Kaplan, 2010). There are also some smaller suppliers such as Siemens (Germany), Barber-Nichols Inc. (USA) and Peter Brotherhood (UK). Table 1-3 shows list of geothermal turbine manufacturer for binary system according to Bertani. Currently, Mafi Trench is a branch of Atlas Copco, Turboden is owned by Pratt & Whitney (a UTC company) and Peter Brotherhood merged with Dresser-Rand.

Table 1-3 List of the geothermal turbine manufacturer for binary systems

Manufacturer	Country	Number of turbine
ORMAT	US	1 074
Mafi Trench	US	72
UTC/Turboden	USA/Italy	13
Enex	Iceland	1
Siemens	Germany	4
Barber-Nichols Inc.	USA	2
Peter Brotherhood	UK	1
TOTAL		1 167

1.2 An Island Mode

There are many places around the world, where there is no access to the electricity grid and the power demand is usually supplied by diesel engines. Usually, it is connected with low per capita electricity demand. It concerns mainly rural, remote areas or small islands where a grid connection meets with many obstacles, comparing to a mainland or big islands. In such places, there is a need of a system, which will provide reliable power, ease of use and guaranteed maintenance even under harsh conditions. An Island Mode is an off-grid power plant unit which is adapted to supply such low demand. Therefore it is connected mainly with binary or flash systems. Flash systems utilize higher enthalpy reservoirs and binary mainly use lower ones.

In many places such as volcanic islands, rural areas like in Tibet or Kenya there are existing geothermal possibilities to supply such a power demand. Table 1-4 presents performance of some small geothermal power plants.

Table 1-4 Small geothermal projects

Country	Project name	Capacity [MW]	Type
Azores archipelago	Ribeira Grande	2.5	Binary
(Sao Miguel Island)	Pico Vermelho	12.6	Binary
Hawaii archipelago	Puna	33	GCCU*
(Island of Hawaii)			
United States	Wineagle	0.6	Binary
(California)			

** Geothermal Combined Cycle Unit*

There is no available overview about island mode ORC units that are currently running. These individual power plants can compete with batteries and photovoltaic systems and in mini-grids with diesel generators, natural gas turbines or mini-hydro. They have many advantages comparing to the other renewable power systems. First of all, geothermal binary units are usually very reliable and have availability levels of 96% and higher (Kaplan, 2010). If we consider off-grid, small units it is usually even better due to a better protection treatment and high level of redundancy. Small geothermal power plants can be a good option for off-grid areas and reduce their dependence on small diesel power generators. They are preferable mainly due to their minimal environmental impact. If the overview of islands or huts in highlands will be carried out, most of them are based on tourism. Noise and pollution issues have to be especially minimized in those cases. Besides, equipment should be installed in a way to be not so visible, which is quite easy in geothermal power applications (pipe can be matched with the environment) and it is not taking much terrain comparing to solar or wind technologies. In islands, like in Indonesia, reduction in utilization of diesel generators can conserve oil for export. Geothermal island mode systems are also more economical in the long run compared to diesel engines. Although capital costs are much higher, life time of a single modular unit is estimated to be over 25 years, which makes it cost-effective. Binary units also have lower maintenance requirements.

Power availability of mini geothermal power plants is quite high. For example in Indonesia, they can usually operate in excess of 22 hours per day comparing to 6-7 hours per day of diesel power plants. If back pressure turbine will have been used in such a system, they would commonly operate more than 8,000 hours per year, thus can deliver source of base load power (Saragih & Radja, n.d.).When selecting an island mode system it is important to consider variable load. The system has to work over a wide range, about 10% under and over the most frequent demand. It should be protected from unexpected shutdowns. To provide good availability there is a need to have design spares available. Commonly up to 20% extra capacity assumes to prevent the system from fluid shortfall.

Another major design parameter is the choice of the cooling system. Selection between cooling water and ambient air is mainly based on ambient average conditions.

Although there are plenty of small geothermal power plants installed, their utilization always needs a huge financial support. Strong government or regional policies' promotion of this applications are at the top of the list to make a project successful to introduce. There have to be subsidies and not so high cost of maintenance, because in some cases the customers have low incomes. Moreover, to be competitive at the market they have to go down with costs, which can be achieved by the reduction of drilling cost. There is an opportunity in the technology of slim-hole drilling, with a diameter less than 6'' (conventional hole - 8''), which is enough to supply sufficient fluid to a small plant. Smaller well diameters require smaller rigs, which results in lower costs, too. However, binary systems require higher geothermal flow rates compared to flash systems, to obtain the same electric power output. Additionally, larger installations will result in re-injection, which also cannot be bypassed in a cost analysis.

2 BINARY CYCLE

Binary systems (or Organic Rankine Cycle units) are mainly utilized in low and medium temperature reservoirs. The geothermal water transfers heat to the working fluid through a heat exchanger. The schematic flow diagram for a basic binary system is given in Figure 2.1. The thermodynamic processes undergone by the working fluid are shown in Figure 2.2, on a temperature-entropy and pressure-enthalpy; T-S and P-h diagrams. A source in this kind of a system can also be waste heat from industrial applications. The three different types of binary power cycles are distinguished below:

- Saturated vapor cycle
- Superheated vapor cycle
- Supercritical vapor cycle

A binary cycle power plant combines two main subsystems:

- Heat recovery cycle - HRC (with recovery heat exchanger - RHE)
- Cooling system - CS

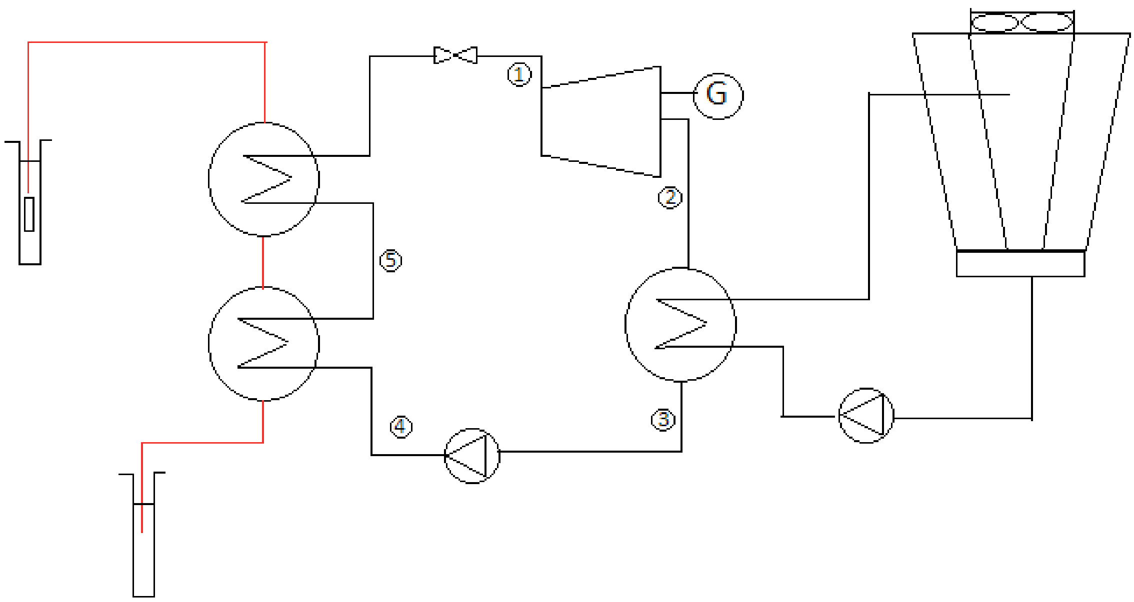

Figure 2.1 Schematic diagram of basic binary geothermal power plant

Firstly, the brine has to be pumped from a well and flowed to the heat exchanger unless there is sufficient artesian (pressurized) flow.

The outlet of the feedpump transfers heat to the secondary fluid through preheater and evaporator, to be finally rejected back to the reservoir. In the preheater it goes to the boiling point (5). Then, while in the evaporator, the working fluid heated to become a saturated vapor at the exit of the heat exchanger. Pressure of the geofluid is constantly above its flash point for fluid temperature to prevent from scaling. Afterwards, the vapor goes to the turbine (1), where useful power is generated.

Next, there is a need to cool down superheated steam in the condenser (2-3), which can be done by the wet or dry cooling system. After leaving the condenser (3), the working fluid is pumped up to the pressure P_4 (4) and can go to the regenerator or straight back to the preheater.

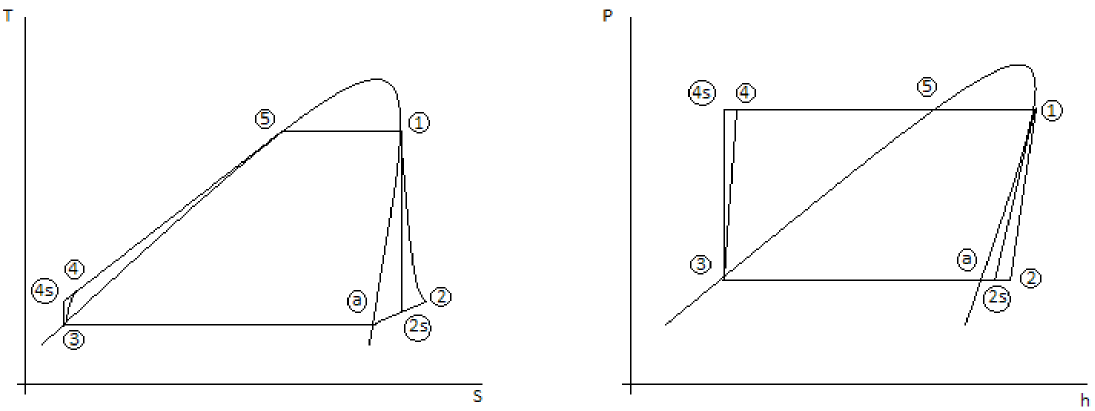

Figure 2.2 Temperature-entropy and pressure-enthalpy diagrams for a binary power plant

The thermodynamic cycle presented in Figure 2.2 can be divided into the following steps:

- 4-5 isobaric heat transfer in the preheater
- 5-1 isobaric and isothermal evaporation in the vaporizer
- 1-2s isentropic (assuming ideal cycle) expansion in the turbine
- 2-3 isobaric condensation in the condenser (with isothermal process between a and 3 state)
- 3-4s isentropic (assuming ideal cycle) compression in the pump

2.1 Turbine

The turbine converts potential energy of high-pressurized saturated vapor of the working fluid into kinetic energy, which rotates the internal parts of the turbine. Electricity is generated by the generator, which is connected to the rotor by a shaft. The power production undergoes with assumptions of steady state and adiabatic operations, and is equal to (DiPippo, 2008):

(1)

where is the isentropic turbine efficiency, known from the manufacturer's specifications, is a mass flow rate of a working fluid, h_1 and h_2 are enthalpy at the inlet and the outlet of the turbine (according to Figure 2.2) and is the enthalpy at the exit of the turbine, for the ideal isentropic process.

2.2 Condenser

The purpose of the condenser is to take the waste heat from the turbine exhausts and pass it to the cooling medium (water or air). This is a very important issue in a geothermal power plant as long as there is a very substantial amount of rejected heat per unit of the electricity output, due to the low thermal efficiency( ). Required dissipated heat is equal to:

(2)

where is a heat inflow with geothermal brine into a system

is a thermal efficiency of a power plant

It can be also calculated as follows (DiPippo, 2008):

(3)

where is enthalpy at 2nd and 3rd state (Figure 2.2)

The condensing temperature is usually in the range of 3 to 6 °C above the average temperature of the cooling water used as the heat sink or about 8 °C above the average dry-bulb temperature of the ambient air if dry cooling towers are used (Kestin, 1980). The relationship between cooling water and the working fluid flow rates is found from the following formula (DiPippo, 2008):

(4)

where, stands for mass flow rate of cooling medium

is enthalpy of cooling medium at state c1 and c2 (Figure 2.2)

2.3 Heat dissipation system

In general, there are two type waste heat rejection systems. They can apply:

1) Cooling towers
2) Direct water

In a range of cooling towers, wet (the heat rejected into cooling water) or dry cooling systems (the heat is rejected directly to ambient air) can be distinguished. A choice between them is based on the ambient conditions and environmental availability (if there is a cold water source). The latter is usually more preferable, due to lower costs but the former is more general.

The wet cooling system has got water as a cooling medium. Energy transferred to water in a condenser is taken by the air stream in the cooling tower. It is important to keep temperature of the cooling water above the freezing point (0°C), since ice expansion will destroy the cooling tower construction. Therefore cooling tower has a factor noted as an Approach. Pursuant to definition of approach as a difference between inlet temperature of cooling water into condenser and wet-bulb ambient temperature (measured by the thermometer freely exposed to the airstream; shielded from radiation and humidity), it ensures proper work of a cooling tower. It will protect water from congeal. It is worth to mention, that costs of the cooling tower for wet systems is relatively high comparing with remains power plant components. It can be estimated to 40-50 $ per each 1kW of dissipated heat (Valdimarsson, P., personal conversation).

The dry cooling system uses air as a cooling medium. It rejects heat from fluid straight to ambient air, in air-cooled heat exchanger, thus does not required water. However, air has four times lower heat capacity than water. For big power plants, which need to reject huge amount of heat in a condenser, it is not the best solution.

In both cases, fans are required which will force air stream to flow into the cooling tower or condenser. Cooling tower fans can consume between 10% and more than 30% of the gross power (Franco & Villani, 2009). Consumed power is given by:

(5)

where is a volumetric flow of air through the fan, is the pressure increase over the fan and is the efficiency of the fan and its electric motor.

2.4 Feed pump

The power required to pressurize the working fluid in the feed pump up to P_4 indicated in Figure 2.2., is obtained from:

(6)

where is the isentropic pump efficiency, h_{4s}, h_3 is enthalpy at states according to Figure 2.2 and is mass flow rate of working fluid.

2.5 Heat exchanger

A typical heat exchanger for most popular, saturated vapor cycles contains a preheater and a boiler (or evaporator). Basic assumptions made for heat transfer through such exchangers are:

1) Steady state operations
2) Pure concurrent or countercurrent flow along all tubes
3) Overall heat transfer coefficient is constant
4) Constant specific heat
5) Negligible heat loss

Conveyed energy stream is derived from:

(7)

Considering above assumptions, the governing equation is:

(8)

(9)

An analysis of the heat exchangers separately for both preheater and evaporator provides these formulas:

Preheater:

(10)

Evaporator:

(11)

where subscripts 'b' refers to the geothermal brine, 'wf' to the working fluid and 'pp' to the pinch point. The pinch point is the place, where the lowest temperature difference between two fluids occurs.

The primary relationship, which allows finding the heat transfer surface area is as follows (Kestin, 1980):

(12)

where is the total overall heat transfer coefficient (estimated by experiment with appropriate fluids to be used in the plant) and is the heat exchanger's surface area, referring to the surface on the outside of the tubes in the heat exchanger.

Logarithmic mean temperature difference ( ) can be obtained using (Kestin, 1980):

(13)

where

– Logarithmic mean temperature difference

GTTD – Greater terminal temperature difference

LTTD – Lesser terminal temperature difference

Given the fact that the logarithmic mean temperature difference is a small number and the overall heat transfer coefficient is also a small number (both U_0 and T_{LM} in equation 13 are small values) it is clear that if large amounts of heat is to be transferred, this will require a binary geothermal power system to have a large heat transfer surface area.

2.6 Environmental impact

A geofluid is usually rejected to the reservoir just after passing through the heat exchanger and the secondary fluid is in a closed cycle, therefore there is no harmful influence on the environment. Due to that, binary power plants with reinjection are included as the most environmentally friendly plants out of all thermal power plants. Contaminations which can occur are only at the heat rejection side, from the cooling system. It is important that in geothermal power plants there is more waste heat per power output than in the other systems. Almost nine times the electric power output is going to the surroundings as thermal waste, as discussed in the next section.

2.7 Efficiency

Binary plants have rather low efficiency as generally all geothermal systems. The First Law[1] efficiency for ORC systems varies between 5-10%, and Second Law[2] efficiency in the 20-45% range (Franco & Villani, 2009). Rejection temperatures and cooling systems have the biggest influence on the efficiency. Small difference between brine goes in and out of the system, results in a low efficiency Carnot cycle. Geothermal temperature depends of the reservoir conditions and rejection is mainly estimated in way to avoid scaling problems (high enough to protect from silica oversaturation). Based on the analyses presented in the literature, it seems difficult to lower the rejection temperature below 70^{o}C (Franco & Villani, 2009). Choice of a wet cooling system relies on the availability of cold water, otherwise one has to use a dry cooling system. Latter option has higher parasitic losses, due to huge power consumption by cooling tower fans.

2.7.1 State of the art

There are different types of innovative and complex binary cycles. The main goal of development in this field is to get better performance of the heat recovery cycle. Opportunities seem to be in fields of:

1) Dual-pressure binary cycle
2) Dual-fluid binary cycle
3) Kalina binary cycle

The aim of the two firsts cycles is to decrease the average temperature difference between the hotter brine and the cooler working fluid. The dual-pressure has a two-stage boiling process and dual-fluid two Rankine cycles, supercritical and subcritical, with different hydrocarbons used. At present, there exist few power plants with dual-pressure levels (12.4 MW at Stillwater, California, US, and 40 MW at Heber, California, US), (Franco & Villani, 2009). In all cases the thermal efficiency of this kind of binary system is lower than for a standard type, but the utilization efficiency is higher, ranging from a 6% advantage at the highest brine temperature to 24% at the lowest.

[1] First law of thermodynamics
[2] Second law of thermodynamics

The first commercial binary power plant running on dual- fluid cycle principles was the Magmamax plant at East Mesa in California's Imperial Valley. Thermal efficiency for the supercritical vapor generator is higher than for the subcritical generator. However, it is also important to consider that the required pump work in the supercritical phase highly exceed the corresponding work in the subcritical cycle (about 45%).

The main feature of the Kalina binary cycle is using water-ammonia mixture as a working fluid. More heat is transferred due to variable evaporation and condensation temperature, which results in a very tight pinch-point temperature difference. There are currently 5 Kalina projects around the world, which are presented in Table 2-1.

Table 2-1 Kalina power plants in the world

Country	Project name	Capacity [MW]	Type of plant
USA	Canoga Park	6	Demonstration
Japan	Fukuoka	4.5	incineration
Japan	Sumitomo	3.1	Waste heat recovery
Iceland	Husavik	2	geothermal
Germany	Unterhaching	3.4	geothermal

There are two geothermal Kalina systems installed. The station in Germany was installed by Siemens but it seems that Siemens will not contribute to further development in Kalina technology. High investment costs and unfamiliar technology makes it less common on the market and difficult to compete with cheaper ORC units.

3 HVERAVELLIR AREA

Hveravellir is a geothermal region near one of the main interior roads (Kjölur), in the center of the Icelandic highlands, between the glaciers Langjökull and Hofsjökull, 30 km to the north of Kerlingarfjoll (Figure 3.1). The area is enclosed by Langjökull in the west, Kjölur highlands in the southeast and Kjalhraun lava to the south.

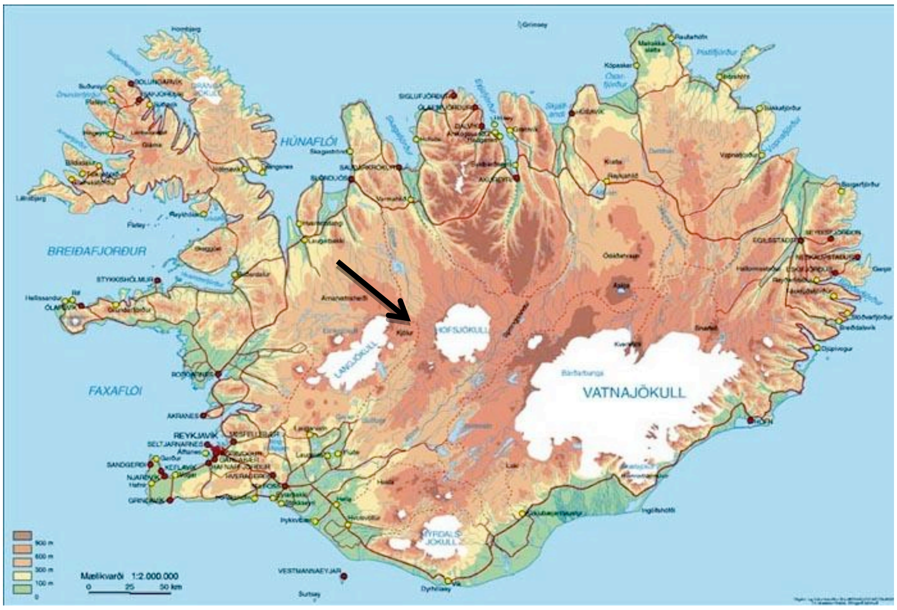

Figure 3.1 Iceland (http://www.icelandprivatetours.is/home/map/)

The total geothermal region is spread over a 20 km^2 area. It is 14 km long and 1-2 km wide. The high temperature region is only a small part of it, extending 5 km and 0-5 km wide, giving a total of about 2.5 km^2. It is located at an altitude of 630 m, as generally most of the interior area. On the northern edge of the post glacial lava field of Kjalhraun (a lava shield), on a glacial outwash plain, the main hot springs are located, although fumarolic activity also occurs in the lava field. The thermal field in the glacial outwash plain indicates that in the neighborhood there are basaltic hyaloclastites and interglacial lava. The main geological structures emerging in this area are rhyolite, interglacial lavas with hyaloclastite, moraines with glaciofluvial sediments, and recent lava. Thermal activity occurs there as steam geysers, mud pools, boiling springs, hot springs, fumaroles and clay rich hydrothermal soil. Geysers are still active, but erupt irregularly. Water in springs is alkaline. Deposits are visible, created by water with high silica content.

The mountain huts have been there for a long time. The oldest is from 1938. The weather observatory was running there between 1963 and 2003. About 30 000 people annually visit Hveravellir. Currently there are two buildings at the geothermal area, which serve as accommodation for tourists.

3.1 Present utilization of geothermal

There is one hot spot, used as a swimming pool for tourist. It is in a warm stream below the Old Lodge. It is about 6x3.5 m, built up with concrete and rocks with flat rock, at the bottom. At its deepest point it is 1.35 m. Some of the water is piped from the stream at a rate of 0.9 l/sec. The temperature of the pool is between 18.6 °C and 39.3 °C, and the water in the intake pipe is 80 °C (Snaeland & Sigurbjörnsdottir, 2010). Over 20 people can comfortably bathe in the same time at pool

Additional, there is a direct geothermal utilization system for heating in three cabins at the field. They include the 'old cabin' (Gamli skáli), the 'new cabin' (Nýji skáli) and the 'staff house' with following areas 80 m^2, 90 m^2 and 50 m^2 respectively. Besides, hot water is used in 3 showers. Geothermal water is transported from one of the hot springs, located 150 m in the northwest of the old cabin, by a 6" tube to heat exchanging pits next to this cabin. There are two heat exchanging loops in pits supplying heat to the old and new cabins. The old one gets heat delivered only to the ovens and the latter to the ovens, showers and taps. Loops are closed with circulating water inside. The system in the old cabin does not need a pump to circulate the water, compared to the new cabin, which is situated further from the pits. Therefore, two 70 W circulating water pumps are used in the larger loop.

3.2 Wells at Hveravellir

Hveravellir region is located in a catchment area of Seydisa River, which is Blanda's glacial river tributary. The aquifer is situated in interglacial lava and hyaloclastite formations. Temperature of the ground water is 2.5°C up to 4.5°C. There was an investigation carried out to the north of the main area at Hveravellir, which led to the discovery of a new geothermal field. Currently there are four wells at Hveravellir, which all are outside the protected area in and to the north of the main field. Location of most of them is presented on Figure 3.2. Well VE-01 is the old well with cold water, which provides drinkable water to the huts. It was drilled in 2005 on the east side of the tourist houses, to 24 m depth and reaching a temperature of 20 °C. In 2008 it was deepened to 102 m. Well VE-02, was drilled in August 2008 for heating purposes inside the weather station area (it is not included in the map). It has a depth of 204 m and 71°C at the bottom, which is a lower temperature than was expected. Well VE-03 is a well with cold water of 5°C and was drilled in 2010 to a depth of 21 m. The newest well is VE-04 with hot water, which is described in more detail in chapter 3.3.

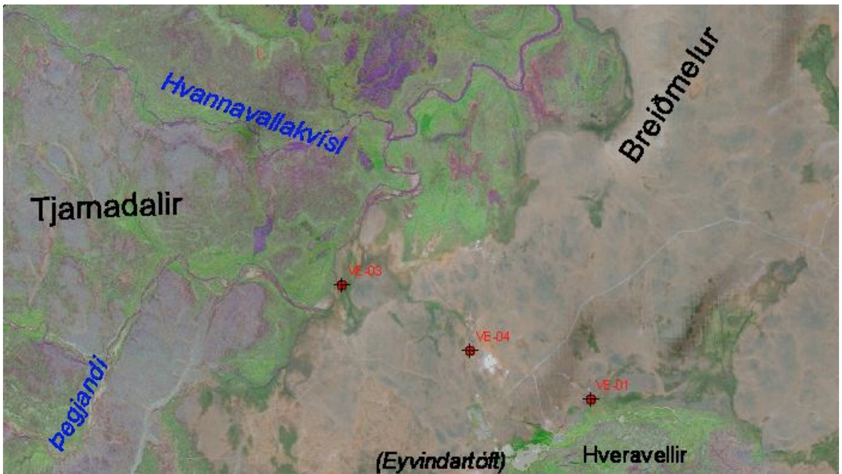

Figure 3.2 Wells at Hveravellir area

3.3 The new well

At the end of the August 2010, Árni Kópsson drilled a hole to 88 m depth using a 5.5'' drillbit; well VE-04. The first 30 m are cased with a 6" casing. The drilling team was expecting the reservoir temperature to be around 120 °C. After drilling and well recovery, the initial measurement of temperature and pressure on the 9th of September 2010 recorded 150 °C with 4.9 bar head pressure in a completely dry steam outlet. The well performance is presented in Figure 3.3.

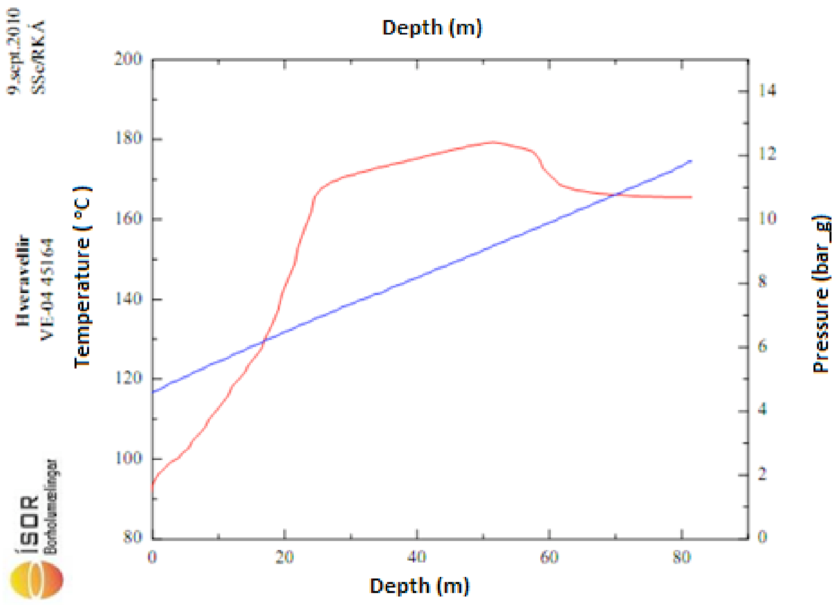

Figure 3.3 Temperature and pressure changes with depth for VE-04 well

A proper flow test still needs to be made, which will qualify the steam production from the well once it has fully stabilized. Results of the chemical analysis from VE-04 are presented in Table 3-1. Samples were taken at 140 °C.

Table 3-1 Chemical analysis for VE-04

Formula	Value Sample 1	Value Sample 2	Unit
Ca	3.1	3.03	mg/l
Cl	52.5	52.7	mg/l
F	2.67	2.68	mg/l
Mg	0.019	0.018	mg/l
SO_4	109	110	mg/l

3.4 Current system for electricity production

Currently, diesel generators are used to supply electric power at Hveravellir. During the summer time (June - September), when there are many people visiting this area, they use a 20 kW diesel engine. It is running 24 hours, 7 days a week. It consumes about 50 l/ 24 hours. In the winter time (October – May), mostly a 5.5 kW generator is in operation (only over some tourist intensive intervals, the bigger generator is turned on). It is working 10 hours per day, on an average. Consumption is equal to 300 l/ 30 days. Table 3-2 shows diesel usage in each season with total costs. It was assumed that 1 l of diesel costs $ 1.161 (Guðjónsson, G., personal conversation).

Table 3-2 Diesel consumption and costs

	Consumption / 30 days [liters]	Costs / 30 days [$]	Costs / season [$]
Summer	1500	1,742	7084
Winter	300	484.8	3927
Annual			11011

Amount of power used over a year, with distinction in each season is presented in Table 3-3.

Table 3-3 Power demand [kWh] over a year

	Operation time [hours]	Numbers of days	Engine type [kW]	Power demand [kWh]
Summer	24	122	20	58560
Winter	10	243	5.5	13365
Annual				71925

Table 3-3 clearly shows that the highest demand is at the summer time, about 80% of annual. Over one year, huts at Hveravellir need to be supplied with 72 MWh, which constitutes a relatively low percentage of the energy that can be supplied with constant peak power.

3.5 Steam engine for electricity production

As long as there is pressurized dry steam coming out from the well, there is an opportunity for using it directly in a steam engine to get power. If the next tests at well VE-04 will show that the steam is stable, this system will be able to operate. According to existing site conditions: 5 bar pressure, 150 °C temperature, 1.5 kg/s of mass flow rate and 20-50 kW of required demand, a preliminary scheme of a such a system was made, using steam engine for electricity production. It was made based on a conversation between ISOR (presented by Ragnar Ásmundsson) and Energent Company (presented by Phillip Welch). Due to the very low demand, the best solution would be production of electricity not only for Hveravellir, but also for users close to this location, to make it more economically profitable. The smallest available unit from Energent is 275 kW_e, which can achieve about 155 kW_e at existing conditions. Turbine outlet pressure is 1.1 bar, slightly higher than atmospheric pressure, which allows skipping condensation of the exhausts, in order to vent it straight to the surrounding. This solution is cheaper since no condensation costs occur. Hveravellir as a remote area and does not have any possibility to connect the generator to the grid. Therefore, a synchronous generator has to be used with the Energent's Turbine. Power control should be down to a load of less than 10 kW. This kind of installation (Microsteam Turbine with synchronous generator) would cost $325 000 according to Energent Company.

4 BINARY SYSTEM FOR ELECTRICITY PRODUCTION

There is a possibility to use geothermal water for an Organic Rankine Cycle for electric power production at Hveravellir. Although this area is considered mainly as a hot temperature area, there is lower temperature in the northern part, which can supply cooling to this type of a power production system.

4.1 Boundary conditions

It is very important to use properly the thermal features of the existing source and ambient conditions on site. The system can perform much better if the best available cooling system will be chosen. Mass flow rate properties (if it is gas or fluid) will result in different type of heat exchangers, as well.

Weather condition

The climate at Hveravellir area is rather harsh. Therefore the whole unit should be installed inside a building. It cannot be open, due to too low temperatures and amount of snow during the winter, which could destroy or impact the workings of the system.

Heat source

Well VE-04 was chosen as the heat source for a prospective binary system. It contains geothermal water with 150 °C and 4.9 bars pressure. The mass flow rate received from the well is about 1.5 kg/s from an 8 mm nozzle. During the first temperature and pressure measurements, dry steam was registered at the outlet.

4.2 Cooling system

Choice of the cooling system is made according to weather conditions at the field. Hveravellir is located in the highland in the middle of the country; therefore the ambient temperatures are rather low. The average temperature is equal 0.9 °C. The lowest is -19.2 °C and the highest 20 °C. Therefore, a dry cooling system, using air as a cooling medium could perform well.

However, Hveravellir has to be considered as a place with a natural cooling system either from air or cold water, which is present in this area. The new well VE-03 with a lot of cold water at a temperature of 5 °C can be applied for cooling in the ORC cycle. If the source of cold water in the well will be assumed as endless, water leaving the condenser could stay in a kind of natural pond and rejoin the ground water system through fractures which are abundant at the field. Then, there would not be no need of any additional components for the cooling system, thus the power plant cost would be lower.

In the other way, if there would not be any well with cold water, a cooling pond could be created. It would be possible because there is available sufficient land at Hveravellir. In a cooling pond thermal energy received from inflowing water is mainly dissipated through evaporation to the atmosphere. Remaining water is cooled down, at a rate depending on the pond’s depth, and can be used again from the bottom as a source with required cold water.

4.3 Assumptions for the model

Assumptions which were made to run a model calculation are the following:

1) Efficiencies (Valdimarsson, P., personal conversation)

   - Isentropic efficiency of turbine η_t= 0.8
   - Isentropic efficiency of working fluid pump η_p= 0.8
   - Efficiency of electric generator η_{gen}= 0.97

2) Pressure loss for working fluid (Valdimarsson, P., personal conversation)

   - Pressure loss on heat exchanger P_{loss_he}= 0.15 bar
   - Pressure loss in pipe line P_{loss_pl}= 0.1 bar
   - Pressure loss at tripvalve $P_{loss\_tripvalve}$= 0.5 bar

3) Heat capacity for water at heat source and cooling system c_p= 4.186 ——
4) Overall heat transfer coefficients (Valdimarsson, P., personal conversation)
   - $U_{preheat}$= 1 —— for preheater
   - $U_{boiling}$= 1.2 —— for boiler
   - $U_{desuperheat}$= 0.5 —— for desuperheating part of condenser
   - $U_{condensation}$= 1.2 —— for condensing part of condenser
5) Pressure at cooling fluid source P_{c1}= 10 bar
6) Rise of the cooling water in the condenser ΔT_{cond}= 15°C

4.3.1 Working fluid

Proper selection of the working fluid has an important influence on a binary power plant performance. Environmental, health and safety properties have to be considered, beyond power and economic side, for different fluids.

All prospective fluids have critical temperatures and pressures far lower than water. It allows them to boil at relatively low temperatures, obtained after heat transfer from the geofluid. As a result, it is also feasible to consider a supercritical cycle for these fluids. Mainly hydrocarbons and refrigerants are considered due to their various thermodynamic properties. A mixture from these fluids can even result in a better match with the cooled brine curve. The latter will not be study in this paper due to program limitation.

Another important characteristic of candidate fluids is the shape of the saturated vapor curve on the temperature-entropy diagram. It can be negative or partly positive for different types. It results in a wetness, which can occur in the turbine. For the retrograde (positive) slope working fluid it is in the superheated region during expansion. Therefore, it does not require additional superheat before entering the turbine to avoid excessive moisture. It always provides superheated vapor conditions at the outlet of the turbine.

In the design process 6 different working fluids were checked, listed in Table 3-3. For comparison, water properties were also presented. All hydrocarbons listed in the table are retrograde type.

Table 3-3 Thermodynamic properties of candidate working fluids

Fluid	T_c [°C]	P_c [bar]
R134a	101.0	40.6
R227ea	102.8	30.0
R236fa	124.9	32.0
Isobutane	134.7	36.4
R245fa	154.1	36.4
Isopentane	187.2	33.7
Water	374.0	220.6

Properties such as flammability, toxicity, ozone depletion potential (ODP) and global warming potential (GWP) cannot be neglected and are shown in Table 3-4. Therefore some of the working fluids with good thermal properties are now illegal to use and some additional ones will be phased out in the next few years. The GWP is defined to be 1.0 for carbon dioxide. Additionally, isopentane and isobutane and they will require excessive fire protection on a site, over and above standard requirements for any power plants.

Table 3-4 Environmental properties of prospective working fluids

Fluid	Toxicity	Flammability	ODP	GWP
R134a	non-toxic	non-flam.	0	1,320
R227ea	low	non-flam.	0	2,900
R236fa	non-toxic	non-flam.	0	6,300
Isobutane	low	very high	0	3
R245fa	non-toxic	non-flam.	0	950
Isopentane	low	very high	0	3
Water	non-toxic	non-flam.	0	-

Considering performance of the working fluid, it is very important that it will be in a liquid phase at the dead state condition (ambient conditions) in a further design.

4.4 Model principle

A simulation of a binary system for Hveravellir was created in the EES (Engineering Equation Solver) program, with the former defined assumptions and processes carried out inside components. Free parameters of the system were the input areas of heat exchangers and pressure in the vaporizer, for a specific working fluid. Initially, 20 m^2 areas were assumed for each exchanger: the vaporizer (preheater and boiler) and for the condenser (desuperheater with condensing part). Pressure in the vaporizer was estimated at 20 bar in the beginning.

The simulation is based on 3 main procedures; a vaporizer procedure, a condenser procedure and a system procedure. The last procedure combines the two previous ones and connects system with physical relationships inside the working fluid pump. The simulation makes it possible to specify each state within the Organic Rankine Cycle and properties of the brine rejected from the system as well as rejected energy from the system exiting with the cooling water. The model work is based on the following steps:

1) Vaporizer procedure
   Procedure is based on calculation of area of the vaporizer (A_{calc}) according to assumed vaporizer pressure and thermodynamic features of heat transfer in a heat exchanger. Afterwards, calculated area of vaporizer is compared to assumed one (A_{vap}) in order to define the total heat transfer. At the beginning of the procedure, the maximum and minimum values of heat exchanged were defined. To have a sufficient range, the minimum is estimated as zero kJ and maximum equals the geofluid's total energy:

   (14)

   The inlet, bubble and outlet points were all defined. When each state is known, an EES procedure is able to calculate exchanged heat in the preheater and the boiler separately and get values of the bubble point temperature of the geofluid (T_{s2}), the temperature of rejected geofluid (T_{s3}), and the mass flow rate of the working fluid ( ). It runs by iterations, with different exchanged heat (, aimed to get A_{vap} and A_{calc} as close as possible. A range of a relative difference between the maximum and the minimum heat exchanged of less than 0.01 ‰ is assumed to be accurate enough. Finally the procedure reveals the calculated values of T_{s2}, T_{s3}, and .

2) Condenser procedure
   The condenser procedure works in a similar way as the vaporizer procedure, only with some minor changes. It takes from the previous procedure, over assumed area of the condenser at the input. Instead of variable exchanged heat, this procedure uses condenser pressure. Pressure of the working fluid, at the dew point (quality equal to 0) and the temperature equal to inlet temperature of cooling water (T_{c1}) is treated as the minimum, and at a temperature of 100 °C the pressure is considered to be at the maximum value. To define each state (inlet, dew point, and outlet) in the condenser, additionally isentropic efficiency of the turbine was used (allowed to specify the inlet state). The range is assumed as a relative difference between P_{max} and P_{min}. The procedure iterates to get the band lower than 0.01‰.
   Finally the procedure provides the condenser pressure, temperature of the dew point, enthalpy at the inlet and the outlet, and mass flow rate of cooling water.

3) System procedure

Combines results from the two previous procedures, by the process taking place in the working fluid pump. The enthalpy increase in the pump is as follows:

$$\frac{\quad}{\quad} \qquad (15)$$

where P_3, P_4 are pressures at the inlet and outlet of the pump, v_3 is volume of the working fluid at the inlet of the pump, and η_p is pump efficiency.

Number 101.3 is applied due to units changes from bar into kPa, because pressure unit in program was set as bar.

This connection allows having results for the outlet of the pump and the inlet to the preheater. Besides getting pressure and enthalpy for each state in the cycle, the goal of the system procedure is to estimate the best pressure and enthalpy especially at the outlet of the condenser/inlet of the preheater. It compares two types of these values. The first value is the dew point at the average cooling water temperature of the inlet and the outlet, which was chosen just to have some starting point. The second value received from the condenser procedure is the condenser pressure and the enthalpy at the outlet. Iteration continues until absolute difference between the first and the second type of values for pressure and enthalpy will be lower than 0.1‰. It makes the simulation of the system more accurate.
Consequently, this procedure gives pressure and enthalpy for each state, temperature T_{s2}, T_{s3} and at the dew point in the condenser (T_{dew}), mass flow rate of the working fluid and cooling water.

There was a function implemented for calculation of the logarithmic mean temperature difference (LMTD) in the condenser and the vaporizer procedure. It is made to avoid iterations when

(16)

For such values of temperatures, the function adds additional 0.001 °C to T_{c_out}, which does not cause too big difference, but dodges problems connected with dividing by ln0.

4.5 Optimization and analysis of the model

Schematic diagram of the model created in the EES program is presented in Figure 4.1 . Pressure at state point 5 is the calculated vaporaztion pressure (derived through iteration).

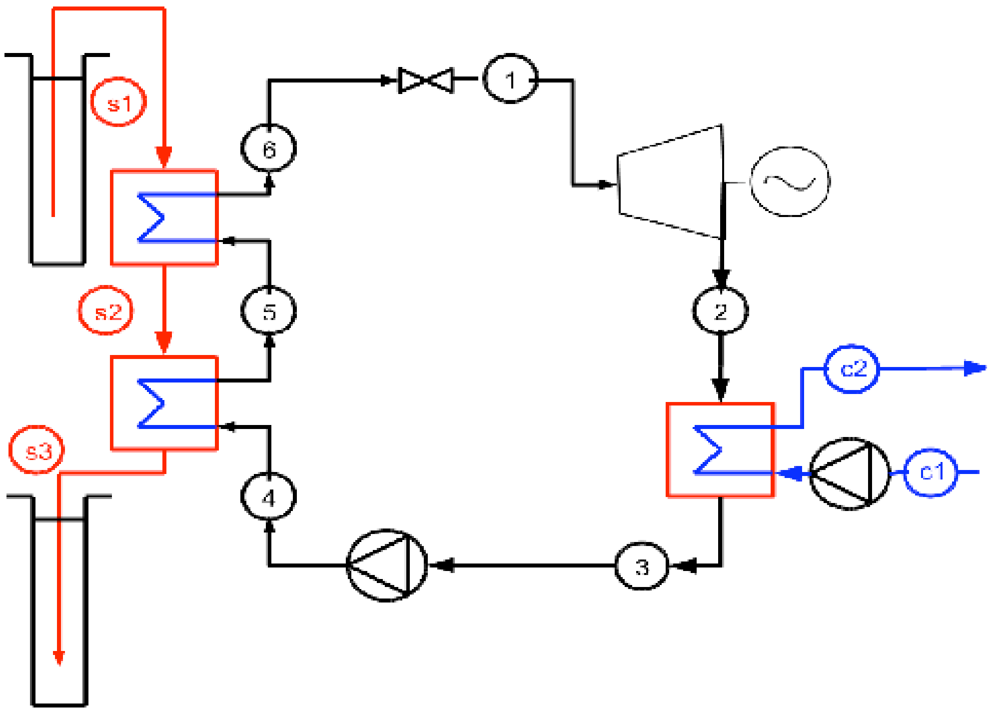

Figure 4.1 Schematic diagram of the binary model

States for three different mediums presented in the diagram are explained below.

States of the geothermal fluid (brine):

S1 – well-head

S2 – pinch point, bubble point of the vaporizer

S3 – rejection from the system

States of working fluid:

1- Inlet to the turbine
2- Outlet of the turbine, condensers' inlet
3- Outlet of the condenser, pumps' inlet
4- Outlet of the pump, preheaters' inlet
5- Outlet of the preheater, vaporizers' inlet
6- Outlet of the vaporizer

States of the cooling water:

C1 – inlet to the condenser

C2 – outlet of the condenser

4.5.1 Choice of the working fluid

The optimization work involved finding vaporaztion pressure to get the maximum net work, which is equal to difference between generated work and pump work and following formula :

(17)

In EES parametric tables, the relation between pressure and work output for different working fluids (defined in chapter 4.3.1) was checked.. Values which were obtained, were for a system with 20 m^2 area of heat exchangers each. Below Figures 4.2 and 4.3 show net work as a function of the vaporaztion pressure.

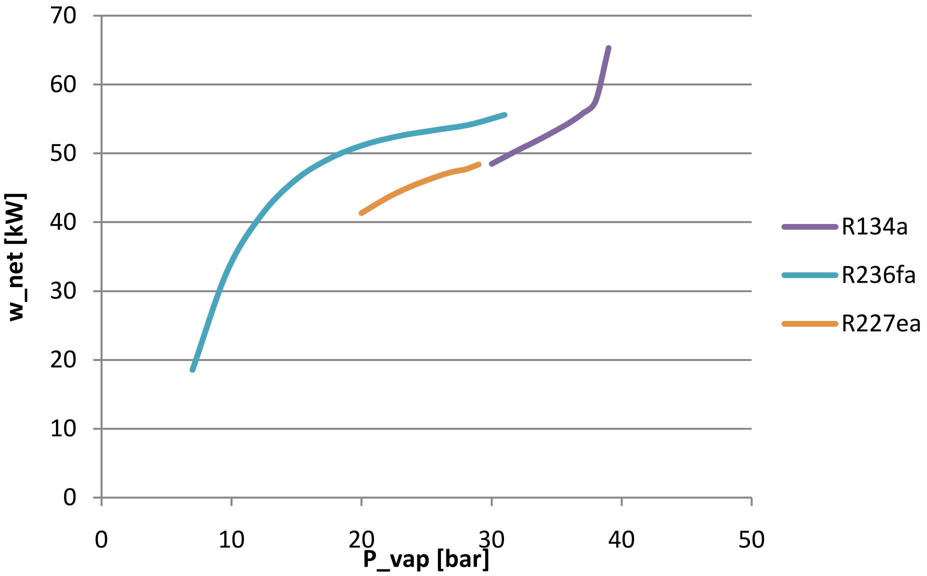

Figure 4.2 W_{net} as a function of P_{vap} for R134a, R236fa, R227ea

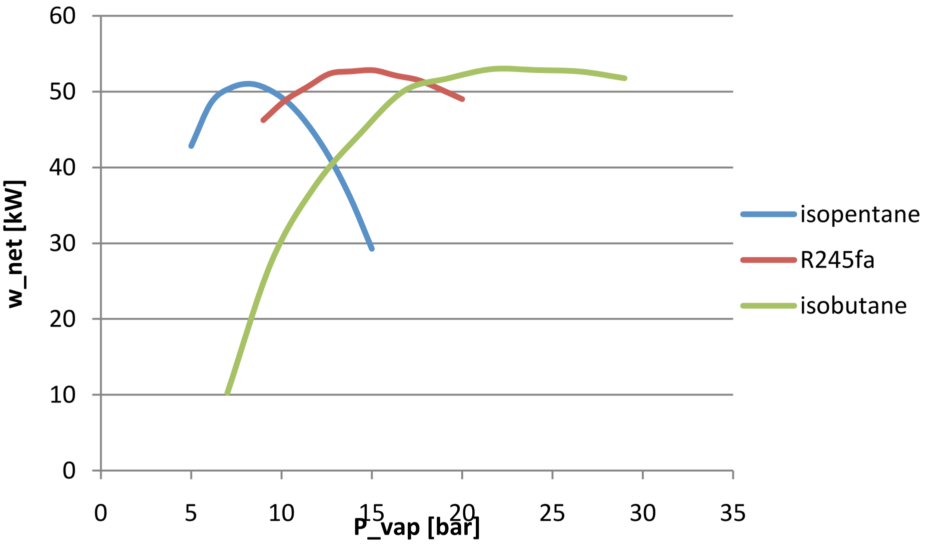

Figure 4.3 W_{net} as a function of P_{vap} for R245fa, isopentane, isobutene

For R134a, R236fa and R227ea work output increases with growth of P_{vap}. Maximum work is limited by the critical temperature for each of them. For working fluids presented in Figure 4.3 – isopentane, R245fa and isobutene; work rises to some maximum value as a function of pressure, and afterwards decreases. Optimal work is at much lower pressure than critical pressure for each fluid. Critical properties for fluids presented in Figure 4.2 are close to condition at the vaporizer, therefore work increases with vaporizer pressure. Critical conditions of working fluids in Figure 4.3 are far lower than the condition in the vaporizer, which results in different types of plots (with pick point). Besides, the pressure value providing maximum work is at lower values than for fluids shown in Figure 4.2, and is therefore plotted in a separate graph. The highest W_{net} with related P_{vap} are gathered in Table 4-1.

Table 4-1 Maximum W_{net} with related P_{vap} for each working fluid

No	Type of working fluid	P_{vap} [bar]	[kW]
1.	Isobutane	23.67	53
2.	Isopentane	7.78	51.32
3.	R134a	39	65.31
4.	R236fa	30.67	55.73
5.	R245fa	15.11	52.81
6.	R227ea	29	48.4

The highest output of 65.31 kW is achieved for R134a. However pressure is also very high, 39 bar. Because of high pressure shell-and-tube heat exchanger will have to be used and the prospective system will be more expensive. At this point of optimization the most promising working fluids seem to be isopentane and R245fa. They have relatively low pressures and reasonable (about 50 kW).

4.5.2 Optimization for specific working fluids

The next step of the optimization process involved matching optimal areas of the vaporizer and the condenser, in a way to get the lowest specific cost, with assumed vaporizer pressure from the first step of optimization (for) . The specific cost is the total power plant cost divided by total net work produced ( ). Estimation of the total power plant cost will be described in chapter 7 (Economic analysis). Required demand was chosen at 40 kW. It was according to the current demand presented in chapter 3.4. It is a much higher value than required in anticipation of future developments in this area, involving larger tourist accommodations.

All second step of optimization was made for R236fa and isopentane. Values were changed in the following order:

- vaporizer pressure
- area of vaporizer
- area of condenser

After finding new values of heat exchanger areas, appropriate pressure of the vaporizer was matched again. Table 4-2 presents results of optimization.

Table 4-2 Optimal design values for isopentane and R245fa

No	Type of working fluid	A_{vap} [m^2]	A_{cond} [m^2]	P_{vap} [bar]	Power Plant Cost [$]	Specific Cost [$/kW]	[kW]
1.	Isopentane	3	4	9	16391	1147	14.29
2.	R245fa	4	6	14	29108	1495	19.47

It can be noticed that the power plant performance is more sensitive to A_{cond} than to A_{vap}, the condensation area values are higher. For comparison, isopentane was presented in Table 4-2 as a cheaper scenario, but less environmental friendly. It is easily visible that isopentane has lowered specific and power plant cost. System generates less power, however.

Optimal output equals 14.29 kW for isopentane and 19.47 kW for R245fa. Required power is 40 kW. Therefore there was a necessity of scaling a model to reach demand. Required output is 2.8 and 2.05, times higher than for the best model, respectively for each working fluid. Area of vaporizer and condenser were multiplied by this factor in order to get proportional unit. Afterwards vaporizer pressure was matched again in order to fulfill demand at the lowest specific costs. As the final results, units with values presented in Table 4-3 were assumed as an optimal.

Table 4-3 Optimal designed values for 40 kW power plant

No	Type of working fluid	A_{vap} [m^2]	A_{cond} [m^2]	P_{vap} [bar]	Power Plant Cost [$]	Specific Cost [$/kW]	[kW]
1.	Isopentane	12	15	8	49345	1207	40.88
2.	R245fa	11	15	13	49306	1222	40.34

Performance of two fluids is pretty similar. Vaporizer pressure can be noticed as higher for R245fa, however. Power plant cost is almost the same for both working fluids, but specific cost is $ 15 cheaper for isopentane.

In Figures 4.4, 4.5, 4.6 were presented optimal values of A_{vap}, A_{cond}, and P_{vap} for unit producing 40 kW. They were estimated through optimization process, with changes of specific cost, for R245fa.

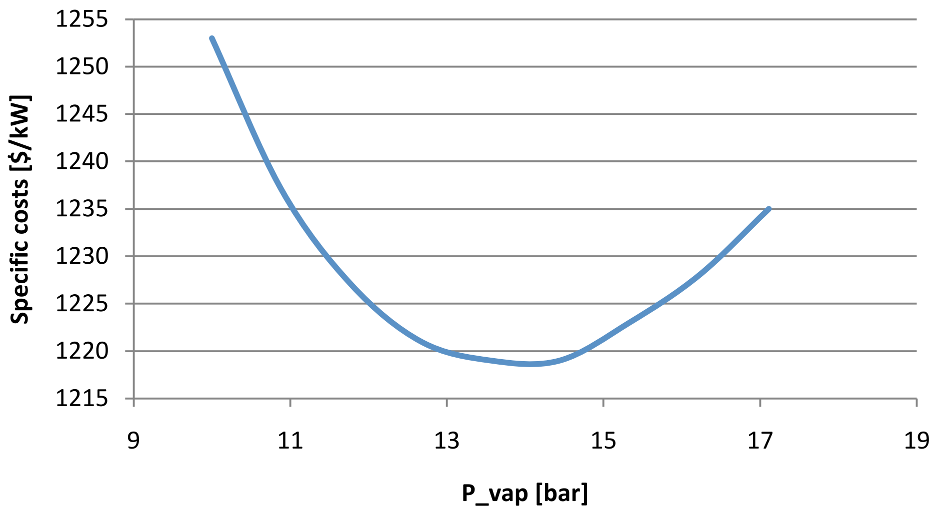

Figure 4.4 Specific costs as a function of P_{vap}

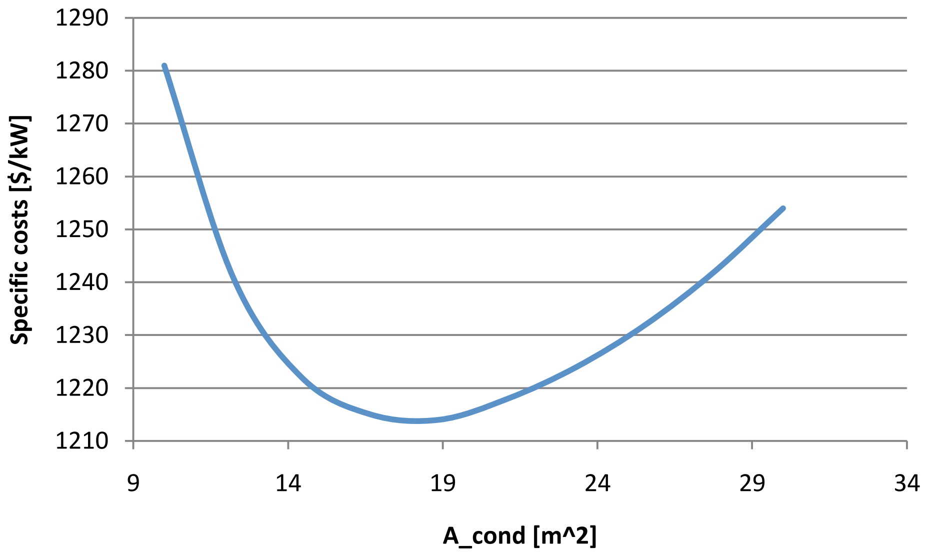

Figure 4.5 Specific costs as a function of A_{cond}

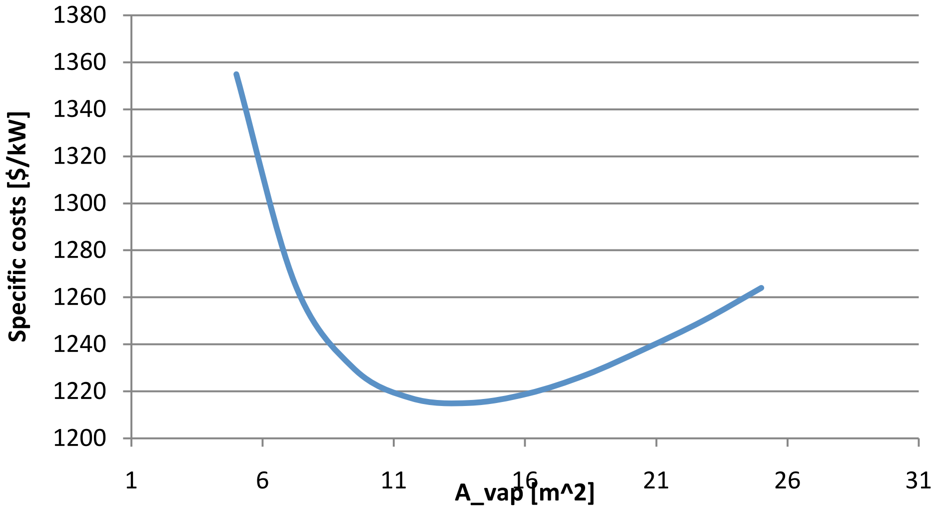

Figure 4.6 Specific costs as a function of A_{vap}

It can be noticed that in above graphs, optimal points do not appear as the same as defined in Table 4-3. It is because of the scaling the optimal unit from Table 4-2. Therefore, the best values according to diagrams are moved more on the right side from chosen one. The optimal point on the graphs would be the same as a selected according to optimization if units from Table 4-2 applies.

Changing of with various factors during optimization for R245fa is shown in Figures 4.7, 4.8, 4.9.

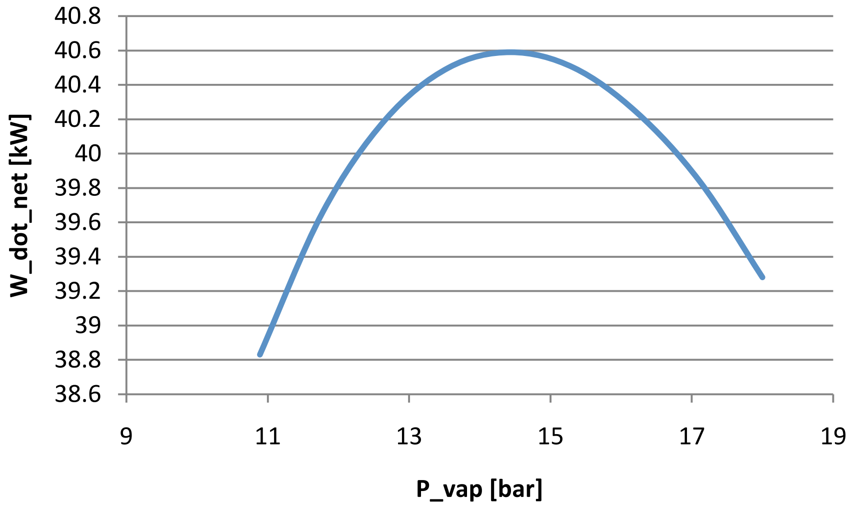

Figure 4.7 as a function of P_{vap}

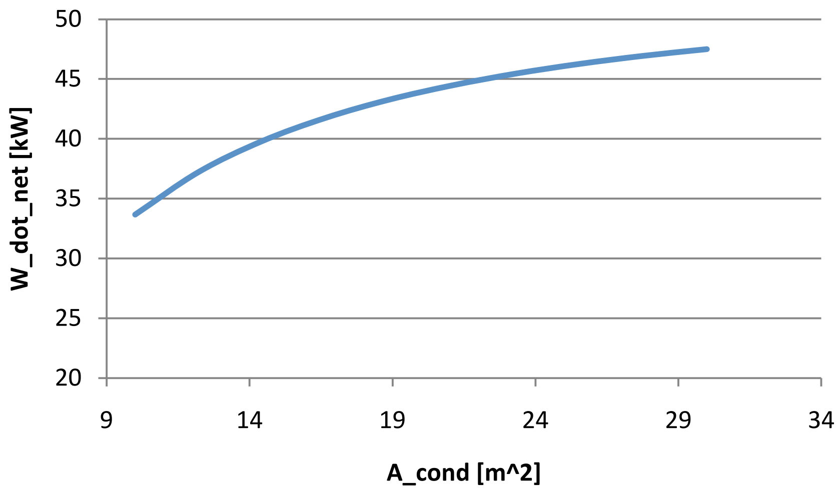

Figure 4.8 as a function of A_{cond}

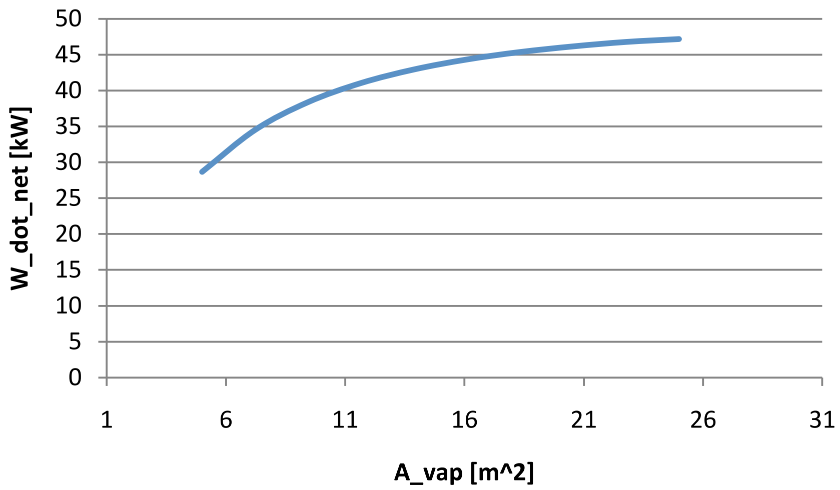

Figure 4.9 as a function of A_{vap}

In Figure 4.7, the maximum net work is at slightly higher vaporizer pressure, than it was estimated in Table 4-3. It was due to the same reason as in Figures 4.4, 4.5, 4.6, which was explained above. According to Figures 4.8 and 4.9 net work increases with area of heat exchangers. It is reasonable, pursuant to more energy transfer with more available area.

4.6 The proposed system

Schema of the proposed binary geothermal power plant from the EES program, with the main features of the system was presented in Figure 4.10.

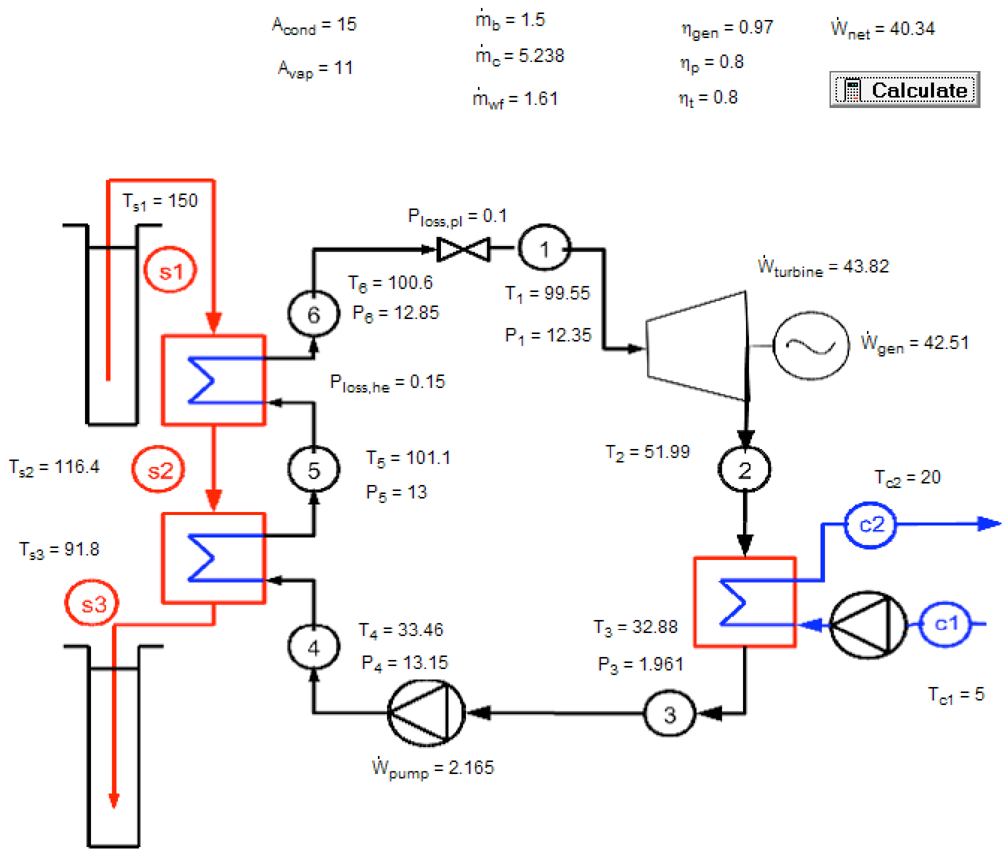

Figure 4.10 Schema of the proposed binary geothermal power plant

A system with R245fa as a working fluid and with parameters shown in Table 4-4 was chosen as the best solution for Hveravellir. It has a similar performance as isopentane, but it is more environmental friendly. If the main goal of design would be creation of the cheapest binary power plant, isopentane would have performed as more attractive. However, difference between power plant cost and specific cost for those two mediums is not significant $39 and $15, respectively.

Performance of the system with R245fa is the following:

Table 4-4 State properties of the cycle

State no	Temperature [^oC]	Pressure [bar]	Enthalpy [kJ]
1.	99.55	12.35	474.5
2.	51.99	1.96	447.2
3.	32.88	1.96	243
4.	33.46	13.15	244

5.	101.1	13	341.5
6.	100.6	12.85	474.5

One of the main rules during the design of ORC systems is that the pressure of the working fluid should always be higher than atmospheric pressure. The purpose of this is that it provides protection from eventual air leakage into the system. According to Table 4-4, for proposed systems all pressures are above 1 bar, fulfilling this requirement.

According to Figure 4.10 and Table 4-4 state 5 shows pressure in the vaporizer to be equal to 13 bar.

The geofluid is cooled down after the boiler to T_{s2}=116.4 ^{o}C, to be finally rejected with temperature T_{s3}=91.8 ^{o}C.

Area of each part of the vaporizer and condenser, and heat transfer in each of them are performed in Table 4-5.

Table 4-5 Performance of heat exchangers

Process in heat exchanger	Transferred heat [kJ/s]	Area [m^2]
Preheating	156.9	4.88
Boiling	214	6.12
Desuperheating	29.9	2.73
Condensing	298.9	12.27

The required and generated work by the system is the following:

Table 4-6 Work performance

Type of work	Work [kW]
Pump	2.17
Turbine	43.82
Generator	42.51
Net	40.34

Achieved net power 40.34 kW covers the assumed demand equal to 40 kW. Pump work is about 5 % of total produced.

Mass flow rate of the working fluid and cooling water was estimated as the following:

- = 1.61 kg/s
- = 5.24 kg/s

There will have to be considered an additional pump for a well with cold water. Work of this pump will have to be supplied, most likely from the binary station. Value can be estimated according to 1) the height difference between a stable water table in the cold water well and the inlet to the condenser and 2) the required mass flow rate:

$$/ \qquad (18)$$

The mentioned height difference was not measured directly at the site in Hveravellir, but if we reasonably assume it to be less than 20 m, the pump power needed is close to 1.5 kW, assuming 70% efficiency of a water pump.

A bypass will also be present. It has to be made between the entrance and exit of the turbine. It is necessary in all ORC systems. The reason for a bypass is to assure the required conditions for turbine operation, before the medium goes into it. A system, just after startup, will not get the desired conditions in each state. It will need some time for it. During that period, the turbine would work improperly, which could cause problems with the turbine mechanics.

4.7 Heat exchanger

Appropriate choice of heat exchangers in the Organic Rankine Cycle is the second most significant thing after proper selection of the working fluid. Heat exchangers have a great impact on power plant cost. The smaller unit to be designed, the greater costs consideration has to take place. There is no interest in getting the highest achievable output for island mode installations, but the required power only in a profitable way. This is due to no possibility for selling additional power to the grid.

The geometry of the flow configuration, the type of the heat transfer surface and the material of construction all vary according to the design requirements (Mills, n.d.). Two fluids can flow in coaxial or parallel tubes to achieve thermal connection. Also, tubes with fluid can be placed in a large shell with another surrounding fluid, creating the so-called shell-and-tube heat exchanger. The tubes are preferred when heat is transferred at high pressure. Plate type heat exchangers are also common, which consist of multiple plates joined with gaskets, common for gases at low pressure. Manufacturers usually produce heat exchanger as brazed, welded or gasketed. Streams can flow in parallel or opposite directions. In practice, counter flow two-stream heat exchanger is the best configuration for binary systems. It has more efficiency than the parallel flow type. Material of construction depends on geothermal fluid properties. Fluid behavior under the designed condition has to be carefully studied before constructing the power plant. For highly corrosive fluids, it is worth applying expensive titanium, nickel or hastelloy. If the brine is not chemically aggressive (not highly acidic), carbon steel or stainless steel can be used. In case studies, the materials such as SMO254 or 316L stainless steel have performed well (Valdimarsson, P., personal conversation).

As the best heat exchanger for the designed power plant, given a vaporizer pressure at 13 bar, shell-and-tube type of heat exchanger was chosen. General calculations about the size and numbers of tubes were carried out. Diameter (D) of tubes was assumed to be 20 mm (Valdimarsson, P., personal conversation). Area (A) obtained during optimization process can be assumed as for a one big tube with length L, thus:

(19)

Above formula allowed finding length of one tube. Cross section of a one tube was estimated as presented in Figure 4.11.

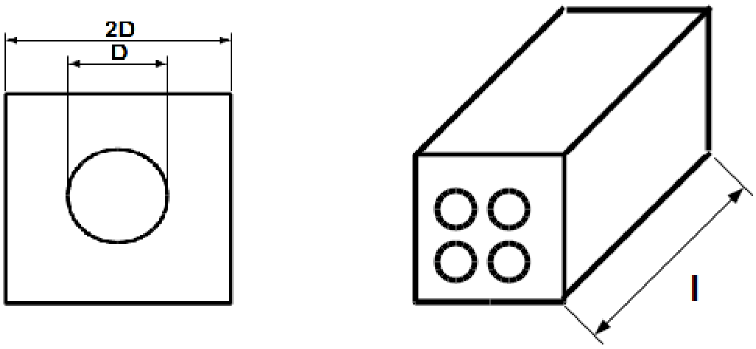

Figure 4.11 Cross section and schema of a one tube

While applying more number of tubes (n), and assuming different length (l) of heat exchanger, it can be done by the following formula:

– (20)

Then, area of a heat exchanger with n tubes was denoted as A'.

Results for vaporizer and condenser are presented in Table 4-7.

Table 4-7 Length and cut area of each heat exchanger for n-tube's number

Component	A [m^2]	L [m]	l [m]	n	**A' [m^2]**
Vaporizer	11	176	7	26	0.04
Condenser	15	239	7	35	1.46

4.8 Turbine performance

In the model a turbine with 80% of isentropic efficiency was assumed. For units which produce less than 1MW, cost of the turbine does not decrease anymore. When there is a small power output, also a small turbine has to be used. Small units do not require high mass flow rate of working fluid (1.61 kg/s in a case study). Consequently, since even a small turbine has to rotate with high speed to sustain even low demand, this has to be achieved with a low mass flow rate. Turbine works ideally, when the ratio of blades velocity over medium velocity equals 0.5. Proper range is estimated usually between 0.47 and 0.7. In Figure 4.12 changes of turbine efficiency with rotor and gas velocity, for different stages turbine is presented. Impulse or Curtis stages are used in low power turbines. In order to obtain higher efficiency of small turbines, high-speed turbines are implemented.

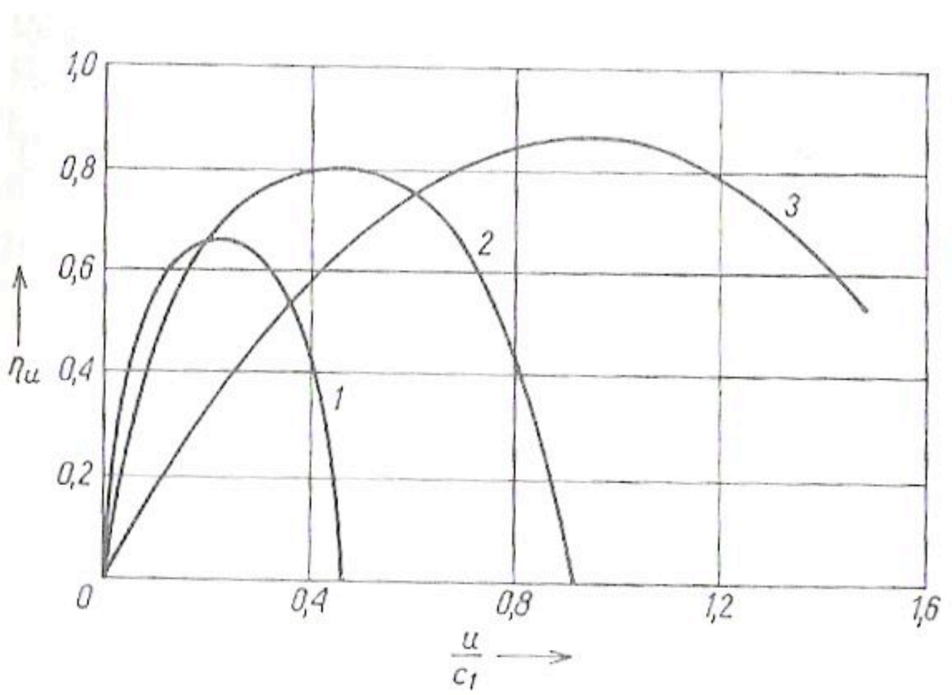

Figure 4.12 Turbine efficiency as a function of ratio between rotor and gas velocity. It is performed for various blading systems: 1-Curtis system with two velocity stage; 2-impulse stage; 3-reaction stage (Laudyn, 2000)

When the volumetric flow rate is too low, efficiency of the turbine can be changed or partial admission applied. The latter is based on closing a part of nozzles. If the volumetric flow rate is too high, the only solution is in its reduction.

It was found that it will be cheaper and better for a system to use a different device than a turbine in a 40 kW power plant, in order to avoid technical problems. There can be applied one from two types:

1) Helical screw expander
2) Piston steam engine

A helical screw expander is known also as a Lysholm engine and works similar to a gear pump, with the difference that it is helical. The diameter of screws increases between inlet and outlet which enables gas expansion. The most important parameter for this device is a ratio of the maximum and minimum volume. Lysholm engine offers more immediate boost than some centrifugal superchargers, which requires peak engine rpm in order to produce full boost. Additionally, a two-phase screw expander increases the recoverable power output from liquid geothermal brines. In the other hand, the necessity of high-precision computer-controlled manufacturing techniques makes the screw type supercharger a more expensive alternative to other forms of available forced induction. Therefore it was decided to not use it in the Lysholm system.

Piston steam engines work like steam locomotive engines. Two strokes with exhaust between each of them take place. During the first stroke high vaporizer pressure gas enters by the valve slide. For the period of expansion, it moves a piston connected to a flywheel and causes rotating motion. At the end of the piston stroke, the valve shifts, allowing the remaining steam pressure to escape at low condenser pressure. At the same time the valve slide begins admitting high pressure steam to the back end of the cylinder. This presses the piston forward, pulling the engine wheels around another half turn.

Steam engines are good for medium-or-small-scale electric power generator units. They are cheaper and do not cause any technical problems during low output production. New design such as in the 'Green Steam Engine'[3] provides power ranges from less than 1 kW up to hundreds of kW and could be the best choice apart from the proposed ORC power plant. Therefore it was decided to apply it in the designed model instead of a micro turbine. In order to choose a proper engine, it is important to define the volumetric flow rate and pressure at the inlet and outlet, respectively. Work done by a steam engine with individual states is presented in a pressure -volume diagram, in Figure 4.13.

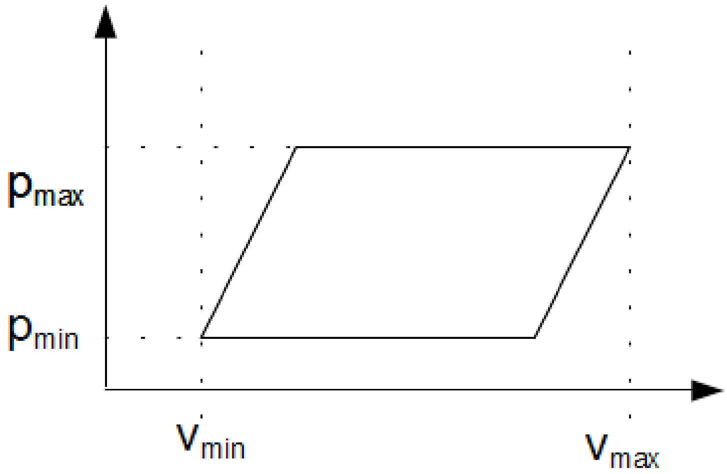

Figure 4.13 Thermodynamic states of steam engine work

To aid in the purchase of the proposed steam engine the required parameters are presented in Table 4-8.

Table 4-8 Properties for steam engine

Property	Symbol	Value	Unit
Vaporizer pressure	P_{vap}	13	bar
Condenser pressure	P_{cond}	1.96	bar
Volumetric mass flow rate at the engine inlet		0.02	m^3/s
Volumetric mass flow rate at the engine outlet		0.15	m^3/s

[3] More information at the webpage http://www.greensteamengine.com/

5 EXERGY ANALYSIS

Energy analysis of the system is not sufficient enough. It is based on energy balance and the first law of thermodynamics, thus energy cannot be destroyed. However the concept of "exergy destruction" has turned out to be useful in the design and analysis of thermal systems. Therefore exergy was denoted and within that concept the 2nd law of thermodynamic is applied. Exergy is defined as the maximum theoretical useful work (shaft work or electrical work) obtainable as the system interacts at equilibrium, heat transfer occurring with the environment only (Bejan, 1996). The method of exergy analysis enables to define the location, cause and magnitude, of waste and losses which appear in the system due to irreversibility. It shows that not only the condenser, as on a basis of the energy conservation, is responsible for rather low overall thermal efficiency of the power plant. Irreversibility inside of the steam generator has significant influence on it, too.

According to Bejan, when there is no nuclear, magnetic, electrical, and surface tension effects, the total exergy of the system is equal:

(21)

where the exergy values are physical exergy E^{PH}, kinetic exergy E^{KN}, potential exergy E^{PT}, and chemical exergy E^{CH}.

When a system is at rest relative to the environment, E^{KN} and E^{PT} are considered as zero. Furthermore E^{CH} is treated as zero, if there will not occur any considerable difference in a chemical composition of the stream. Thus, assuming that initial state of the stream is h_1, s_1 and the exit state correspond to the environmental state T_0, h_0, s_0 equation for the exergy balance will be:

(22)

Any system operates in some kind of a surrounding. It is important to distinguish between the environment and the system surrounding. Bejan describes the surrounding as everything not included in the system. Part of it, where there is no irreversibility, is the environment. The intensive properties of each phase are uniform and do not change significantly as a result of any process that undergoes inside of the environment. The border is within Earth's atmosphere, oceans and crust.

Work is feasible if there is a difference between the pressure, temperature, composition, velocity or elevation of the system and the environment. During changes it trends to equilibrium. There can be two types of equilibrium. First is so called the dead state and takes places when equilibrium of mechanical, thermal and chemical conditions is satisfied between the system and the environment. Another type is when only mechanical and thermal conditions are in equilibrium. There is a physical barrier that prevents the transfer of matter in the latter one. This state is described as restricted dead state.

The closed system exergy balance is as follows (Bejan, 1996):

Exergy transfer depends on what kind of process is concerned and is associated with the transfer of energy by heat and by work. The exergy destruction is related to irreversibility within the system. It is caused by the entropy generation.

As long as an analysis of a geothermal power plant processes involve control volumes at the steady state, it is worth defining the exergy balance for it. It is given by the following equation (Bejan 1996):

(23)

where and are exergy transfer at the inlets and outlets, respectively, is exergy transferred by heat, represents the time rate of energy transferred by work other than flow work and is exergy destruction.

The exergy transfer rates at control volume inlets and outlets are denoted, respectively:

(24)

(25)

In a geothermal binary power plant, exergy is lost when a stream leaves a system. It takes place in a preheater and a cooling tower.

For the proposed system the dead state is established as the ambient condition. Thus, a temperature equal to the temperature of the cooling water and pressure equal to one atmosphere, T_0= 5 °C and P_0= 1 bar, respectively.

Table 5-1 shows times rates of exergy in the proposed system with R245fa as a working fluid.

Table 5-1 Times rates of exergy

Type of exergy time rate	Notation	Times rates of exergy [kW]
Heat source		
Exergy inflow to vaporizer		179.9
Exergy outflow from vaporizer		112.1
Exergy loss in preheater		71.15
Working fluid		
Preheater		
Exergy destruction in preheater		12.32
Exergy at inlet of preheater		4.28
Exergy at outlet of preheater		32.89
Vaporizer		
Exergy destruction in vaporizer		13.1

Exergy at inlet of vaporizer	32.89
Exergy at outlet of vaporizer	87.65
Piping between evaporator and turbine	
Exergy destruction	0.83
Turbine	
Exergy drop in turbine	53.3
Exergy destruction in turbine	10.79
Work done by turbine	43.82
Exergy at inlet to turbine	86.82
Exergy at outlet of turbine	33.52
Condenser	
Exergy drop in condenser	30.64
Exergy destruction in condenser	22.14
Exergy at inlet of condenser	33.52
Exergy at outlet of condenser	2.88
Pump	
Exergy rise in pump	1.40
Exergy destruction in pump	0.76
Work done by pump	2.17
Exergy at inlet of pump	2.88
Exergy at outlet of pump	4.28
Cooling water	
Condenser	
Exergy rise at condenser	8.5
Exergy at inlet of condenser	4.71
Exergy loss in condenser	13.21

Exergy of the cooling water at the inlet of the condenser is equal to 4.71 kW, even though one would expect it to be equal to zero, since this is the environmental state (dead state). The reason for the non-zero value is the elevated pressure; the dead state pressure was assumed to be 1 bar (atmospheric) but the cooling system water pressure is assumed to be 10 bars. At 10 bar inflow pressure of cooling water into the condenser, the cooling water will at no point evaporate over the temperature and pressure range.

Exergy time rates in the system are also shown in Figure 5.1 in a popular Grassman diagram.

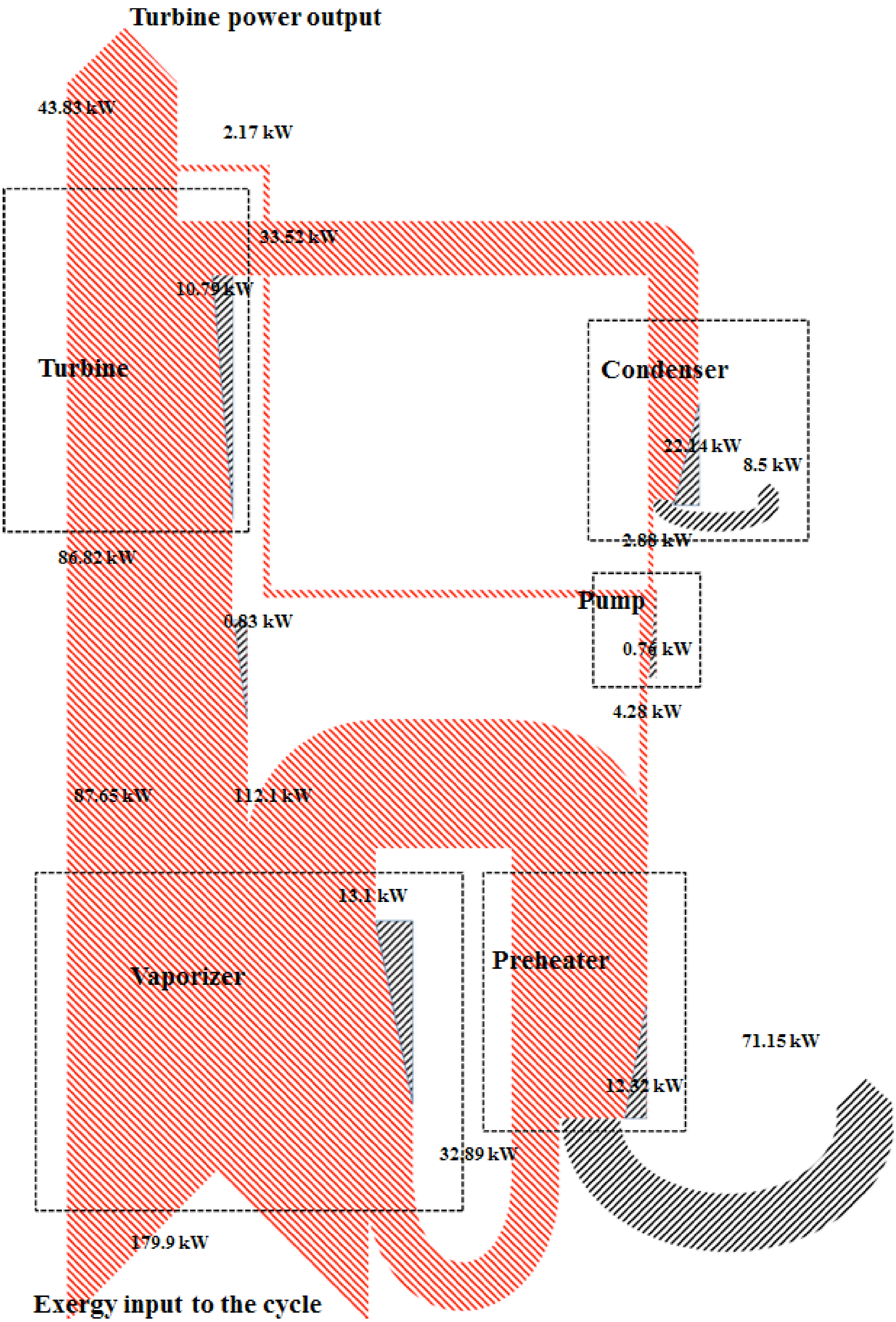

Figure 5.1 Grassman diagram

The Grassman diagram is a graphical method to present exergy flow in a thermal power plant. It is created based on data from Table 5-1. Exergy destruction is denoted by a triangle marked by a black wide upward diagonal and exergy losses from the power plant by a semicircle marked by a black wide upward diagonal. In the diagram, the width of each flux is matched with the magnitude of exergy in the stream. The wide arrows indicate flow direction of streams. Initially, ORC starts with heating up cold working fluid stream in a preheater and vaporizer. The exergy inflows to the system with a geothermal steam and supplies heat for working the fluid evaporation. Afterwards exergy flux changes pursuant to thermodynamic processes within the ORC. The most important process is the flux converted to work in a turbine. The remaining flux is going to the condenser and preheater, where it is partial loss to the environment. Finally, some exergy stream comes back to the vaporizer and closing the entire cycle.

6 EFFICIENCY

There are two possibilities to evaluate the performance of a power plant, according to the first or the second thermodynamic law. It varies how a system is treated.

6.1 First Law assessment of a Power Plant

If the power plant is consider as a cycle, thermal efficiency can be defined. It is based on the energy conservation, thus the net heat added to the cycle must equal the net work delivered by the cycle (DiPippo, 2008).

Thermal efficiency is as follows:

— (26)

Maximum theoretical efficiency for any system is the Carnot efficiency. It is calculated from the values of the highest temperature (T_H) and the lowest temperature (T_L) in a cycle, a heat source and a heat sink, respectively. In the ideal Carnot Cycle entropy changes follow isothermal paths. A geothermal source cannot be treated as isothermal, it cools when it transfers heat. DiPippo introduces a triangular cycle, as the ideal one for a geothermal binary power plant. It includes isobaric heat addition (up to T_H), isentropic expansion and isothermal heat rejection (at T_L). The efficiency for that cycle is following:

— (27)

In order to obtain maximum efficiency the lowest temperature is assumed as the dead state temperature, thus

— (28)

6.2 Second Law assessment of a Power Plant

It is useful to perform a power plant efficiency assessment based on the second law of thermodynamics. For this purpose the so-called Second Law or utilization efficiency is applied. Basically it is the ratio of the actual net plant power to the maximum theoretical power obtainable from the fluid in the reservoir state, thus exergy. DiPippo additionally suggests separating into two types of exergy efficiency, “brute force” and “functional”.

A “brute-force” exergy efficiency is the ratio of the sum of all output exergy terms to the sum of all input exergy terms. It can be used for any particular system, only when all exergy flows are determined. For the prospective system it is the following:

(29)

where is the net power output and e_i is the exergy at i-th state according to states denoted in Table 5-1.

A "functional" exergy efficiency is defined as the ratio of the exergy associated with the desired energy output to the exergy associated with the energy expended to achieve the desired output. Comparing to the brute-force exergy, this functional exergy is required to understand the nature of the system. If there will be assumed that the heat coming from a stream of hot geothermal brine has to preheat working fluid in a heat exchanger the below formula will be valid (Bejan, 1996):

(30)

On the other hand, it can be observed that the brine at state s3 is simply disposed of by means of reinjection back into the formation with no further use made of it, in which case we might use the following definition (DiPippo, 2008):

(31)

DiPippo defined it also as a utilization efficiency of a geothermal power plant.

All efficiencies are performed in Table 6-1. The first three lines present efficiency calculations based on the first law and the final three lines are exergy calculations using the second law of thermodynamics.

Table 6-1 First and Second Law assessment of the proposed system

Type of efficiency	%
Thermal	11.35
Triangular cycle	16.06
Maximum for triangular cycle	20.67
Brute force	67.53
Functional 1 (Bejan)	37.08
Functional 2(DiPippo)	22.42

6.3 Exergetic efficiency

Exergetic efficiency is important in a thermodynamic analysis of the system. As long as it is a ratio of the product over the fuel in exergy terms, it shows the percentage of the fuel found in the product. It allows comparison of components of the power plant between each other. Therefore it will be easily visible, which component of a system is the most responsible for any inefficiency and where improvements should be made.

Equations for exergetic efficiency for each component of the system as in Table 5-1, are gathered Table 6-2.

Table 6-2 Exergetic efficiency equation and value for particular components

Component	Exergetic efficiency formula	%
Preheater		69.89
Evaporator		80.69
Turbine		82.22
Condenser		27.73
Pump		64.67
Piping (6-1)		99.05

The highest exergetic efficiency is in the turbine and the evaporator, neglecting piping where exergy is destroyed only at the valve. The condenser has the lowest efficiency, due to heat dissipation there.

7 ECONOMIC ANALYSIS

The cost of the final product of the thermal power system constitutes the most important factor in selecting the best design option. In the chapter 4.5.2 it was described, that optimization was made based on each component's cost. Therefore, proper estimation of the major costs involved in the project is significant. Although it is hard to give specific price for some parts of the power plant construction, even a rough cost is useful. The market price of an item is, in general, affected not only by production cost of the item and the desired profit but also by other factors, such as demand, supply, competition, regulation, and subsidies. In designing a thermal system there is primarily interesting in production costs and it uses market prices only to value the system's by-products.

The total cost is split into fixed and variable costs. The latter depends stronger on the quantity of the output. It can be presented by the following equation (Bejan 1996):

where,

TCI – the total capital investment

FCI - the fixed-capital investment

SUC – startup costs

WC – working capital

LRD – licensing, research, and development

AFUDC – allowance for funds used during construction

7.1 Fixed-capital investment cost

This cost involves ones time cost, which are made to construct a power plant. There are distinguished direct and indirect costs.

7.1.1 Direct costs

PURCHASED EQUIPMENT

This cost is partly onsite costs, related with the purchase of components and spare parts of a power plant. During the design of a system, its size is estimated, material is selected, operation ranges specified and the choice of particular equipment is made. Quality of estimation depends on the reliability of given data. The best cost evaluation is according to vendors' quotations, at least for the most expensive parts. There should be considered the higher than the lower price in each case. Remaining components of direct costs are the percentage of the purchased equipment costs.

PURCHASED EQUIPMENT INSTALATION

This cost covers the freight and insurance for the transportation from the factory, the cost of for labor, unloading, handling, foundations, supports and all other construction expenses related directly to the erection and necessary connections of the purchased equipment (Bejan 1996). If the system is considered as a complete group, these costs are usually included before in different cost components. The installation cost for equipment mainly varies from 20 to 90%.

PIPING

Includes the material and labor costs connected with the construction of all the piping used directly in the system. In general, ranges from 50 to 70% of the purchased equipment costs.

INSTRUMENTATION AND CONTROLS

Vary with system automation between 6-40% of the purchased equipment costs.

ELECTRICAL EQUIPMENT AND MATERIALS

This cost, which includes materials and installation labor for substation, distribution lines, switch gears, control centers, emergency power supplies, area lighting, and so forth, is usually 10-15% of the purchased equipment cost (Bejan 1996).

LAND

It strongly varies with the location, where a power plant is planned to be built. If the land has to be bought, about 10% of the purchased equipment cost can be used.

CIVIL, STRUCTURAL, AND ARCHITECTURAL WORK

It contains all services related to buildings, as well as the cost for roads, sidewalks, fencing, landscaping, yard improvements, and so forth. Depends, if a system is a new or an expanded, affects costs as 15-90% of the purchased equipment cost.

SERVICE FACILITIES

States for supplying the general utilities required to operate the system such as fuel(s), water, steam and electricity (assuming that these utilities are not generated in the main process), refrigeration, inert gas, and sewage. This category also includes the cost of waste disposal, environmental control, fire protection, and the equipment required for shops, first aid, and cafeteria. The total cost of service facilities may range from 30 to 100% of the purchased equipment cost.

7.1.2 Indirect costs

ENGINEERING AND SUPERVISION

This category includes the cost for developing the detailed plant design and drawings, and the costs associated with cost engineering, scale models, purchasing, engineering supervision and inspection, administration, travel, and consultant fees. In general, it may be 25-75 % of the purchased equipment cost.

CONSTRUCTION

It consider everything, which is related with construction site as temporary facilities and operations, tools and equipment, home office personnel located at the construction site, insurance. Bejan advises this cost as 15% of the total direct costs.

CONTINGENCIES

Output of a power plant will vary during later work. It will be caused, among other things, changes of weather, work stoppages, transportation difficulties or sudden price changes. It is assumed from 8 to 25% of the sum of the above costs. This factor will change with feathers of a system as complexity, size and uniqueness of the plant.

7.1.3 Other outlays

STARTUP COSTS

Everything which is connected with startup time, when the system is not operating or operating at only partial capacity is covered by this category. It can be presented as a part of total investment costs or separately. In first option, it might run from 5 to 12% of the fixed-capital investment.

WORKING CAPITAL

It represents cost which has to be cover for sustain work of a power plant, before it will get incomes from sold products. According to Bejan it is 10 to 20% of the total capital investment.

LICENSING, RESEARCH, AND DEVELOPMENT

If there was a necessity of research or development connected directly to a system, it is put into this cost. For a binary geothermal power plant there is no license required. Kalina system does, however.

ALLOWANCE FOR FUNDS USED DURING CONSTRUCTION

Time between the beginning of design and system startup can be 1-5 years. During that period money are spend without any revenue. Therefore money will come from loans and company resources. The allowance for fund used during construction represents the time value of money during construction, and is based on an interest rate.

Purchased components costs were estimated using approximated market values (Valdimarsson, P., personal conversation). They are shown in Table 7-1.

Table 7-1Cost evaluation of main components in a binary geothermal power plant

Component	Factor	Factor unit	Cost of 1 unit factor [$]	Number of units	Total component cost [$]
Vaporizer	A_{vap}	m^2	400	11	4400
Condenser	A_{cond}	m^2	400	15	6000
Pump		kW	600	2.165	1299
Turbine		kW	600	42.51	25504
Total cost					37203

The total power plant costs are usually calculated as a percentage of main component costs, as was described above according to Bejan. In a case of a big power plant, factor of 60-80% is sufficient enough (Valdimarsson, P., personal conversation). Putting it on Hveravellir case, it proved to be an exaggerated estimation. Thus, based on a second reconsideration, remaining cost (all costs of the power plant excluding main components costs) were assessed as $ 300 per each 1kW produced by the power plant (Valdimarsson, P., personal conversation). It is not entirely correct, because some components of the power plant do not decrease below some point (e.g. turbine price). However, this assumption was treated as good enough for the rough economic analysis which conducted in the project. Subsequently, the following formulas were used:

$$Power\ Plant\ Cost = Main\ Component\ Cost + 300 * \quad _{net}$$

For the designed ORC power plant, this cost equals \$ 49306.

The specific cost was calculated using:

$$Specific\ Cost = Power\ Plant\ Cost / \quad _{net}$$

The result is \$ 1222 for the proposed system.

A division of particular fixed-investment costs was made based on the description given above. Assumed percentages with related value in \$ are gathered in Table 7-2.

Table 7-2 Breakdown of total capital investment (TCI)

I. Fixed capital investment (FCI)	Cost percentage	Value
A. Direct costs (DS)		
1. Onsite costs		
• Purchased-equipment cost	PEC	37203
• Purchased-equipment installation	9% of PEC	3348
• Piping	2% of PEC	744
• Instrumentation and controls	3% of PEC	1116
• Electrical equipment and materials	2% of PEC	744
2. Offsite costs		
• Land	0% of PEC	0
• Civil, structural, and architectural work	7% of PEC	2604
• Service facilities	3% of PEC	1116
B. Indirect costs		
1. Engineering and supervision	4% of PEC	1488
2. Construction costs including contractor's profit	4% of DC	387
3. Contingencies	2% of FCI	231
II. Other outlays		
A. Startup costs	1% of FCI	118
B. Working capital	2% of TCI	238
C. Costs of licensing, research, and development		

D. Allowance for funds used during construction	
TCI of power plant	49337

The percentages used were in most of cases based on Lukawski, M. "Design and optimization of standardized organic Rankine cycle power plant for European condition". Most of the changes made are due to the remote plant location, thus it was estimated that 9% of PEC would be purchased-equipment installation instead of 6%. Due to small size of a plant, remains onsite costs are pretty low.

It is worth to mention, that in the case of Hveravellir there will not have to be any investment related to well drilling. There is already an existing well at the site and therefore the significantly lower capital cost is most beneficial in order to construct a system.

Based on the annual demand of 71925 kWh/year (chapter 3.4) electricity production using ORC power plant would cost \$ 10034 annually, with price of 0.1395 \$/kWh. But if the unit would built it would probably be owned by Hveravellir, thus the only cost which they will have to pay will be the initial costs. Calculation of the Net Present Value (NPV) shows as following:

$$- \qquad \frac{\qquad}{\qquad} \tag{32}$$

Where I_{total} is the total investment cost, B is a benefit, d is a discount rate and n is a time in year's number.

Yearly benefits were estimated as \$ 10034 and typical interest rates in the geothermal industry of about 7%. Result showed that after about 6 years, the power plant cost would be returned, which is a fairly short time in geothermal power projects. If money saved from diesel engine will be assumed as an annual benefit, \$ 11011, investment would payback even after 5 years.

8 SOUTHERN POLAND

In Poland only low temperature reservoirs are available, between 30 and 130 °C, at the depths from 1 to 4 km. Southern part of the country own one of the best sources (between 10 to 90 °C), thus was considered in this paper. In particular, geothermal aquifers in the Malopolska Voivodship were studied. Geothermal utilization in Poland is not developed, although heating system is working properly at water temperatures range from 20 to 60 °C. Considering the highest feasible temperatures, binary geothermal power plant would perform profitable. Unfortunately, only 46 TWh of power, which is 1.6 % of total achieved from renewable, comes from geothermal in Poland.

In this chapter, feasibility of applying the proposed binary geothermal system from Hveravellir in the southern Poland will be presented.

8.1 Geological and geothermal conditions

There are distinguished three main geological units in South Poland (Figure 8.1):

1) The Carpathians
2) The Miechow Trough
3) The Silesian – Cracow Monocline

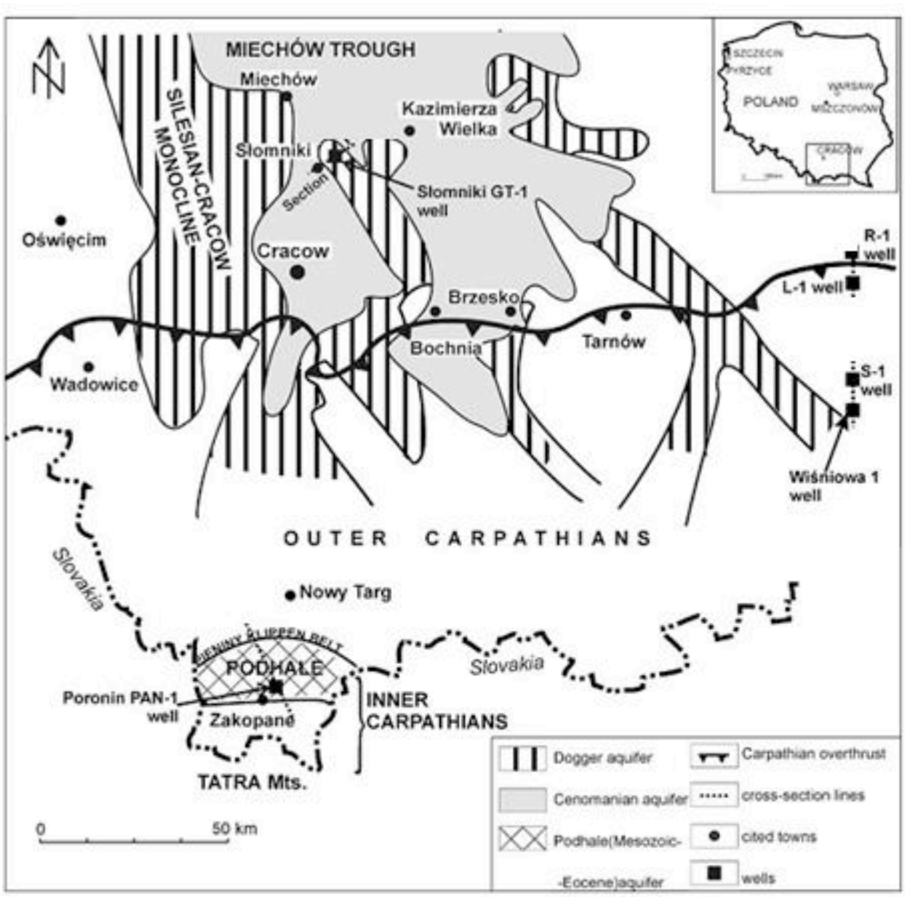

Figure 8.1 Main thermal aquifers of Southern Poland (Bujakowski, 2003)

The groundwater in these units is deposited in aquifers of various age: Paleozoic (located in the basement of all the units), Mesozoic (located in the Miechow Trough, Silesian – Cracow Monocline and the Carpathians), Tertiary (the Carpathians) and Quaternary (Bujakowski, 2006). The last was exluded from the analysis, due to low temperatures and limitiaion of source.

THE CARPATHIANS

Based on a structure and hydrogeology the Carpathian region can be divided into two different sections:

a) The Inner Carpathians (the Pieniny Klippen Belt, the Tatra Mountains and the Podhale Trough)
b) The Outer Carpathians (the flysch band)

Particularly favourable geothermal features come about in the Podhale Trough, the Inner Carpathians. It has got unique properties in Europe, which is the homogenous distribution of reservoir parameters throughout entire unit (there were high artesian flows from the Triassic and Eocene basin in nearly all boreholes). The stability of this these parameters are due to the strong krastification and fissuring of the Tatra Mesozoic Layers underlying the whole Podhale Trough (Bujakowski, 2003). In the Outer Carpathian, thermal water only occurs locally and is characterized by the low yield. It is also uncertain whether their flow-rate parameters would remain stable during exploitation, thus required more research.

MIECHOW TROUGH

The Miechow Trough is located in the north of the Carpathians overthrust, where Mesozoic aquifers form part of the regional reservoir structure of the Polish Lowland stretching all the way north to Szczecin (north-west Poland, Figure 8.1). Geothermal water feasible for exploration occurs in two sandstones and one carbonate aquifers. First is hosted in Cenomanian and the underlying Dogger, latter in Upper Jurassic. These reservoirs lie at great depth beneath the Outer Carpathian flysch in the south; however, it is only north of the Carpathians, in the Miechow Trough area, that they show favourable reservoir parameters. Unstable tectonics and differential dip are typical of these reservoir structures (Bujakowski, 2003).

SILESIAN-CRACOW MONOCLINE

The Silesian-Cracow Monocline is situated at the north-west part of study region. Due to the shallow depth of Mesozoic formation (resulting in low temperatures) and unfavorable reservoir parameters (low inflows) this area does not exhibit as promising for the utilization of geothermal energy. Some potential is provided by old mines flooded after the closure of coal mines, where temperature up to 45 °C were measured and where large, exposed areas permit high inflows (Bujakowski, 2006).

The best feasible features of above's described geological units are combined in Table 8-1.

Table 8-1 The highest obtainable reservoir's features for each geological unit

Geological unit	Average heat flow [mW/m^2]	Depth [m]	Temperature [°C]	Inflow [kg/s]

The Carpathian	56	Up to 2900	90	Up to 200
The Miechow Trough	50	Up to 2300	20-70	< 2.8
The Silesian-Cracow Monocline	55	shallow	< 40	Low

The highest temperatures are achievable in the Carpathian unit. There are also the best yields from boreholes. In the Silesian-Cracow Monocline the lowest geothermal temperatures occur, though the deep seated groundwater horizons (in Paleozoic strata) still remain unexplored.

8.2 Boundary conditions

Temperatures ranging from less than 20 to 40 °C are particularly suitable for heating systems, often using heat pumps. In order to utilize the geothermal potential of Southern Poland for power production using ORC systems, temperatures should be at least 80 °C, to make extraction economically viable. However, the lowest source temperature of a commercially operating ORC geothermal power plant in the world is 70 °C at Chena Hot Springs in Alaska, which came online in late June 2006, putting Alaska squarely on the map for new geothermal technologies. In comparison, the designed system for Hveravellir runs at relatively high temperature for an ORC plant. Therefore, to apply such a system in Poland, the highest feasible condition had to be chosen. These conditions are in the Podhale Through. Consequently, onsite data was assumed as the following:

- Well-head pressure - P_{s1}= 26 bar
- Well-head temperature - T_{s1}= 90 °C
- Mass flow rate - in a range of 2 – 25 kg/s

To start with, mass flow rate was assumed to be 10 kg/s, which is inside the predefined range. Choice was made according to the resource features and required pipe diameter. For 10 kg/s diameter of the pipe is reasonable.

8.3 Cooling system

In a case of implementation of the same unit, as in Hveravellir, in Poland, natural cooling was considered. However, it is not certain that accessibility of cold water is in place. Thus, range of area where unit could be applied was narrow. Besides, it could not be assumed that a cold well will ever be available. As a result, in Poland there will be necessary to create a cooling pond, close to the water source.

In the other way, the most suitable cooling system in Poland would obviously be air cooling. Average temperature in the country is about 8 ^{o}C, and varies between -16 and 27 ^{o}C during a year.

8.4 Assumption for the model

The assumptions used for model calculations are the same as described in chapter 4.3, thus will not be presented again here. Additionally, choice of the working fluid was not repeated, but follows the previous analysis made in chapter 4.5.1. Refrigerant R245fa was estimated as the best medium, in a view of power, economic and environmental performance for a unit operating in Southern Poland, as well.

8.5 Performance of the designed model in Poland

Model optimized for operation at Hveravellir, was used with different field data (assumed in chapter 8.2), in Southern Poland. Heat exchanger areas were 11 and 15 m^2 of vaporizer and condenser, respectively. Vaporizer pressure had to be calculated again. Assuming mass flow rate of 10 kg/s the best pressure was estimated. It is shown in Figure 8.2

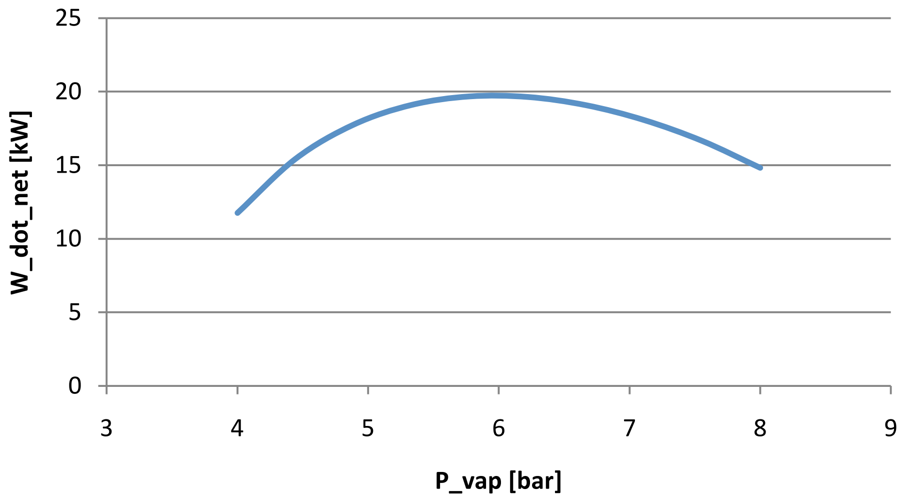

Figure 8.2 as a function of P_{vap}

Optimal pressure, at which the highest output was reached, was at 5.7 bar. The power generated is then about 19.68 kW. Low values are due to the low field temperature. Therefore the Hveravellir system will not able to run at full load in Poland at pressure equal to 13 bar.

Changes of Specific Cost with vaporizer pressure are shown in Figure 8.3. The lowest specific cost is $ 1470, while power plant cost is $ 28846. It is very low, but small size of the system and low output cannot be neglected, though.

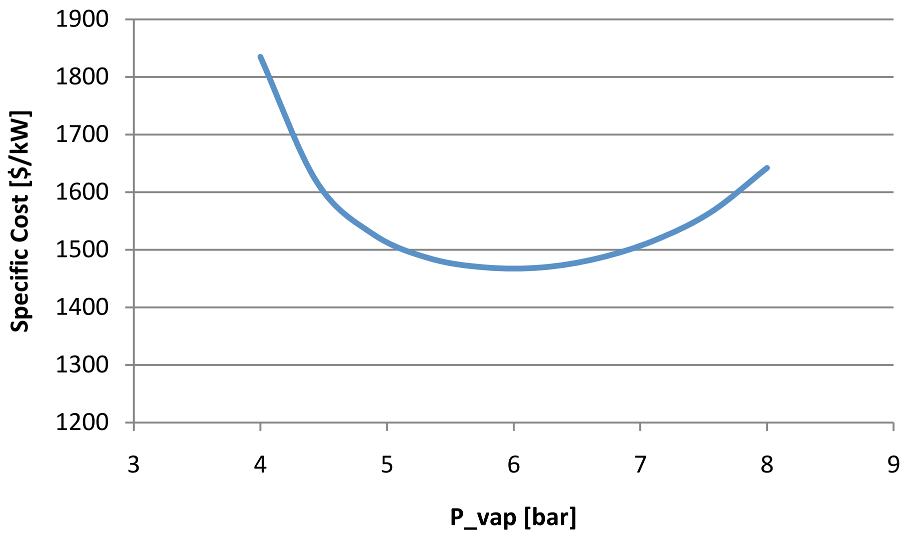

Figure 8.3 Specific costs as a function of P_{vap}

System behavior according to the various brine mass flow rate ( ) was checked, in order to increase the net power output ( ). Figure 8.4 presents the system performance for different mass flow rates.

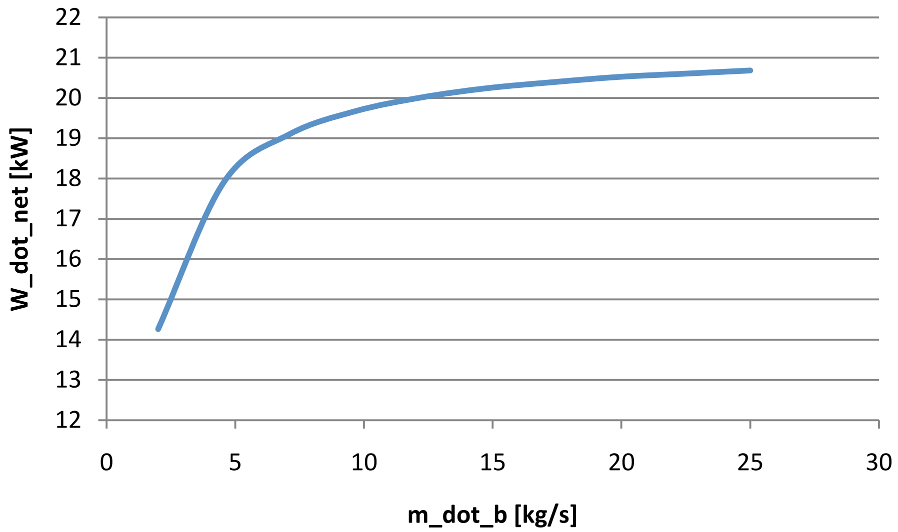

Figure 8.4 as a function of

From the above diagram it can be clearly observed that work produced by the system increase up to same value and then it is rather stable. From 2 kg/s of a mass flow rate up to 10 kg/s, unit production increases from 14 to 20 kW. At higher flow rates the output is not changing significantly, thus applying higher mass flow rate would be useless. Additionally, it was checked, what would be the maximum mass flow rate, which system could handle, and what output would be achieved then. It was realised that at 70 kg/s the system would generate about 22 kW. It was the highest output a unit designed for Hveravellir could perform in Poland.

8.6 Comparison between Southern Poland and Hveravellir

Geothermal binary power plant, which was designed and optimized in a way to have the best performance while operating in Hveravellir, Iceland, was researched how it would appear in the Podhale Through, Southern Poland condition. Both of fields are treated as low temperature reservoirs, although the meaning of this term is according to the geothermal condition in each of those countries. As long as there is a quite high temperature at Hveravellir, the borehole will not require a high mass flow rate and high well-head pressure to power the plant. Geothermal potential of both study cases is compared in Table 8-2.

Table 8-2 Comparison of onsite condition of Hveravellir and Podhale Trough

Place	Temperature [°C]	Pressure [bar]	Mass flow rate of geothermal brine [kg/s]
Hveravellir	150	4.9	1.5
Podhale Trough	90	26	10

Various results achieved from the same binary geothermal power plant running in different places were combined and put together into Table 8-3.

Table 8-3 Performance of the system operating in Hveravellir and the Podhale Trough

Place	Vaporizer pressure [bar]	[kW]	Specific Cost [$]	Power Plant Cost [$]
Hveravellir	13	40.34	1222	49306
Podhale Trough	5.7	19.62	1470	28846

At Hveravellir, the unit was operating at 13 bar of vaporizer pressure, and producing about 40 kW. The same system, in the Southern Poland condition, required 5.7 bar to reach optimal work output, which was about 20 kW. It can be easily noticed, that it is a half of the Hveravellir output. Although the power plant cost is around 40% lower in the case of Poland, specific cost was $ 248 higher, which means that the specific cost in Poland is 20% higher than the specific cost at Hveravellir.

Pursuing the previous methods, the following conclusions were made. The designed geothermal binary power plant, running in Hveravellir, although producing low power, would sufficiently supply demand for huts which are there. While application of the same system, in Podhale Trough, Southern Poland, was simulated, it resulted in the extremely low output of the power plant. It would hardly provide electricity for about two houses, if assumption of roughly 10 kW per house was made. That means that it is pointless to run this unit in Poland. There are too big differences in the geothermal reservoir features, which would require a different design of the unit.

8.7 Recommended modifications

For assumed condition in Poland, mass flow rate was almost ten times higher than in Hveravellir. Therefore a redesigned system would need a bigger area of both the vaporizer and condenser. While geothermal water has temperature of about 90°C, pressure in the vaporizer will be much lower, as well. Also, it cannot be neglected, that the weather condition is the Southern Poland are different than in the middle of Iceland. Thus, it is evident that the cooling system should be replaced. According to the climate in Poland and ambient temperatures in the Podhale Trough, the best choice would be a dry cooling system. The ambient temperature in the south is lower than in the rest of the country, due to the higher altitude above sea level. Annual average is about 10 °C Performance of yearly dry bulb temperature is presented in Figure 8.5. It is for Zakopane, which is in the border between the Podhale Trough and the Tatra Mountains. It varies during the year between -16 and 27 °C, thus affects power performance over a year. It can be expected, that the highest output will occur during the winter time, when the cooling air temperature has the lowest, and the lowest output will be during the summer time, when there are the lowest air temperatures, respectively.

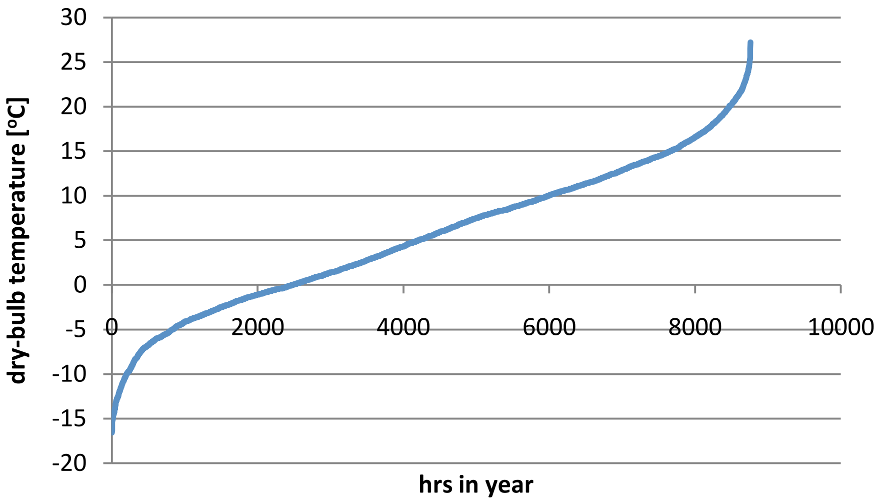

Figure 8.5 Duration curve of dry bulb ambient temperature for Zakopane[4]

A project called "Geothermal binary system for rural area in Poland" Bagierek, M. , was conducted in conjunction with this thesis in which the optimal design of a geothermal ORC power plant utilizing the Polish conditions was investigated. The main properties of the Polish system were the following:

- Temperature of the cooling air – T_{c1}= 10 °C
- Mass flow rate - = 10 kg/s
- Vaporizer area - A_{vap}= 41 m^2
- Condenser area - A_{cond}= 41 m^2
- Vaporizer pressure - P_{vap}=6.7 bar

[4] Duration curve was created according to data taken from http://apps1.eere.energy.gov

- Net work output - = 40.28 kW

It is clearly evident that heat exchanger areas are almost 3 times higher. The power output was 40.28 kW, while ambient temperature was assumed to be stable. In the referred paper the variation of power during an entire year was also presented. It showed, that the difference between the minimum and maximum power was about 60 kW. This disadvantage of the system, required consideration on how many customers could be sufficiently supplied each year.

9 CONCLUSIONS

This project clearly showed that binary geothermal power plants are sensitive to many factors. Each plant design depends on the required output and ambient conditions at the plant location, using tailored components for each working fluid.

The main goal was to design an optimal small power plant which will supply the predefined demand of 40 kW, operating in island mode. The reservoir properties were good and can easily supply more electric power output, it will not be of any current use in the area. There is no electrical grid network and the area is remote and uninhabited, which makes it an unfeasible place for excessive power production. Therefore, well head pressure and mass flow rate from the well were kept at fairly low values in the design, 4.9 bar and 1.5 kg/s.

Thermodynamic optimization gives significant conclusions about the design of a small ORC system. Working fluid choice showed that R245fa and isopentane assures the best performance among all six investigated fluids. They operate at relatively low pressure, compared to the rest. As long as demand was low, there was no problem with fulfilling it. Only R134a appears with slightly lower output, than the others. Mainly environmental aspects justified the final choice of R245fa.

Vaporizer and condenser area had a fair influence on cost and work output, as well. Although in a small unit the heat exchange areas are also small, an optimal value was reached at which specific cost occurs as the lowest. It was shown also that the system is more sensitive to condenser than vaporizer area. The optimal results were 15 and 11 m^2, respectively.

Optimal size of a unit and vaporizer pressure of 13 bar resulted in specific cost equaled $ 1222, and $ 49 306 total power plant cost. Considering application of individual steam engine system power plant cost was proposed for $ 325 000, by Energent Company. In that comparison it can be stated that a binary geothermal power plant at Hveravellir would be a proper choice. Advantage of the ORC power plant is that it produces only the essential demand, while the smallest feasible steam engine would produce about 150 kW, which had results in a higher cost.

Comparing to the existing diesel operated electricity system, the binary system is cheaper in operation. Cost of electricity over a year would be $ 10034 from ORC system, comparing with currently $ 11011, from applying diesel engine. If consider ownership of power plant by Hveravellir, it would require only investment money instead electricity cost additionally. In that way, cost would return in about 5 years. This is also significant profit of the designed power plant.

Presence of the cold well on site is also huge advantage. A cooling system, which usually is expensive and a parasitic power consumer, can be easily avoid by applying natural water cooling. In addition, geological structure, abundance of fissures and fractures simplifies the disposal of waste water from condenser and evaporator.

The disadvantage of a binary unit is obviously its high investment cost. The area is located in central Iceland, thus construction and shipping cost will be fairly high. As long as in Hveravellir the wells to be utilized already exist, their economic analysis was neglected.

Since the Hveravellir region is famous for geothermal hot springs, using a binary system in the area would make this field more attractive to tourist. An environmentally friendly power plant instead of the current diesel engine would reduce emission of CO_2.

The designed unit, which would perform well while operating in Hveravellir, Iceland, would not perform the same under Southern Poland conditions, due to different reservoir properties, such as temperature, pressure and enthalpy. The Polish binary unit will required a different optimal scheme. An ORC power plant should not be installed without following a site-specific design, in order to perform profitably.

In Southern Poland, the geothermal waters have far lower temperatures. The Hveravellir system would not produce the same power in Poland, without significant changes. Adjusting the vaporizer pressure was not enough. It would be necessary to provide new, optimally designed vaporizer and condenser.

Additionally, a different cooling system would perform better in Polish conditions. Assumption about an access to a cold water source, which was made during the application of the Icelandic model in Poland, gave a narrow range of possible locations for the binary power plant. Polish climate appears as proper for dry cooling systems, using air as cooling medium. However, it will have an influence on the power plant's annual performance. Various ambient temperatures introduce cooling irregularities.

A binary geothermal power plant designed for special Southern Poland condition was studied in another project (Bagierek, 2011) . It considered all the required changes in a system and further optimization. Besides, performance of the system over a year was presented.

In both cases, there was the possibility of making a heating system from rejected geothermal brine. In Hveravellir it was not considered, since a heating system with a heat exchanger using geothermal water as a source has already been installed. In Poland this solution cannot be neglected. It could make a system more beneficial and help in the reduction of CO_2 emission, which is mostly produced from coal burning heating system, currently used.

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