



Utilization of Geothermal Brine for Electrical Power Production

A dissertation by
Marta Rós Karlsdóttir

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University of Iceland
Faculty of Engineering
VR-II, Hjarðarhaga 2-6, IS-107 Reykjavik, Iceland
Phone +354 525 4000, Fax +354 525 4632
verk@hi.is
www.hi.is

Abstract

The objective of the study is to compare methods that increase power generation from conventional single flash power plants by utilizing waste heat contained in brine from steam separators. Three different utilization methods are investigated by constructing thermodynamical models of different power cycles and optimizing the specific net power output for each cycle using conditions of hypothetical geothermal areas. The cycles investigated include a conventional double flash cycle, an organic Rankine bottoming cycle in parallel with a single flash cycle using isopentane as a working fluid and a modification of the double flash cycle involving an added recuperator between the geothermal brine and the steam at the high pressure turbine outlet. Also, a model of a single flash power plant is constructed for comparison. The specific net power outputs of the different cycles are compared along with the overall efficiency of the cycles. Finally, an economical analysis is performed to further compare the feasibility of the different cycles using conventional economical analysis to estimate the cost of the final product.

The results of the study can be used to compare the different conventional methods of utilizing geothermal energy for electrical power production with respect to the energy content, or the enthalpy, of the geothermal fluid produced from production wells. Also, a new method of utilizing geothermal brine by modifying the double flash cycle is introduced and compared to the conventional cycles. The new method could provide useful results for increased power production compared to the other cycles and also, as a future study, provide a method of better controlling the double flash cycle in case of changes in the fluid characteristics from production wells, which is expected in future operation time of geothermal power plants.

Útdráttur

Markmið þessa verkefnis er að bera saman aðferðir til þess að auka rafmagnsframleiðslu hefðbundinna eins þrepa hvellsuðuorkuvera (e. Single Flash Power Plants) með því að nýta betur varma í skiljuvatninu frá gufuskiljunum. Þrjár mismunandi vinnsluaðferðir veru greindar með því að útbúa varmafræðileg líkön af vinnslurásum fyrir hverja gerð orkuvers og raforkuframleiðslan bestuð á hverja massaeiningu úr vinnsluholum. Skilgreint er fræðilegt jarðhitasvæði með ákveðnum skorðum sem tekið er tillit til við bestun vinnuhringjanna. Vinnuhringirnir sem rannsakaðir voru eru sem hér segir; tveggja þrepa hvellsuðuorkuver (e. Double Flash Power Plant), hvellsuðuorkuver með tvívökva organic Rankine rás (e. Hybrid Single Flash - ORC Power Plant) sem nýtir skiljuvatnið til þess að hita og sjóða isopentan vinnuvökva og breytt tveggja þrepa hvellsuðuorkuver (e. Modified Double Flash Power Plant) með endurhitara sem yfirhitar gufuna úr háþrýstihverflinum. Einnig var byggt líkan af hefðbundun eins þrepa hvellsuðuorkuveri til samanburðar. Bestuð raforkuframleiðsla hvers hrings eru bornar saman ásamt nýtni orkuveranna. Að lokum er gert kostnaðarmat á framleiðslukostnaði fyrir hverja virkjun með aðferðum hagverkfræðinnar.

Niðurstöður verkefnisins geta nýst í að bera saman hefðbundna nýtingarmöguleika skiljuvatnsins frá eins þrepa hvellsuðuorkuverum miðað við vermi jarðhitavökvans sem um ræðir. Einnig er kynnt til sögunnar ný aðferð við að nýta skiljuvatnið með því að bæta við endurhitara í hina hefðbundnu uppsetningu á tveggja þrepa hvellsuðuorkuveri og sá hringur borinn saman við hina hefðbundnu hringi. Hin nýja aðferð gæti gefið áhugaverðar niðurstöður fyrir aukna raforkuframleiðslu og einnig, sem framhaldsverkefni, boðið upp á stýrimöguleika fyrir hinn hefðbundna tveggja þrepa hvellsuðuhring ef komi til verulegra breytinga á eiginleikum háhitasvæðisins eftir að vinnsla hefst úr því sem gjarnan gerist við nýtingu jarðvarma á jarðhitasvæðum.

Contents

1	Introduction	1
1.1	Description of the Study	1
1.2	Litterature Review	2
1.3	Prerequisites and Limitations of Study	2
1.4	Structure of the Thesis	3
2	Geothermal Energy	5
2.1	Geothermal Energy in Iceland	6
2.2	Geothermal Power Plants in Iceland	7
2.2.1	Flash-Steam Power Plants	8
2.2.2	Binary Cycle Power Plants	9
2.3	Geothermal Power Worldwide	9
3	Thermodynamical Modelling and Economical Analysis of Geothermal Power Plants	13
3.1	Modelling of a Single Flash Power Plant	13
3.1.1	Thermodynamics of a Single Flash Cycle	16

3.2	Modelling of a Double Flash Power Plant	23
3.2.1	Thermodynamics of the Double Flash Cycle	23
3.3	Modelling of an Organic Rankine Bottoming Unit	27
3.3.1	Thermodynamics of the Organic Rankine Bottoming Cycle . . .	30
3.4	A Modification of the Double Flash Power Plant	34
3.4.1	Thermodynamics of the Modified Double Flash Cycle	34
3.5	Economical Analysis	39
3.5.1	Capital and O&M Cost of the Power Plants	39
4	Optimization of Net Power Output	43
4.1	Optimization Variables and Constraints	44
4.1.1	Single Flash Cycle	44
4.1.2	Double Flash Cycle	44
4.1.3	Hybrid Single Flash Cycle with Organic Rankine Bottoming Unit	44
4.1.4	Modified Double Flash Cycle	45
4.1.5	Overview of the Optimization Problems	45
4.2	Wellhead Pressure Limitations	45
5	Results	47
5.1	Assumptions	47
5.1.1	Geothermal Reservoir	47
5.1.2	Power Plant Equipment	48
5.2	Optimization of Net Power Output	49

5.2.1	Single Flash Cycle	50
5.2.2	Double Flash Cycle	50
5.2.3	Single Flash Cycle with Organic Rankine Bottoming Unit . . .	53
5.2.4	Modified Double Flash Cycle	55
5.3	Comparison of the Power Cycles	58
5.3.1	Net Power Output and Efficiencies of the Cycles	58
5.3.2	Economical Comparison Between Cycles	62
6	Conclusion	67
6.1	Further Studies	68
	Bibliography	71

List of Tables

2.1	Installed power capacity of geothermal power plants in Iceland in January 2008 (Flóvenz, 2008)	8
2.2	Worldwide installed capacity of geothermal power plants in May 2007 (DiPippo, 2005)	10
3.1	Typical values for the scaling component α	40
4.1	Optimization variables and constraints for each power cycle	45
5.1	Isentropic efficiency of different power plant equipment	48
5.2	Overall heat transfer coefficient \bar{U} for various heat exchangers	48
5.3	Minimum pinch in various heat exchangers	49

List of Figures

2.1	<i>The formation of high temperature geothermal areas (Iceland GeoSurvey, 2008)</i>	6
2.2	<i>Geological map of Iceland showing the location of high and low temperature areas (Iceland GeoSurvey, 2008)</i>	7
2.3	<i>Overview of the different types of geothermal powerplants in 2007 by installed capacity for each type of plant (DiPippo, 2005)</i>	11
2.4	<i>Overview of the different types of geothermal powerplants in 2007 by number of units for each type of plant (DiPippo, 2005)</i>	11
3.1	<i>A schematic of a simple single flash power plant</i>	14
3.2	<i>A temperature - entropy diagram for a single flash power plant</i>	18
3.3	<i>A schematic of a double flash power plant</i>	24
3.4	<i>A temperature - entropy diagram for a double flash power plant</i>	25
3.5	<i>A schematic of the ideal Rankine cycle</i>	28
3.6	<i>A Temperature entropy diagram of the Ideal Rankine process with water as the working fluid.</i>	29
3.7	<i>A schematic of a single flash power plant with an ORC bottoming unit</i> 30	
3.8	<i>A temperature - entropy diagram of the Organic Rankine bottoming cycle using isopentane as a working fluid</i>	31

3.9	<i>A temperature - quality diagram of the heat exchange between the geothermal brine and the binary working fluid</i>	32
3.10	<i>A schematic of a modified double flash power plant with added recuperator</i>	35
3.11	<i>A temperature - entropy diagram for the reheat Rankine cycle</i>	36
3.12	<i>A temperature - entropy diagram for the modified double flash cycle . .</i>	37
4.1	<i>Productivity curves for geothermal wells producing two phase flow and liquid flow(Iceland GeoSurvey, 2008)</i>	46
5.1	<i>Optimized net power output from the single flash cycle</i>	50
5.2	<i>Optimized wellhead pressure for the single flash cycle</i>	51
5.3	<i>Steam quality at the turbine outlet for the single flash cycle</i>	51
5.4	<i>Optimized net power output from the double flash cycle</i>	52
5.5	<i>Optimized wellhead pressure and second flashing pressure for the double flash cycle</i>	52
5.6	<i>Steam quality at the turbine outlets for the two turbines in the double flash cycle</i>	53
5.7	<i>Optimized net power output from the hybrid single flash - ORC plant and the individual power output from each unit</i>	54
5.8	<i>Optimized wellhead pressure of the single flash unit and the optimum pressure of the isopentane in the ORC bottoming unit</i>	54
5.9	<i>Steam quality at the steam turbine outlet in the single flash unit in the hybrid single flash-ORC power plant</i>	55
5.10	<i>Optimized net power output from the modified double flash cycle and the individual power output from the high pressure turbine, W_{hp}, and the low pressure turbine, W_{lp}</i>	56
5.11	<i>Optimized wellhead pressure and second flashing pressure for the modified double flash cycle</i>	57

5.12	<i>Steam quality at the turbine outlets for the two turbines in the modified double flash cycle</i>	57
5.13	<i>The mass fraction of geothermal brine that is led from the separator to the recuperator in the modified double flash cycle</i>	58
5.14	<i>The temperature differences in both ends of the recuperator. T_7 and T_{13} are the temperatures at the inlet of the geothermal brine and outlet of the superheated steam, whereas T_8 and T_{12} are the temperatures of the return brine and the saturated steam respectively</i>	59
5.15	<i>A comparison of specific net power output for the different cycles that were modelled</i>	60
5.16	<i>A comparison of the optimum wellhead pressure for the the different power cycles</i>	61
5.17	<i>A comparison of the thermal efficiency for each cycle</i>	62
5.18	<i>A comparison of the product cost for each cycle</i>	63
5.19	<i>A comparison of the required mass flow from production wells to produce 100 MW_e for each cycle</i>	65

Nomenclature

A	heat exchanger area [m^2]
A_{equal}	Equal-amount money transactions [US\$]
C	cost [US\$]
\bar{c}	average specific heat [kJ/kg ·K]
c_p	specific heat [kJ/kg ·K]
CRF	Capital Recovery Factor
DDE	Dynamic Data Exchange
e	specific exergy [kJ/kg]
\dot{E}	exergy [kW]
EES	Engineering Equation Solver
f_{reheat}	mass fraction of brine to reheater
h	enthalpy [kJ/kg]
i	interest rate
LMTD	logarithmic mean temperature difference
M	molar mass [g/mol]
\dot{m}	mass flow [kg/s]
n	number of money transactions
NCG	non-condensable gas
O&M	operation and maintenance
ORC	organic Rankine cycle
p	pressure [bar],[Pa]
P	present worth [US\$]
\dot{Q}	heat [kW]
R	ideal gas constant [J/kg ·K]
s	entropy [kJ/kg ·K]
t	time [s]
T	temperature [°C],[K]
U	overall heat transfer coefficient [$W/m^2 \cdot K$]
\dot{W}	work [kW]
x	steam quality

Greek Symbols

α	scaling exponent
Δh	enthalpy change [kJ/kg]
ΔT	temperature change [°C],[K]
η	isentropic efficiency
η_{th}	thermal efficiency
η_{II}	second law efficiency

Nomenclature in Figures

B	boiler
BC	binary condenser
BT	binary turbine
C	condenser
CP	compressor
CT	cooling tower
G	Generator
HPS	high pressure steam separator
HPT	high pressure turbine
LPT	low pressure turbine
P	pump
R	recuperator
S	steam separator
T	turbine
TV	throttle valve
W	production well

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1 Introduction

1.1 Description of the Study

Geothermal energy has been utilized for centuries for heating and bathing and nowadays it is used for production of both heat and power. The possibilities for utilizing geothermal heat is constrained by the geology of each area and there are relatively few places worldwide where utilization of geothermal energy is currently economically viable. Iceland has an advantage over many other countries as its geographical position places it on the top of the mid-Atlantic ridge where the North America plate and the Eurasian plate are drifting apart. The drifting of the tectonic plates causes both volcanic and seismic activities that result in a formation of a high-temperature geothermal belt across the island. At such high-temperature areas, the possibility for geothermal power production is feasible (Pálmason, 2005).

Geothermal energy has been harnessed in Iceland for various purposes. Today, it is most commonly utilized for heating purposes and for electrical power production. The most common type of power plant in Iceland and one of the most common type of power plant worldwide is the single flash power plant. Various technical developments have been introduced to better utilize the energy potential in the geothermal fluid than is done in the single flash cycle. The main developments are the double flash cycle and the binary bottoming cycle (DiPippo, 2005). These cycles have the potential to improve the power production from geothermal areas considerably.

The aim of the study is to compare methods that increase the power generation from the conventional single flash cycle by utilizing the heat in the brine from the steam separator. Three different power cycles were examined by constructing thermody-

namical models of the cycles and optimizing their net specific power output for a given enthalpy from the production wells. These cycles are; a conventional double flash power cycle, a bottoming organic Rankine cycle coupled in parallel to the single flash cycle using isopentane as a working fluid and a modification of the double flash cycle with an added recuperator used to superheat the steam at the outlet of the high pressure turbine. The comparison of the cycles for the given enthalpy range of 1000 kJ/kg to 2500 kJ/kg can be used as an indicator for what type of power plant would be suitable for a certain geothermal area with respect to the enthalpy of the production wells and the corresponding net specific power output. A cost estimation of the production cost of electricity was also compared for the different cycles.

1.2 Literature Review

Single flash cycles produce about 43% (in May 2007) of all the electricity generated from geothermal energy worldwide (DiPippo, 2005). If the temperature of the brine from the steam separator in the single flash unit is high enough, the brine can be utilized further to produce more electricity. The double flash cycle uses the geothermal brine by producing excess steam in a second flashing stage. The double flash cycle has been shown to be able to produce up to 20-25% more power than the single flash cycle (Dagdaz, 2007).

Studies have shown that adding a binary unit as a bottoming cycle to the single flash unit in areas with low- and medium enthalpies can be preferable to a flashing unit when the composition of the geothermal fluid is likely to cause scaling in the power plant equipment after flashing and cooling of the fluid. Studies have also showed that the lowest cost per kilowatt hour would also be obtained using an organic Rankine unit instead of a second flashing unit, although the efficiency would be smaller (Moya and DiPippo, 2007). The bottoming binary cycle coupled to the single flash cycle has shown an increased power production of 13-28% compared to the conventional single flash cycle (Paloso Jr. and Mohantly, 1993). The Kalina technology has shown increased efficiency from the conventional organic Rankine cycles (Heppenstall, 1998; Wall et al., 1989). An organic Rankine cycle is often a more natural choice than the Kalina cycle since the technology is well known (Paloso Jr. and Mohantly, 1993).

1.3 Prerequisites and Limitations of Study

The modelling of the different power cycles in this study is not associated with a specific geothermal area that has known characteristics of the geothermal fluid and corresponding limitations to the possibilities of utilization. Geothermal areas can

have different characteristics depending on the geological conditions. The chemical content of the fluid can vary greatly, both regarding dissolved minerals from the rock formation and the amount of non condensable gases that travel with the fluid to the surface (Pálmason, 2005). The chemical content of the fluid results in some limitation of utilization possibilities, as the minerals can precipitate and cause damage to the power plant equipment due to scaling. Also, the amount of non condensable gasses directly affects the net power output of the power plants as they have to be removed from the energy conversion process with mechanical equipment such as a compressor or an ejector (Dickson and Fanelli, 2005). If the amount of non condensable gasses is too high, it can become economically unfeasible to produce power from the geothermal fluid.

In the modelling of the geothermal power plants in this study, the effects of scaling were not taken into consideration and it was assumed that the geothermal fluid could be cooled down and flashed freely in order to achieve the maximum power output for each power cycle. Also, the amount of non condensable gasses was assumed to be a constant value of 1% of the total mass flow from the wells.

Another factor that affects the possible utilization of the geothermal fluid is the productivity curve for the production well. The productivity curve describes the relationship between the wellhead pressure and the corresponding mass flow from the production well. At a certain pressure, the well stops to produce any mass flow so it is obvious that the wellhead pressure is a limiting factor in the power production process. Production wells can have different production curves depending on the characteristics of the geothermal reservoir. The maximum allowable wellhead pressure for the optimization of the different power cycles was estimated using information about typical productivity curves for production well at various locations in Iceland (Steingrímsson, 2007). The wellhead pressure in the power plant models was thus restricted to a maximum of 35 bar (Valdimarsson, 2008).

The results of the optimized specific power outputs for the different cycles can not be directly used as a measure of production possibilities for all geothermal areas. The models built in this study, on the other hand, can easily be adjusted to fit a specific geothermal area and take into account all the limitations associated with the reservoir. The results of the present work can only be used as a measure of comparison between the cycles and as an indication of the possible power output associating the given average enthalpy of the geothermal wells within reasonable bounds.

1.4 Structure of the Thesis

The thesis is structured as follows:

Chapter 2 serves as a discussion of geothermal energy in Iceland as well as the utilization of geothermal energy for electrical power production in Iceland and worldwide.

Chapter 3 introduces in details the thermodynamical modelling techniques used to model the production of electricity from the four different cycles; a single flash cycle, a double flash cycle, a single flash cycle with a binary bottoming unit and a modification of the double flash cycle. The chapter also discusses the economical calculations used to estimate the investment cost and the operation and maintenance cost used to estimate the production cost of the electricity produced in each cycle.

Chapter 4 describes the optimization methods used to optimize the specific power output for the different cycles.

Chapter 5 presents the results of the simulation and optimization and a comparison of the different power plants is made with respect to power output and economical considerations.

Finally, Chapter 6 includes conclusions and suggestions for future work.

2 Geothermal Energy

Geothermal energy is often referred to as the part of the Earth's heat that mankind is able to utilize. The total heat content of the earth is estimated to be $12.6 \cdot 10^{24}$ MJ and only a small fraction of that energy can be exploited by modern technology (Dickson and Fanelli, 2005). Exploitation of geothermal energy is mostly restricted to areas where geological conditions are favourable, where heat is carried to or near the surface in the form of a fluid creating geothermal hot spots.

Geothermal energy has been used by man for centuries for providing heat for various purposes. The first district heating system was constructed in France in the 14th century but long before that time had geothermal water been used for washing and bathing in countries where geothermal hot springs were to be found. The possibility of generating electricity by using geothermal energy was first discovered in the beginning of the 20th century at the geothermal field in Larderello in Tuscany, Italy, when experimental drilling showed that the temperature of the steam in the geothermal reservoir was high enough to be used to produce electricity in an economical way (Pálmason, 2005).

Since Larderello, technology for harnessing geothermal energy has been evolving and feasible areas for geothermal exploitation have at the same time been growing in numbers. New technologies for enhanced geothermal systems now make it possible to extract heat from the earth's crust where no heat carrier is present by nature and geothermal projects are increasing in popularity due to substantial growth in energy consumption and global environmental concerns. In May 2007, a total of about 9,500 MW_e were produced in geothermal powerplants worldwide and the direct use of geothermal energy e.g. for space heating and bathing accounted for 28,268 MW_t in the year 2005 (DiPippo, 2005; Lund, 2006).

In the following sections, the status of geothermal utilization in Iceland in particular and also worldwide will be discussed further.

2.1 Geothermal Energy in Iceland

Iceland has a vast potential to utilize geothermal energy as it is accessible in a large part of the country. The geothermal energy is originated from volcanic activity due to the tectonic plate boundary that forms the mid-Atlantic ridge that crosses the country, from the southwest region to the north. The high temperature geothermal areas are all formed within the plate boundary as the bedrock is highly fractured and allows access to magma intrusions in the earth's crust. Water circulation in the fractured bedrock transfers the heat provided by the hot rock or magma chambers to the surface creating so called hot spots as explained in figure 2.1. The temperatures of the high temperature geothermal areas are typically above 200 °C on 1 km depth (Pálmason, 2005).

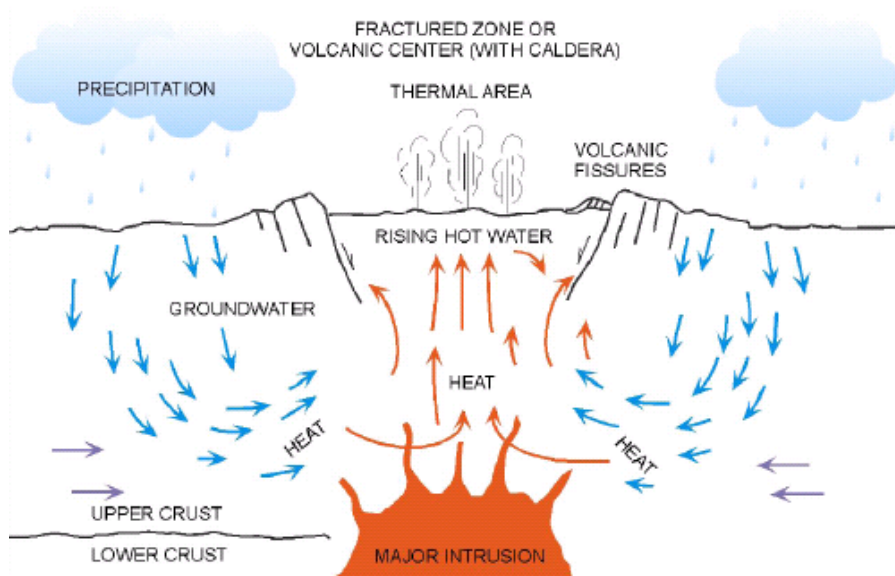


Figure 2.1: The formation of high temperature geothermal areas (Iceland GeoSurvey, 2008)

The plate boundaries result in both volcanic and seismic activity and the plates are drifting apart roughly one centimeter per year. As the plates drift apart, new crust is formed and older crust that was previously formed at the plate boundaries slowly moves away from the volcanic zones and cools down. The older crust is still fractured

and hot enough to be able to form low temperature geothermal areas. The low temperature areas are defined as the geothermal areas that have temperature below 150 °C on 1 km depth (Pálmason, 2005). Figure 2.2 shows the location of high and low temperature geothermal areas in Iceland

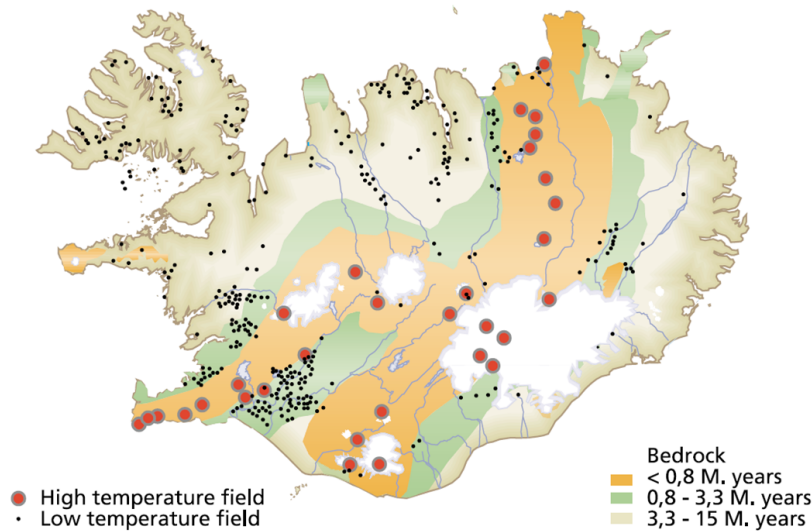


Figure 2.2: Geological map of Iceland showing the location of high and low temperature areas (Iceland GeoSurvey, 2008)

In 2006, 60% of the total primary energy consumption in Iceland was based on geothermal energy used for both power and heat production. Electricity generated in hydro power plants accounted for approximately 15% of the primary energy consumption and oil, mostly used for transportation, for about 17%. The last remaining 3% is energy from coal which is used as a raw material in ferro-silica production (Flóvenz, 2008). Thus, geothermal energy account for a vast majority of the primary energy consumption in Iceland. The utilization of geothermal energy is of various forms and apart from electricity production and district heating systems, geothermal energy is used in swimming pools, for snow melting, fish farming, in greenhouses and for industrial processes (Flóvenz, 2008).

2.2 Geothermal Power Plants in Iceland

An overview of the existing geothermal power plants in Iceland is given in table 2.1 along with the installed capacity of the plants and the power plant type. The most

common type of power plant in Iceland is the single flash plant, with a total of about 290.5 MW of installed capacity. Double flash plants are operating in two places in Iceland, at the Hellisheiði and Krafla geothermal power plants, with the combined installed capacity of 180 MW. The binary plants such as organic Rankine cycle and Kalina power plants are not common in Iceland, with only one Kalina power plant in operation in Húsavík producing 2 MW of electricity and binary units at Svartsengi producing in total about 8.5 MW of electricity. The design of the different power plants that exist in Iceland will be discussed in general in the following sections.

Table 2.1: Installed power capacity of geothermal power plants in Iceland in January 2008 (Flóvenz, 2008)

Power plant	Instaled capacity	Power plant type
Hellisheiði	120 MW	Double flash
Nesjavellir	120 MW	Single flash
Reykjanes	100 MW	Single flash
Svartsengi	76 MW	Single flash, ORC
Krafla	60 MW	Double flash
Bjarnarflag	3 MW	Single flash, back pressure
Húsavík	2 MW	Kalina

2.2.1 Flash-Steam Power Plants

Flash steam power plants are designed to produce power from geothermal fluid that exists as a mixture of liquid and steam. The term flash steam power plant is derived from the process when the fluid that is extracted from the bottom of the geothermal well undergoes a flashing process on its way up to the surface, converting it further into a two-phase mixture. Production wells producing a two-phase flow are the most common geothermal wells in Iceland (Pálmason, 2005).

The most common type of flash steam power plant is the single flash plant, where the liquid and steam from the production wells are separated in a steam separator. The steam is then led through a turbine which converts the thermal energy in the steam into mechanical energy used to produce electricity. The liquid, or the geothermal brine, that is drained from the steam separator is exposed of either into the surrounding environment or reinjected into the ground (DiPippo, 2005).

A double flash plant takes further advantage of the geothermal brine by introducing a second flashing process after the single flash process. The brine from the steam separator undergoes a throttling process that decreases the pressure of the saturated fluid so that a two phase mixture is achieved at a lower pressure. The two phase

flow produced in the throttling process is then led through a second steam separator where the low pressure steam is separated from the brine and led towards a low pressure turbine. The overall efficiency of the double flash power cycle is higher than the efficiency of the single flash cycle but the need for extra equipment makes the double flash cycle more expensive (Dickson and Fanelli, 2005).

A more complete discussion of the flash-steam power plants is found in Sections 3.1 and 3.2 where the thermodynamical equations needed for analysis of the cycles are introduced.

2.2.2 Binary Cycle Power Plants

Binary cycle power plants use a different working fluid than water. The binary fluid extracts heat from the geothermal fluid in a heat exchanger. The main advantage of binary plants is the possibility to utilize lower-grade geothermal fluid with low to medium temperatures. Another practical property of the binary geothermal power plants is that they allow utilization of geothermal fluid that has a high concentration of dissolved chemicals that could produce scaling if utilized in conventional flash power plants (Dickson and Fanelli, 2005).

The binary plants are based on the conventional organic Rankine cycle and a number of possible working fluids are available for optimum design in each case. In Iceland, two binary plants exist; a binary plant in Svartsengi power plant that uses isopentane as a working fluid and a Kalina plant in Húsavík that uses the Kalina technology with a mixture of ammonia and water as a working fluid. In this project, a binary plant using isopentane is modelled and the methods are described further in Section 3.3

2.3 Geothermal Power Worldwide

Geothermal energy is utilized for power production all over the world and a total of 23 countries were producing electricity in geothermal power plants in the year 2007 (DiPippo, 2005). An overview of the installed capacity in each of these countries is listed in table 2.2.

The countries with the greatest production of electricity from geothermal power plants are the United States and the Philippines. Although utilization of geothermal energy for power production is steadily increasing, it accounts for only less than a half percent of the total electricity generated worldwide. The growth of geothermal power production has experienced a series of different development periods. The 1973 and

Table 2.2: Worldwide installed capacity of geothermal power plants in May 2007 (DiPippo, 2005)

Country	MW_e
Australia	0.15
Austria	1.25
China	27.6
Costa Rica	163
El Salvador	204.3
France (Guadeloupe)	14.7
Germany	0.2
Guatemala	44.6
Iceland	422.4
Indonesia	807
Italy	811.2
Japan	537.74
Kenya	130.2
Mexico	953.3
New Zealand	572.1
Nicaragua	108.9
Papua New Guinea	56
Philippines	1979.9
Portugal (The Azores)	16
Russia (Kamchatka)	79
Thailand	0.3
Turkey	27.8
USA	2555.5
Total	9513.15

1979 oil crisis drove countries to explore geothermal energy utilization, resulting in a vast increase in the installed capacity but the annual growth has been deteriorating ever since and from 1985 to 2007 the annual growth rate was measured only at 3.2%. However, the recent increase in oil prices could lead to an increase in the annual growth rate in the coming years (Bertani, 2005; DiPippo, 2005). Although the worldwide annual growth rate of geothermal power produced has been small, countries like Costa Rica, France, Kenya, Mexico, Nicaragua, Russia and Iceland have all increased significantly their geothermal power production percentage wise in the years 2000-2005, with an increase varying from 11% to 250% of previously installed capacity (Bertani, 2005).

The most common types of geothermal power plants worldwide are the single flash cycle and the binary cycles. The total installed capacity of single flash power plants

was 4015.39 MW_e in May 2007 which accounted for about 43% of the total installed capacity of geothermal power plants worldwide. At the same time, binary cycles only accounted for 4% of the total installed electrical capacity. Binary power plant units usually produce electricity ranging from a few hundred kilowatts to several megawatts while single flash unit can produce up to 110 MW_e per unit (Dickson and Fanelli, 2005). Statistics for the generated electricity from the different types of geothermal power plants and for the number of units for each power plant type worldwide are shown on Figures 2.3 and 2.4 (DiPippo, 2005).

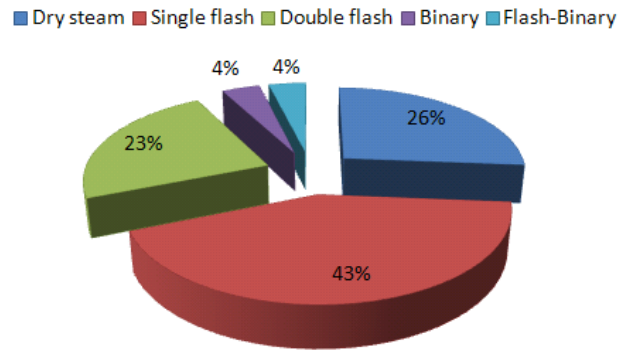


Figure 2.3: Overview of the different types of geothermal powerplants in 2007 by installed capacity for each type of plant (DiPippo, 2005)

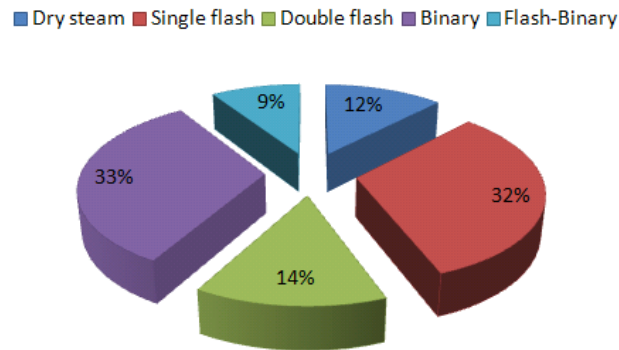


Figure 2.4: Overview of the different types of geothermal powerplants in 2007 by number of units for each type of plant (DiPippo, 2005)

3 Thermodynamical Modelling and Economical Analysis of Geothermal Power Plants

3.1 Modelling of a Single Flash Power Plant

Single flash power plants accounted for about 29% of all geothermal power plants in 2004 but generated almost 40% of the electricity produced from geothermal energy. This type of power plant is often the first power plant installed at newly developed liquid dominated geothermal areas (DiPippo, 2005).

As the name implies, the geothermal fluid from the production wells undergoes a single flashing process, i.e. a process where pressurized fluid is converted into a mixture of steam and liquid by decreasing the pressure below the saturation pressure corresponding to the temperature of the fluid. The flashing can occur in different places in the process;

- in the reservoir itself as a result of pressure drops caused by friction when the fluid flows through permeable rock formations
- in a production well due to pressure drops and pressure heads while the fluid flows up to the surface, and
- at the inlet of the steam separator by a throttling process controlled by a valve or an orifice plate (DiPippo, 2005)

A simplified schematic of a single flash power plant is shown in Figure 3.1. The figure shows the most important equipment that affects the thermodynamics of the energy conversion process. The two phase mixture flows from the production well, W, and is led through a steam separator, S. In a vertical separator as shown on the schematic, the mixture enters a cyclonic vessel and due to centrifugal forces and gravitation, the liquid phase is pressed against the separator walls and drained from the vessel as the steam is trapped in the middle of the separator and can be collected and led through towards the power house which contains the turbine, T, and the electrical generator, G.

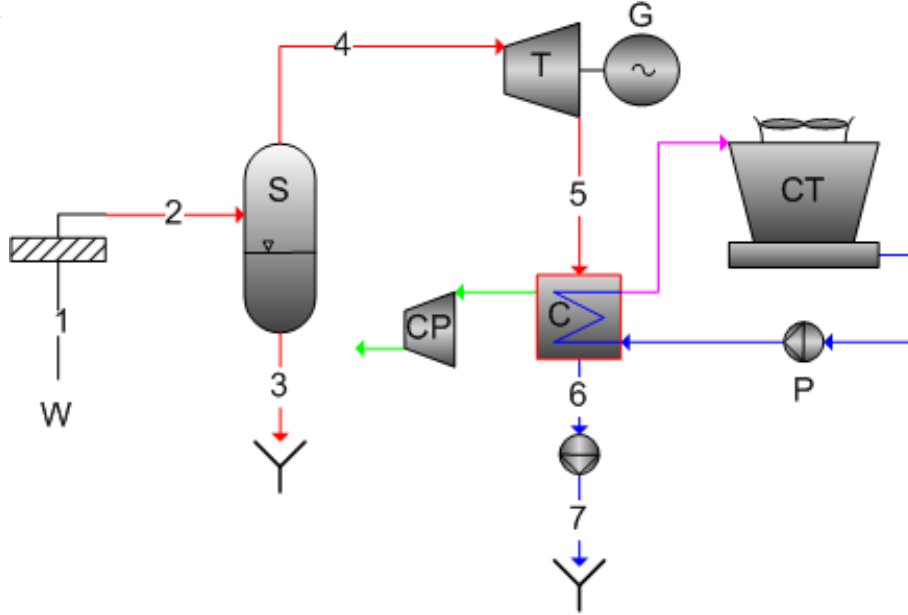


Figure 3.1: A schematic of a simple single flash power plant

In the separation process, it is important that the steam collected is relatively dry because liquid entrapped in the steam can cause both scaling and/or erosion of piping and the turbine. To prevent excessive moisture in the steam, the steam is often led through moisture removers prior to entering the turbine inlet. Also, if the separator station is located at some distance from the power house, traps in the piping components are implemented to capture any droplets formed by condensation at the pipe walls prior to entering the turbine. This moisture removing equipment is not shown on the schematic for the reason of simplification.

The turbine, T, transforms the thermal energy in the pressurized steam into useful mechanical work when the steam hits the turbine blades causing pressure drop of the steam (expansion) and a rotary motion in the turbine. The rotary motion can then be converted to electricity in the generator, G. Turbine sizes in single flash

power plants are usually less than 55 - 60 MW but steam turbine units of up to 100 MW have been designed for geothermal powerplants. The low density of the steam at the pressure range restricted by the geothermal reservoir results in these size limitations of the steam turbines, whereas steam turbines in conventional fuel-fired power plants are commonly in the size range of 600 to 1,000 MW (Dickson and Fanelli, 2005). The isentropic efficiency for a geothermal turbine is typically 81 - 85% and the mechanical efficiency of the turbine generator is approximately 96.3% (Dickson and Fanelli, 2005). Due to pressure drops in the steam across the steam turbine, the steam quality also decreases and small droplets can form that can damage the turbine blades. Thus, the pressure drop across the turbine is limited to produce no less than 85% quality of steam at the steam outlet. This criteria affects the allowable pressure at the high pressure end of the turbine as the low pressure end is usually restricted by the condensing unit, C. Due to the presence of gases in the steam, such as carbon dioxide (CO_2), hydrogen sulfide (H_2S) and hydrogen (H_2), a corrosive environment can occur and the turbines in geothermal power plants must be made out of corrosion resistant materials (Thorbjörnsson, 1995).

At the turbine outlet, the steam is discharged to a condensing chamber, C, that is maintained at a very low absolute pressure, typically at around 0.10 bar. These condensing exhaust steam turbines are far more common than atmospheric exhaust steam turbines, where the exhaust from the turbine is discharged directly to the atmosphere at atmospheric pressure. The thermodynamic improvement (greater pressure drop across the condensing turbine) of the condensing steam turbine leads to approximately twice as much power generated than from the back pressure turbine (Dickson and Fanelli, 2005).

Shell- and tube condensers are generally used in geothermal power plants where the amount of non-condensable gases is high (Dickson and Fanelli, 2005). The shell- and tube condensers allow physical and chemical separation between the geothermal steam and the cooling water that allows more effective removal and treatment of the non-condensable gases (DiPippo, 2005). The heat exchangers in this study were assumed to be conventional counterflow heat exchangers and thus modelled without correction factors needed for modelling shell- and tube heat exchangers.

An important part of the condensing process is the extraction of non-condensable gases (NCG) that are mixed within the geothermal fluid and travel with the steam through the energy conversion process. The non-condensable gases are mostly carbon dioxide (CO_2 , often about 95% (Thorbjörnsson, 1995)), hydrogen sulfide (H_2S) and hydrogen gas (H_2) and the composition varies depending on the different geothermal fields. The amount of non-condensable gases can range from a low gas content of about 1.5% of the total mass flow of steam from the wells up to over 12% (Dickson and Fanelli, 2005). These gases accumulate in the condenser and must be pumped out of the condensing unit using some gas extraction equipment such as a gas compressor, shown on Figure 3.1 marked with the symbol CP. If the gasses are not pumped out

of the condenser, the pressure in the condenser will accumulate and thus the power output of the power plant will decrease (DiPippo, 2005).

The cooling tower circulates cooling water to the condensers and where the cooling water extracts heat from the geothermal steam during the condensation process. The most common cooling towers use partial evaporation in the presence of moving airstream to cool the cooling water to near the wet-bulb air temperature (DiPippo, 2005). Common types of evaporation cooling towers are the crossflow and counterflow cooling towers. In the crossflow design, the air flow is directed perpendicular to the cooling water flow but in the counterflow design, the air flow is directed in opposite direction of the cooling water flow. In both designs, the interaction of the air and water flow allow a partial equalization and evaporation of the cooling water and the air becomes saturated with water vapour as it is discharged from the cooling tower. After the interaction with the air, the water is collected in basins at the bottom of the cooling tower and then recirculated in the cooling water circuit (Wikipedia, 2008a).

In the following section, the thermodynamical equations used for modelling of each component in the single flash power plant are described and discussed further.

3.1.1 Thermodynamics of a Single Flash Cycle

Thermodynamic balancing equations

The analysis of the single flash cycle is based on thermodynamical equations that describe the energy and mass conservation in each power plant component described above. The mass balance equations take into account that the mass flow, \dot{m} , into a component must be equal to the mass flow out of the component in a system at steady state conditions. The energy balance equations state that the flow of energy into a component must equal the energy flow out of the component. The energy transfer in and out of the component can occur by means of:

- Energy by fluid flow, expressed as $\dot{m}h$, accompanying the fluid as it enters or exits the component
- Work, \dot{W} , performed or consumed
- Heat, \dot{Q} , flowing in or out of a component

For the thermodynamical modelling of each component in an energy conversion system, the control volume method is applied where the principles of mass and energy conservation is used for all entering and exiting streams to and from the control volume boundary (Moran and Shapiro, 2002).

Thermodynamical terminology

The processes undergone in each component of the geothermal powerplant are based on various assumptions and simplifications that can be described thermodynamically as follows:

- **Isentropic process** is a process where the entropy, s , remains a constant. Modelling equations for turbines, pumps and compressors make use of this assumption which is relevant when work is delivered or consumed without any losses to the environment throughout the process.
- **Isenthalpic process** is based on the assumption that the enthalpy, h , is constant throughout the process. In isenthalpic processes, no work nor heat is delivered or consumed to/from the environment and the energy content of the fluid remains constant. Throttle valves are an example of components that are modelled as an isenthalpic process. Mixing of two or more working fluid streams is also an isenthalpic process.
- In an **isobaric process**, the pressure p remains constant. This idealistic assumption is used for modelling of steam separators and heat exchangers.

The processes described above are useful for estimating thermodynamic state of the working fluid entering and exiting different power plant components (Pálsson, 2006).

Temperature - Entropy process diagram

A temperature - entropy diagram for the single flash cycle can be seen in Figure 3.2. The numbering on the diagram corresponds to the numbering in Figure 3.1.

Flashing process

The cycle begins with a pressurized geothermal fluid at state 1 seen on the schematic on Figure 3.1 and the T-s diagram on Figure 3.2. The flashing is modeled as an isenthalpic process, where the enthalpy of the fluid does not change in the flashing procedure because no work or heat is extracted or added during the process. Changes in kinetic and potential energy are also neglected, thus,

$$h_1 = h_2 \quad (3.1)$$

Separation process

In the separation process, it is assumed that no pressure change occurs so the process

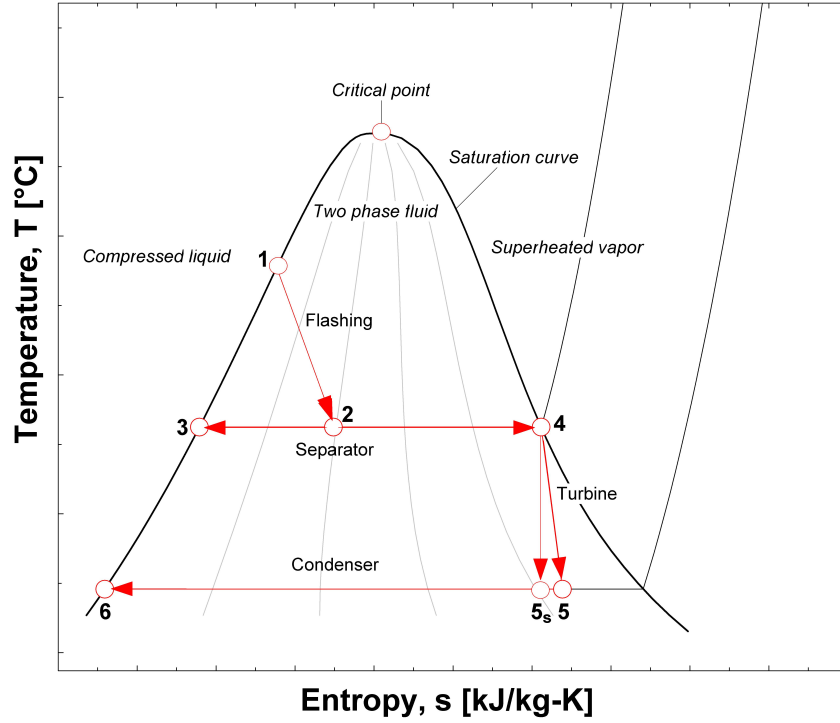


Figure 3.2: A temperature - entropy diagram for a single flash power plant

is isobaric. The quality of the steam, x , is determined by

$$x_2 = \frac{h_2 - h_3}{h_4 - h_3} \quad (3.2)$$

where h_3 and h_4 are the saturated liquid enthalpy and the saturated steam enthalpy respectively, according to the separator pressure. Equation 3.2 gives the mass fraction of the steam that enters the turbine after the separation process where the mass flow of separator water and steam is defined as follows:

$$\dot{m}_3 = (1 - x_2)\dot{m}_1 \quad (3.3)$$

$$\dot{m}_4 = x_2\dot{m}_1 \quad (3.4)$$

Turbine expansion process

In an ideal turbine the steam undergoes an isentropic process where the entropy s is constant as seen in Figure 3.2 for the process from 4 to 5_s . The enthalpy at 5_s is then calculated from the pressure and the entropy at state 5_s . In reality, the turbine is not isentropic as the entropy increases to state 5 during the expansion. The isentropic efficiency, η_t , is defined as:

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}} \quad (3.5)$$

If the isentropic efficiency is known or estimated, Equation 3.5 is used to calculate the enthalpy at state 5. Isentropic efficiency of turbines in a geothermal power plant can range from 81 to 85% (Dickson and Fanelli, 2005)

The power output of the turbine will then be

$$\dot{W}_t = \dot{m}_4(h_4 - h_5) \quad (3.6)$$

which is the total mechanical power developed by the turbine. The total electricity generated will then be

$$\dot{W}_e = \eta_g \dot{W}_t \quad (3.7)$$

where η_g is the generator efficiency. The net power output of the cycle is obtained by subtracting all auxiliary power needed for driving pumps, cooling tower fans and other electrical equipment present at the power plant.

Condensing process

The condenser is modelled as an counterflow heat exchanger. The mass flow of cooling water required to condense the steam in the condenser is calculated by using the energy balance equation for the heat exchanger:

$$\bar{c}\Delta T \dot{m}_{cooling} = \dot{m}_4(h_5 - h_6) \quad (3.8)$$

where \bar{c} is the average specific heat of the cooling water over the temperature difference in and out of the condenser, ΔT . The average specific heat of the cooling water is normally taken as $\bar{c} = 4.2 \text{ kJ/kg} \cdot \text{K}$.

In order to determine the size of the heat exchanger to be used in cost evaluations, the overall heat transfer coefficient, U , must be calculated or estimated for the heat exchanger. The following equations can then be used to calculate the total heat exchange area needed for the cooling requirements. The cooling requirement is calculated as

$$\dot{Q}_{cooling} = (h_5 - h_6)\dot{m}_4 \quad (3.9)$$

and the total heat exchange area of the condenser is calculated from

$$A = \frac{\dot{Q}_{cooling}}{U \cdot LMTD} \quad (3.10)$$

Where $LMTD$ is the logarithmic mean temperature for the heat exchanger. For a counter flow heat exchanger, the logarithmic mean temperature is defined as

$$\frac{(T_5 - T_{c,out}) - (T_6 - T_{c,in})}{\ln((T_5 - T_{c,out})/(T_6 - T_{c,in}))} \quad (3.11)$$

where $T_{c,in}$ and $T_{c,out}$ are the inlet and outlet temperatures of the cooling water (DiPippo, 2005).

Gas extraction process

Since the high-temperature geothermal fluid always contains some amount of non-condensable gasses, there is a need for gas extraction in the condenser. During the extraction process, some amount of steam will always be extracted along with the non-condensable gasses because the steam is mixed along with the gasses. The gas mixture is then assumed to be saturated with steam when it is extracted from the condenser. If the gas content of the geothermal fluid is known, the mass flow of steam travelling with the non-condensable gasses can be calculated as (Pálsson, 2006)

$$\dot{m}_v = \frac{M_v p_s}{M_g (p_c - p_s)} \dot{m}_g \quad (3.12)$$

where \dot{m}_g , \dot{m}_v , M_g and M_v are the mass flows and molar masses of the gas and vapour respectively, p_c is the condenser pressure and p_s is the saturation pressure of steam at the gas outlet temperature, T_s . In order to minimize the amount of steam traveling with the gasses out of the condenser, the temperature T_s at the gas outlet should be

lower than the average temperature in the condenser so that most of the vapour phase would be condensed before entering the gas extraction process (Pálsson, 2006).

The power needed for compressing the gasses can be calculated as an ideal isentropic process between the condenser pressure, p_c , and the atmospheric pressure, p_{atm} . The properties of the gas mixture are calculated as follows:

$$c_p = c_{pg} + (c_{pv} - c_{pg}) \frac{p_s M_v}{p_c (M_g + M_v)} \quad (3.13)$$

$$R = R_g + (R_v - R_g) \frac{p_s M_v}{p_c (M_g + M_v)} \quad (3.14)$$

where c_p is the specific heat of the gas and vapour mixture that is pumped out of the condenser and R is the ideal gas constant for the mixture. The ideal enthalpy change of the fluid when compressed to atmospheric pressure is given as

$$\Delta h = c_p T_s \left(\left(\frac{p_{atm}}{p_c} \right)^{\frac{R}{c_p}} - 1 \right) \quad (3.15)$$

The required pumping power is then calculated as

$$\dot{W}_c = (\dot{m}_g + \dot{m}_v) \frac{\Delta h}{\eta_c} \quad (3.16)$$

where η_c is the compressor efficiency.

Pumping power requirements

In order to calculate the power needed for the pumps in the power plant, the pressure head produced by pumping from the low pressure state p_{low} to the higher pressure state p_{high} needs to be known. Then, the pumping power requirements are calculated by

$$\dot{W}_p = \frac{v \cdot (p_{high} - p_{low}) \cdot \dot{m}_{wf}}{\eta_p} \quad (3.17)$$

where \dot{W}_p is the pumping power, v is the specific volume of the working fluid that needs to be pumped, p_{high} and p_{low} are the pressure states in kPa, \dot{m}_{wf} is the mass

flow of the working fluid and η_p is the pump efficiency taken to be 50% in this project. The enthalpy of the pressurized state after the pumping process can be calculated as:

$$h_{\text{high}} = h_{\text{low}} + \frac{\dot{W}_p}{\dot{m}_{\text{wf}}} \quad (3.18)$$

where h_{low} is the enthalpy at the low pressure state and h_{high} is the enthalpy at the high pressure state. The energy content of the fluid has thus been increased by the amount of energy that was needed for the pumping process. Equations 3.17 and 3.18 can be used for every pump located within the power plant.

Efficiency

The thermal efficiency of the single flash cycle can be calculated as the rate between the power output and the heat added to the cycle (Moran and Shapiro, 2002):

$$\eta_{th} = \frac{\dot{W}_e}{\dot{Q}_{\text{in}}} \quad (3.19)$$

where \dot{W}_e is defined in Equation 3.7 and \dot{Q}_{in} can be calculated as

$$\dot{Q}_{\text{in}} = h_1 \dot{m}_1 \quad (3.20)$$

The performance of the entire plant can be estimated using the second law of thermodynamics by comparing the actual power output to the maximum theoretical power that could be produced from the given geothermal fluid, or the exergy of the geothermal fluid. The specific exergy, e , of the geothermal fluid at a given pressure, p , and temperature, T , in the presence of an ambient temperature, T_0 , and ambient pressure, p_0 , is given as:

$$e = h(T, p) - h(T_0, p_0) - T_0(s(T, p) - s(T_0, p_0)) \quad (3.21)$$

The exergy, or the maximum theoretical power that could be extracted from the fluid is then

$$\dot{E} = \dot{m}_1 e \quad (3.22)$$

and the second law (exergetic) efficiency is defined as

$$\eta_{II} = \frac{\dot{W}_{\text{net}}}{\dot{E}} \quad (3.23)$$

3.2 Modelling of a Double Flash Power Plant

The double flash cycle is a thermodynamical improvement of the single flash cycle, where the waste heat in the geothermal brine from the separator is flashed in a throttling process that decreases the pressure of the brine allowing it to boil to produce steam that can then be used to drive a low pressure turbine. A schematic of a typical double flash cycle is shown in Figure 3.3.

Double flash powerplants can be found in various countries around the world and they account for about 1.5% of all geothermal power plants (DiPippo, 2005). In Iceland, this type of powerplant has been installed at Hellisheiði and in Krafla.

The process is similar to the single flash cycle discussed in Section 3.1. After the geothermal fluid has been separated in the high pressure steam separator, HPS, the brine is flashed by using a throttle valve, TV, that decreases the pressure of the brine and a two phase fluid is produced. The two phase fluid is then led through the low pressure separator, LPS, to separate the steam from the brine. The brine is then disposed off to reinjection wells or out to the natural surroundings. The low pressure steam from the separator is then combined with the steam from the high pressure turbine, HPT, which has been set to operate at the same low pressure conditions as the low pressure separator. The combined mass flow of steam is then led through a moisture remover (not shown on the figure for simplification) before entering the low pressure turbine, LPT, producing a pressure drop down to the condenser pressure at 0.1 bar. The condensing process, non-condensable gas extraction and cooling circuit processes follow the same procedure as described for the single flash cycle, see Section 3.1.

3.2.1 Thermodynamics of the Double Flash Cycle

The temperature-entropy diagram for the double flash cycle can be seen in Figure 3.12. The thermodynamical equations for the double flash cycle are similar to the equations used to model the single flash cycle. Below, only the components present in the double flash cycle that are not common with the components in the single flash cycle will be discussed. The equations in Section 3.1 for the mutual components

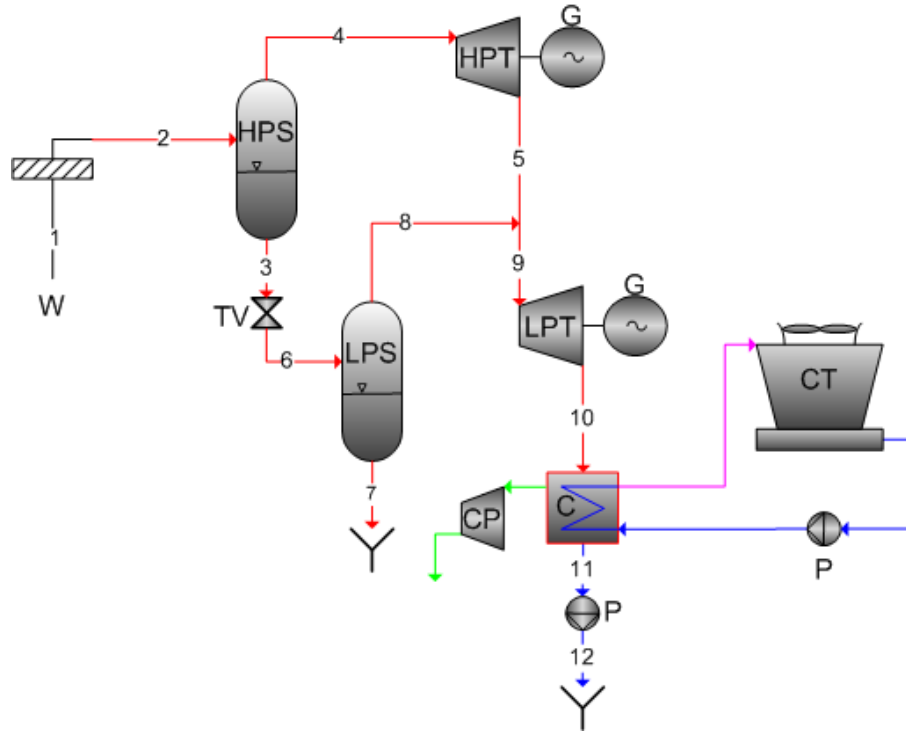


Figure 3.3: A schematic of a double flash power plant

between the single flash and double flash cycles are valid for modelling of the double flash cycle. The mutual components include the high pressure separator HPS, the high pressure turbine HPT, the cooling tower CT, the condenser C, the gas extraction unit CP and the pumps P.

Flashing and separation processes

The second flashing process in the double flash cycle is modelled the same way as the flashing process in the single flash cycle. The thermodynamical equations are based on that the process is isenthalpic, thus

$$h_3 = h_6 \quad (3.24)$$

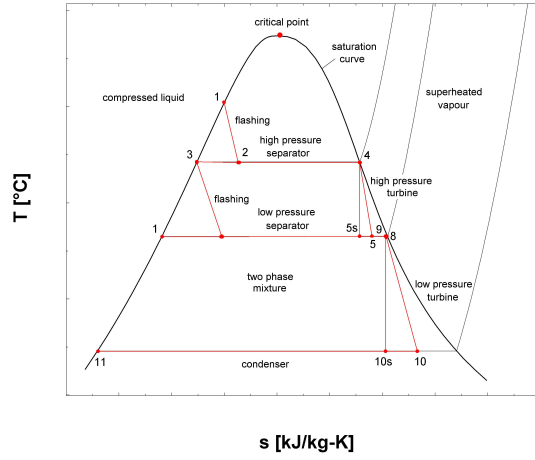


Figure 3.4: A temperature - entropy diagram for a double flash power plant

and the steam quality after the flashing becomes

$$x_6 = \frac{h_6 - h_7}{h_8 - h_7} \quad (3.25)$$

where h_7 is the enthalpy of the saturated liquid at state 7 and h_8 is the enthalpy of the saturated steam at state 8 given at the separation pressure p_6

The mass flow of liquid and steam after the separation process are given respectively as

$$\dot{m}_7 = (1 - x_6)\dot{m}_6 \quad (3.26)$$

and

$$\dot{m}_8 = x_6\dot{m}_6 \quad (3.27)$$

Expansion process in the high- and low- pressure turbines

The mass flows from the high pressure turbine and the low pressure separator are combined into a single stream at state 9 before entering the low pressure turbine. The mixing of streams 5 and 8 is modelled with both energy and mass conservation equations as

$$\dot{m}_{\text{mixed}} = \dot{m}_5 + \dot{m}_8 \quad (3.28)$$

$$\dot{m}_{\text{mixed}} h_{\text{mixed}} = \dot{m}_5 h_5 + \dot{m}_8 h_8 \quad (3.29)$$

Before entering the low pressure turbine, excess moisture is removed in a moisture remover. The steam quality after the mixing of streams is calculated as:

$$x_{\text{mixed}} = \frac{h_{\text{mixed}} - h_{\text{moisture}}}{h_9 - h_{\text{moisture}}} \quad (3.30)$$

Here, h_{moisture} is the enthalpy for saturated liquid and h_9 is the enthalpy for saturated steam at state 9.

Then, the mass flow of steam entering the turbine, \dot{m}_9 and the mass flow of moisture removed, $\dot{m}_{\text{moisture}}$ are defined as:

$$\dot{m}_{\text{moisture}} = (1 - x_{\text{mixed}}) \dot{m}_{\text{mixed}} \quad (3.31)$$

and

$$\dot{m}_9 = x_{\text{mixed}} \dot{m}_{\text{mixed}} \quad (3.32)$$

The power generated in the high- and low pressure turbines is calculated the same way as in Section 3.1 where the power output of the high pressure turbine is defined as

$$\dot{W}_{\text{hpt}} = \dot{m}_4 (h_4 - h_5) \quad (3.33)$$

and the low pressure turbine produces a power output of

$$\dot{W}_{\text{lpt}} = \dot{m}_9(h_9 - h_{10}) \quad (3.34)$$

The enthalpy h_5 is calculated as before in Equation 3.5 where the isentropic efficiency of the turbine, $\eta_t = \eta_{\text{hpt}}$, is known. Similarly, the enthalpy h_{10} is calculated as follows:

$$\eta_{\text{lpt}} = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \quad (3.35)$$

The total power generated in the double flash cycle is then

$$\dot{W}_{\text{total}} = \dot{W}_{\text{hpt}} + \dot{W}_{\text{lpt}} \quad (3.36)$$

and the total electricity generated is

$$\dot{W}_e = \eta_g \dot{W}_{\text{total}} \quad (3.37)$$

The net electrical power output of the cycle is calculated by subtracting the electricity requirements of pumps, fans and compressors that are present at the power plant.

3.3 Modelling of an Organic Rankine Bottoming Unit

Adding a binary bottoming cycle is an alternative utilization method to the double flash cycle to utilize the waste heat in the brine from a single flash cycle. This alternative could be preferable to the double flash cycle if mineral concentration in the brine is relatively high which could cause scaling if the brine would be flashed as it would in the double flash cycle.

An organic Rankine cycle is a binary cycle using an organic working fluid such as isopentane or isobutane to produce power. Organic Rankine cycle is based on the ideal Rankine cycle that operates in a closed loop cycle that includes a boiler, a turbine, a condenser and a pump as seen in Figure 3.5 with water as the working fluid. A temperature-entropy diagram of the ideal Rankine cycle is shown in Figure 3.6. The cycle is based on the following series of internally reversible processes:

Process 1-2: Isentropic compression in a pump.
 Process 2-3: Constant pressure heat addition in a boiler.
 Process 3-4: Isentropic expansion in a turbine.
 Process 4-1: Constant pressure heat rejection in a condenser.

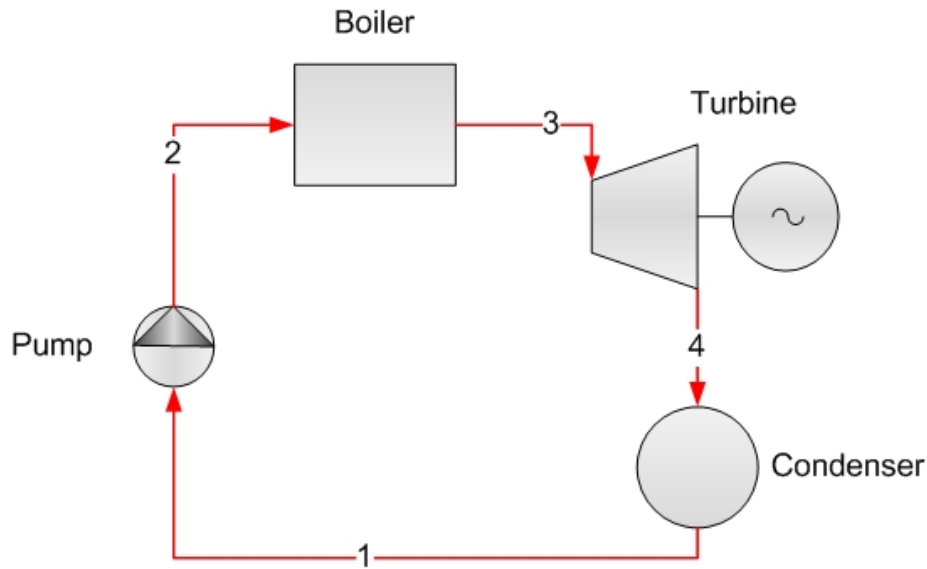


Figure 3.5: A schematic of the ideal Rankine cycle

The actual Rankine cycle differs from the ideal Rankine cycle due to irreversibilities in various components such as fluid friction and heat loss to the surroundings (Cengel and Boles, 2006). Another important cause for irreversibilities are the pump and the turbine, that cannot operate with 100% efficiency as the turbine produces less work output and the pump requires more work input due to irreversibilities. Under ideal conditions, these processes are isentropic as shown with the red lines on the T-s diagram in Figure 3.6, but in reality the processes are not isentropic and cause an increase in entropy, making the vertical lines $1 - 2_s$ and $3 - 4_s$ become skewed (lines $1 - 2$ and $3 - 4$).

An Organic rankine cycle uses an organic working fluid instead of water in the Rankine cycle. The organic fluids have an advantage over water as a working fluid due to the shape of the saturation curve as seen for isopentane in Figure 3.8. The shape of the curve eliminates the need for superheating, as is required when water is used as a working fluid since the vapour becomes superheated at the outlet of the turbine. This results in avoiding any formation of moisture during the turbine expansion and thus reducing erosion of the turbine blades. The organic working fluids also typically have lower boiling temperatures than water, making them well suitable for utilizing

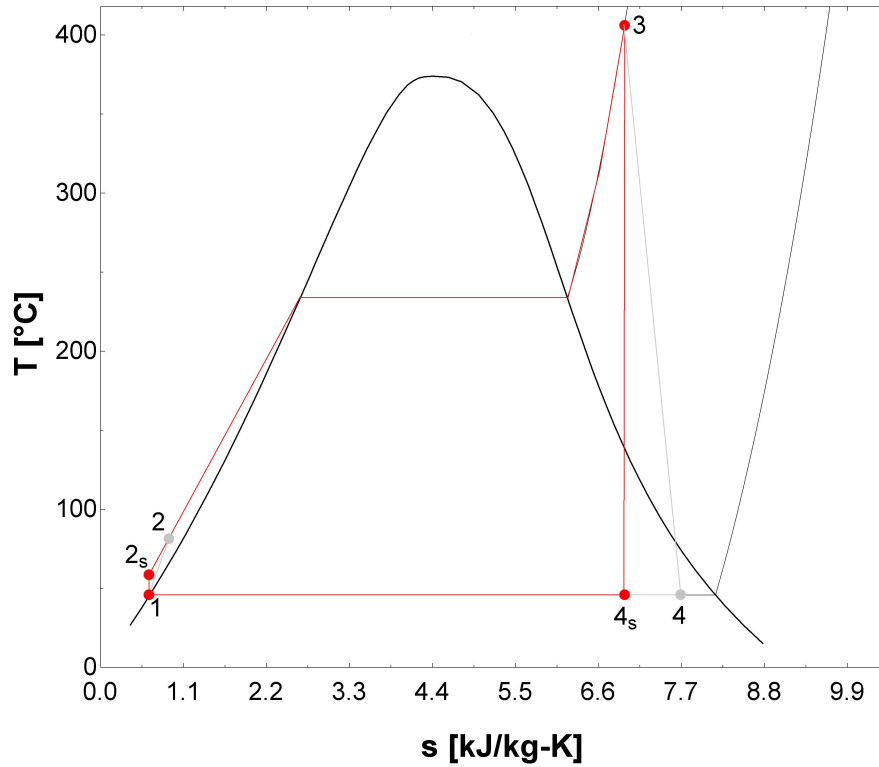


Figure 3.6: A Temperature entropy diagram of the Ideal Rankine process with water as the working fluid

lower temperature geothermal brine for power production.

For modelling a bottoming binary cycle, an organic Rankine cycle was coupled in parallel to the single flash cycle as shown on Figure 3.7. The brine from the steam separator is then led through a heat exchanger, or a boiler B, that transfers heat to the working fluid, causing it to boil. The saturated vapor of the working fluid is then led through a turbine, BT, and the superheated vapor from the turbine outlet is pre-cooled in a recuperator, R, that preheats the compressed working fluid at state 13. Then, the superheated vapour is condensed in the condenser, BC and finally through a pump to the appropriate working pressure of the power cycle.

The thermodynamical relationships needed to model the Organic Rankine bottoming cycle are discussed thoroughly in Section 3.3.1.

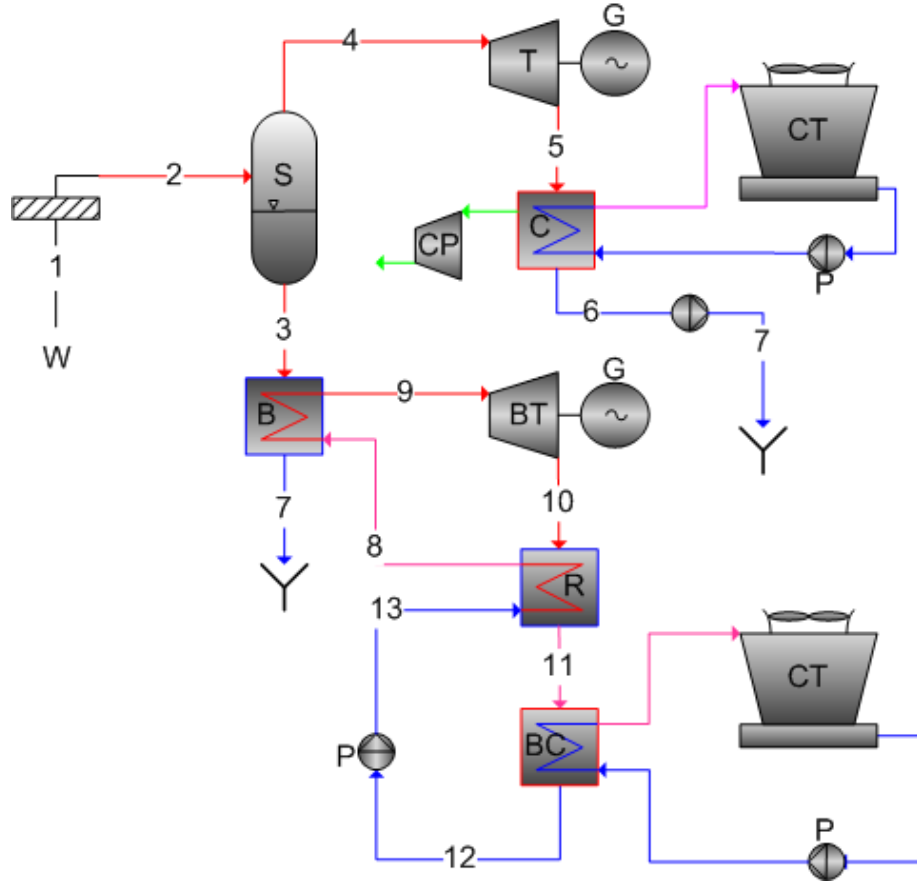


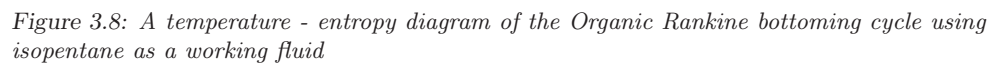
Figure 3.7: A schematic of a single flash power plant with an ORC bottoming unit

3.3.1 Thermodynamics of the Organic Rankine Bottoming Cycle

The thermodynamical modelling of the single flash unit of the hybrid flash-binary plant is the same as described in Section 3.1.1. The modelling equations for the bottoming Organic Rankine unit will be discussed in this section.

Heat transfer to binary working fluid

The heat transfer between the geothermal brine and the working fluid takes place in the boiler, B. The heat transfer has to be modelled in two stages as the working


$$\dot{m}_3(h_3 - h_{3-7}) = \dot{m}_{wf}(h_9 - h_{8-9}) \quad (3.38)$$
$$\dot{m}_3(h_{3-7} - h_7) = \dot{m}_{wf}(h_{8-9} - h_8) \quad (3.39)$$

Here, \dot{m}_3 is the mass flow of geothermal brine from the steam separator in the single flash unit and \dot{m}_{wf} is the mass flow of the working fluid. The enthalpy of the working

fluid at the end of the heating process and beginning of the boiling process is denoted as h_{8-9} and is calculated as the saturated liquid enthalpy for the operating pressure of the binary cycle. The corresponding enthalpy of the brine is denoted as h_{3-7} . The pinch in the heat exchanger is chosen to be 5°C.

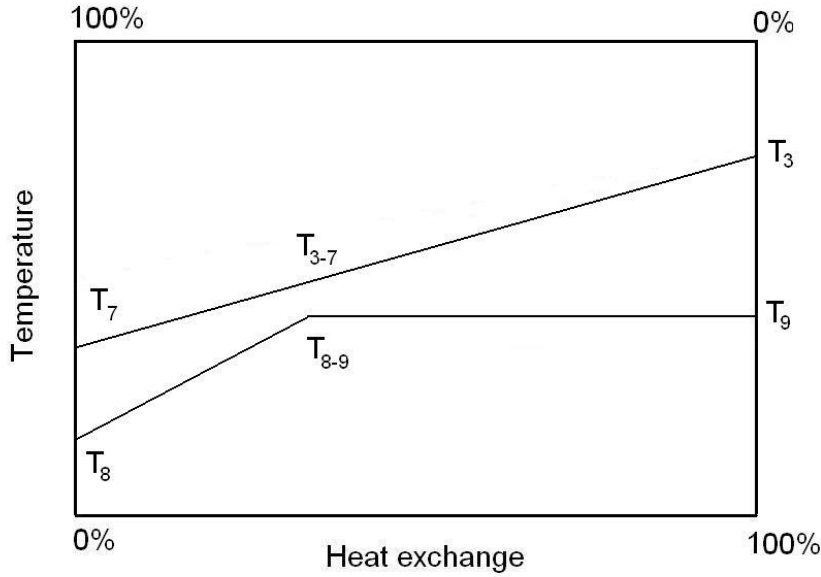


Figure 3.9: A temperature - quality diagram of the heat exchange between the geothermal brine and the binary working fluid

The binary turbine

The modelling of the binary turbine is the same as for the steam turbine. The power output of the binary turbine is calculated as

$$\dot{W}_{bt} = \dot{m}_{wf}(h_9 - h_{10}) \quad (3.40)$$

Here, the turbine is assumed to have an isentropic efficiency of about 85% as for the steam turbine, although turbines for binary cycles can be designed as a single

stage turbine with higher efficiency due to the small enthalpy drop across the turbine (Dickson and Fanelli, 2005).

The enthalpy at state 10 can be calculated the same way as was done for the single flash turbine in Equation 3.5.

Recuperator

Studies show that having an internal heat exchanger (recuperator) between the superheated stream at the turbine outlet and the compressed working fluid after condensation, can increase the efficiency of the cycle (Drescher and Bruggemann, 2007). The recuperator preheats the compressed working fluid before it enters the boiler to better utilize the geothermal brine. The balancing equation for the recuperator is given as:

$$\dot{m}_{wf}(h_{10} - h_{11}) = \dot{m}_{wf}(h_8 - h_{13}) \quad (3.41)$$

The pinch in the recuperator is usually located at the cold end of the recuperator but its location is not guaranteed and has to be monitored during the thermodynamical modelling.

Binary condenser

The balancing equations for the binary fluid condenser are similar to the ones for the single flash condenser described in Section 3.1.1 but here the condenser must cool the superheated vapour prior to condensation and thus, the process has to be divided into two steps. First, the cooling of the superheated vapour is modelled as

$$\bar{c}\Delta T_{\text{cooling}}\dot{m}_{\text{cooling}} = \dot{m}_{\text{wf}}(h_{11} - h_{11-12}) \quad (3.42)$$

Second, the condensation of the working fluid vapour can be described as

$$\bar{c}\Delta T_{\text{condensing}}\dot{m}_{\text{cooling}} = \dot{m}_{\text{wf}}(h_{11-12} - h_{12}) \quad (3.43)$$

where $\Delta T_{\text{cooling}}$ and $\Delta T_{\text{condensing}}$ are the temperature differences of the cooling water in the cooling and condensing processes respectively, \dot{m}_{cooling} is the mass flow of the cooling water and h_{11-12} is the enthalpy of the working fluid at saturated vapour state. The pinch point in the condenser is located at the saturated vapour state of the working fluid and chosen to be 5°C.

The heat exchanger area can be calculated as before by using the logarithmic mean temperature difference (LMTD) described previously in Section 3.1.1 and the overall heat transfer coefficient, U , must be estimated or calculated for the specific heat exchanger and working fluids.

Working fluid and cooling water pumps

The pumping processes for the working fluid and the cooling water are identical to the process described with Equations 3.17 and 3.18 in Section 3.1.1.

3.4 A Modification of the Double Flash Power Plant

A modification of the double flash cycle can be seen in Figure 3.10. The modification involves adding a recuperator, R , that uses a fraction of the waste brine from the high pressure steam separator, HPS, to reheat the steam from the outlet of the high pressure turbine, HPT. The rest of the brine from the high pressure separator is then flashed and used for the low pressure turbine as done in the conventional double flash cycle.

The recuperator R is a heat exchanger that transfers thermal energy from the separator brine to the steam from the high pressure turbine. The temperature of the separator brine in the recuperator, T_7 , is higher than the temperature of the steam from the turbine, T_5 , so the steam can be heated up to approximately the same temperature as the brine in the heat exchanger. The minimum temperature difference or the pinch in the recuperator can be located either at the steam inlet (cold end) or the steam outlet (warm end) and is defined as 5 °C in this study. The possible improvement compared to the conventional double flash cycle is due to superheating of the steam from the high pressure turbine which has a possibility to improve the power output of the low pressure turbine.

3.4.1 Thermodynamics of the Modified Double Flash Cycle

The idea of the modified double flash cycle is to increase the overall efficiency of the double flash cycle by superheating the steam from the high pressure turbine to enhance the power output of the low pressure turbine. The idea is similar to the idea behind an reheat Rankine cycle which will be described below.

From thermodynamics, the ideas behind increasing the efficiency of a rankine cycle are all based on increasing the average temperature at which heat is transferred to

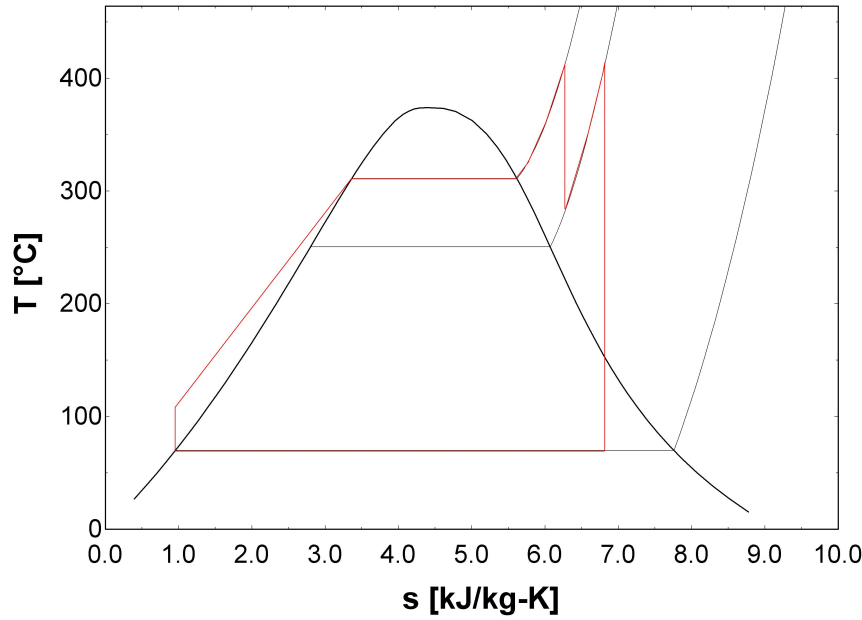


Figure 3.11: A temperature - entropy diagram for the reheat Rankine cycle

fluid itself and thus, no boiler is present to transfer thermal energy to the working fluid as in the ideal Rankine cycle. The geothermal fluid is commonly a mixture of liquid and steam and thus the steam is not naturally in the superheated vapor area in the T-s diagram, although it is known that some dry steam wells can produce a slightly superheated steam (DiPippo, 2005). The possibility to produce power from superheated steam is thus limited in geothermal power plants.

In a double flash power plant, where there is access to waste heat in the geothermal brine from the high pressure separator, the possibility for superheating arises. After the steam has been exhausted from the high pressure turbine, it is at a lower pressure and temperature than before entering the turbine as can be seen at state 5 in Figure 3.12. At the same time, the waste heat in the separator brine at point 3 is at the same temperature as the steam was before entering the turbine. This gives the possibility to transfer heat from the separator brine to the low pressure steam in a recuperator as explained before. By reheating the steam at a constant pressure up to almost the same temperature as the separator brine, the steam is superheated to state 13 and the thermodynamic gain is represented by the area under the superheated curve in Figure 3.12. The downside to this reheating process is that the brine discharge from the recuperator is at a slightly higher temperature than the discharge from the second

flash steam separator, causing the average temperature at which heat is rejected from the working fluid to rise. Then the question is if the increase in power output due to the reheating process is larger than the decrease in power production due to the increase of the average discharge temperature from the cycle.

To investigate if the reheating process gives increased power output compared to the conventional double flash cycle, the possibility to use a fraction of the brine for the reheat process and the rest for the second flashing state is introduced by optimizing the amount of brine mass flow going through the recuperator and the second flashing state.

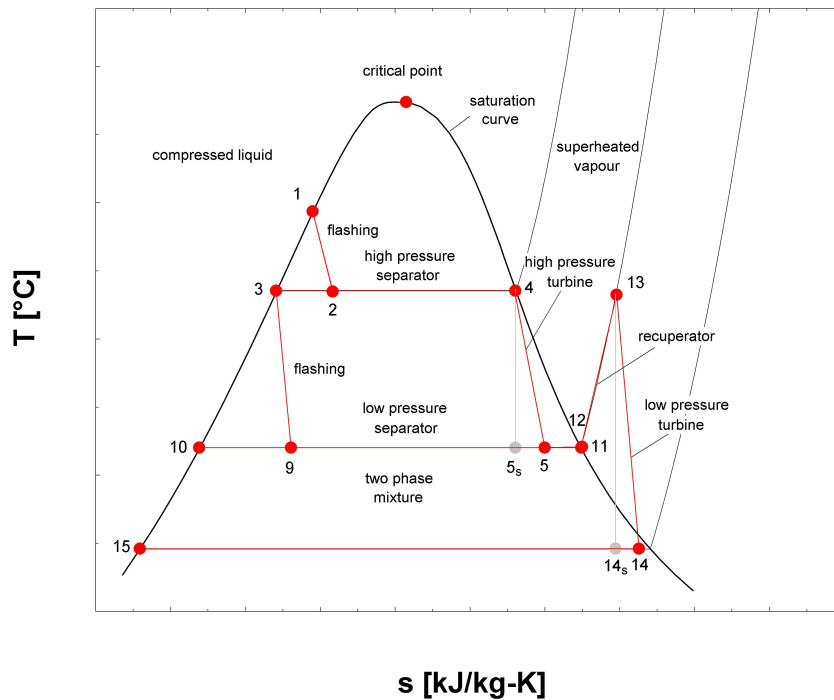


Figure 3.12: A temperature - entropy diagram for the modified double flash cycle

The setup of the modified double flash cycle is almost identical to the conventional double flash cycle. Only the possibility to extract some of the separator brine to a recuperator has been added. The thermodynamic balancing equations for the added equipment are given below.

Division of the brine between the recuperator and the second flashing

state

The division of the stream from the high pressure separator is modelled as

$$\dot{m}_3 = \dot{m}_6 + \dot{m}_7 \quad (3.44)$$

where the amount of mass flow that is led through the recuperator, \dot{m}_7 is defined as

$$\dot{m}_7 = f_{\text{reheat}} \dot{m}_3 \quad (3.45)$$

The constant f_{reheat} defines the fraction of the mass flow from the high pressure separator to be used in the reheating process.

The recuperator

The energy balance equation for the recuperator is given as

$$\dot{m}_7(h_7 - h_8) = \dot{m}_{12}(h_{13} - h_{12}) \quad (3.46)$$

The pinch point can be located on either the cold or the hot end of the heat exchanger, but due to similar temperatures and thus, similar average heat capacity of the streams, the slope of the heating and cooling processes is similar and the pinch could appear simultaneously on both ends of the recuperator.

The size calculations for the recuperator are identical to those described in Equations 3.9 to 3.11 where the heat transferred is defined as

$$\dot{Q}_{\text{reheat}} = \dot{m}_{12}(h_{13} - h_{12}) \quad (3.47)$$

and the size of the recuperator can be calculated as

$$A = \frac{\dot{Q}_{\text{reheat}}}{U \cdot LMTD} \quad (3.48)$$

Except for Equations 3.44 to 3.48 described above, the thermodynamical equations for the modified double flash cycle is identical to the conventional double flash cycle described in Section 3.2.

3.5 Economical Analysis

The capital an operating and maintenance (O&M) costs for each power plant type was estimated with conventional methods of economical analysis for geothermal power plants. The production cost of the final product, the electricity, was then evaluated for each power plant. In the following sections, these methods will be described within the scope of the study.

3.5.1 Capital and O&M Cost of the Power Plants

The capital cost is the initial investment cost of the power plant that is needed to purchase land, build all the necessary facilities and purchase and install the required equipment. The difference in capital investment of the different geothermal power plants lies mainly in the purchased equipment cost. For this study, the purchased equipment cost was estimated for all the different power plants in order to make a cost comparison of the different energy conversion systems. Also, the cost of the geothermal well was taken into account to calculate the total production cost of the electricity produced in the geothermal plant.

The total purchased equipment cost was estimated from classified actual data and levelized before it was used for the cost comparison. To estimate the cost of the machinery, the price and the size of the component have to be known or estimated to be able to give a price estimation to the same component of another size. The effect of size on equipment cost can be found by plotting all available cost data versus the equipment size on a log-log plot and it has been shown that the data correlation normally results in a straight line within a given capacity range (Bejan et al., 1996). The slope of this line, α , represent an important cost estimating parameter such that the cost of a component of any size can be estimated by knowing the cost for the same component for a given size, given by the relation

$$C_{equipment} = C_{base} \left(\frac{size_{equipment}}{size_{base}} \right)^{\alpha} \quad (3.49)$$

where $C_{equipment}$ and $size_{equipment}$ is the cost and size of the equipment to be estimated and C_{base} and $size_{base}$ is the known cost and size for the same component. The scaling exponent, α , for the main components in the goothermal power plants studied in this project is given in Table 3.1

The cost of drilling the geothermal well is estimated to be about 250 million ISK for a 2 km deep geothermal well. In order to adapt the cost of drilling to the unit mass flow

Table 3.1: Typical values for the scaling component α

Component	Sizing	Exponent α
Compressor	Power	0.95
Cooling tower	Cooling water rate	0.93
Heat Exchanger, shell and tube	Surface area	0.66
Pump, centrifugal	Power	0.37
Separator, centrifugal	Capacity	0.49
Steam turbine, condensing	Power	0.9
Source (Bejan et al., 1996)		

analysis of the power plants, the specific cost of geothermal drilling was calculated by dividing the cost for a single well with the average mass flow from actual wells connected to the Svartsengi geothermal power plant located in the southern peninsula in Iceland. The specific cost of the geothermal well was then estimated to be 115.000 US\$ per unit mass flow.

For the operating and maintenance (O&M) cost for the geothermal power plants, a rule of thumb was used to estimate the annual O&M expenses. The rule of thumb states that the O&M cost for a geothermal power plant can be estimated as roughly 2% of the total purchased equipment cost (Geirsson, 2008).

The purpose of the cost estimation in this project is to compare the difference in final production cost of the electricity and not to give a realistic view of the total cost associating the building and operating of a geothermal power plant. The capital cost of land, civil, structural and architectural work, piping, installation of equipment, instrumentation and controls and electrical equipment and materials are roughly estimated as 2/3 of the total capital cost and the purchased equipment cost accounts for 1/3 of the total capital cost.

The capital investment is the part of the capital cost that investors put into the project. The capital investment was assumed to account for 30% of the total capital cost of the different power plants, and a 70% loan was assumed to cover the rest of the required capital cost.

To calculate the annual cost of the capital investment and the annual cost of loan, the initial investment cost and the loan must be divided into equal-amount money transactions, A_{equal} , which can be calculated as

$$A_{equal} = P \cdot CRF \quad (3.50)$$

where P is the present worth of the capital cost and CRF is the capital recovery factor used to determine the equal amounts A of a series of n money transactions that have the present value equal to P . The CRF is defined as

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (3.51)$$

Here, i is either the required rate of return for the investment capital, chosen to be 15% in this study, or the loan interest rate, chosen as 6% for the loan for the capital cost (Valdimarsson, 2008). Inflation and escalation also affect the value of the annual equal amount over the project lifetime. The effect of inflation and escalation were neglected in this study for reasons of simplification as they do not greatly affect the cost comparison between the power cycles.

Similar calculations would have to be done for the O&M costs for the entire power plant. Due to cost escalation, the estimated O&M cost increases from year to year causing nonuniform annual payments. By neglecting the effects of cost escalation, the O&M costs can be assumed to be fixed over the lifetime of the project as was done in this study.

The annual revenue needed to cover the investors requirements of return, costs associated with operation and maintenance of the power plant and costs associated with the downpayments of loan can then be calculated as:

$$A_{revenue} = A_{return} + A_{loan} + A_{O\&M} \quad (3.52)$$

To evaluate the required cost of the product, the annual required revenue, $A_{revenue}$, is divided by the net annual power production to get the cost per kWh for the electricity. The amount of kWh produced where calculated as

$$W_{kWh} = \dot{W}_{net} \cdot t_{op} \quad (3.53)$$

where t_{op} is the total annual operating time of the power plant in seconds. In this project, the operating time was estimated as $3600 \cdot 8040$ seconds by assuming 30 days of downtime due to preventive maintenance of the plant.

4 Optimization of Net Power Output

The optimization of the maximum net power output for each cycle was done by interaction between the thermodynamical calculations in the software Engineering Equation Solver (EES) and an optimization routine in Matlab by using the function `fmincon`. This was done due to problems of restricting the optimization problem in the optimization routine provided in EES. The Matlab function `fmincon` uses constrained nonlinear optimization and is an effective tool for this kind of optimization.

The interaction between EES and Matlab was in the form of a dynamic data exchange (DDE). Dynamic Data Exchange is a technology for communication between multiple applications under Microsoft Windows software operating system and its primary function is to allow Windows applications to share data (Wikipedia, 2008b). The DDE interaction between Matlab and EES allows the possibility to use EES as a database for thermodynamical values and solve the balancing equations used to model each power plant and send the solutions to Matlab where the optimizations routine can process the data and find the optimized values for each cycle.

In the following sections, the optimization problems and limitations for each power cycle are described.

4.1 Optimization Variables and Constraints

4.1.1 Single Flash Cycle

Optimization of the power output of the single flash cycle is based on choosing the optimum wellhead (or separator) pressure that gives the maximum power output for the cycle. The optimization routine is relatively simple since there is only one optimization variable and it can be determined from varying the value of the wellhead pressure to locate the power output maximum.

The steam quality at the output of the turbine is a constraint in the optimization problem, as the quality may not go below $x = 0.85$ and that limits the maximum wellhead pressure allowed as can be seen on a temperature-entropy diagram for the single flash process, e.g. in Figure 3.2.

4.1.2 Double Flash Cycle

For optimizing the power output of the double flash cycle, two optimum pressure states, p_2 and p_6 , need to be found. Thus, an extra degree of freedom has been added to then optimization routine compared to optimization of the single flash cycle that makes the procedure more complicated. For each value of the operating pressure of the high pressure separator HPS, an optimum pressure value can be found for the low pressure separator LPS. The problem is then to find the two corresponding pressures at which the double flash cycle gives the highest net electrical power output \dot{W}_{net} .

The constraint in the optimization process is, as for the single flash cycle, the steam quality at the outlet of each turbine. This constraint affects the allowable pressure at the high pressure state of each turbine. Thus, there are two steam quality constraints for the double flash cycle; constraints for the steam quality at state 5, x_5 and at state 10, x_{10} .

4.1.3 Hybrid Single Flash Cycle with Organic Rankine Bottoming Unit

There are two different optimization variables for the hybrid single flash plant with the organic Rankine bottoming cycle. First, the wellhead pressure must be optimized as for the other cycles and second, the pressure in the organic Rankine bottoming cycle needs to be optimized.

For this cycle, only the steam quality at the outlet of the steam turbine becomes a constraint due to the fact that the isopentane vapour at the outlet of the binary turbine is superheated and thus, no moisture is present.

4.1.4 Modified Double Flash Cycle

Four optimization variables were used to optimize the net power output of the modified double flash cycle. First, the separator pressures, p_2 and p_9 have to be optimized as well as the constant describing the fraction of mass flow taken from the separator brine and used for reheating, f_{reheat} . The temperature of the waste brine at the cold end of the recuperator, T_8 , was also optimized to monitor the location of the pinch point in the recuperator.

4.1.5 Overview of the Optimization Problems

An overview of the optimization variables and constraints for each power cycle can be seen in Table 4.1.

Table 4.1: Optimization variables and constraints for each power cycle

Power cycle	Optimization variables	Constraints
Single flash	p_2	$x_5 \geq 0.85$
Double flash	p_2, p_6	$x_5 \geq 0.85, x_{10} \geq 0.85$
Hybrid single flash and ORC	p_2, p_{13}	$x_5 \geq 0.85$
Modified double flash	$p_2, p_9, f_{reheat}, T_8$	$x_5 \geq 0.85, x_{14} \geq 0.85$

4.2 Wellhead Pressure Limitations

The wellhead pressure directly affects the mass flow from the well. A typical geothermal well productivity curve for a water fed well and a well producing two phase flow is shown in Figure 4.1. As the pressure increases, the mass flow produced by the well will eventually decrease to a point that the well is completely closed and no mass flow is produced. That is why production curves for the wells connected to the geothermal power plant must be available before optimization of the wellhead pressure is determined. At a specific pressure range, the mass flow from the well with the two phase flow is relatively stable with a low decrease in the production. This pressure range is different from well to well and depends on the geothermal reservoir. In Iceland, some

high-enthalpy wells at the high temperature field at Námafjall in northern Iceland can hold a steady production up until 40-50 bars before the mass flow begins to decrease (Steingrímsson, 2007).

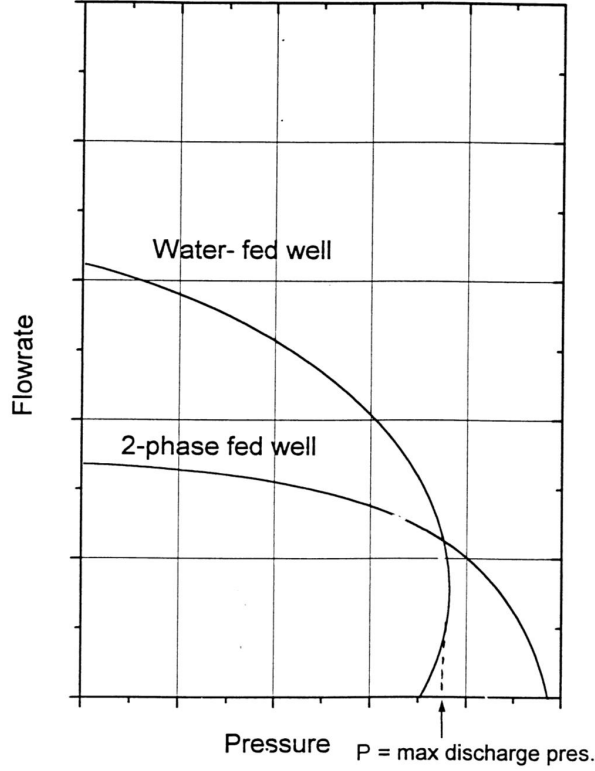


Figure 4.1: Productivity curves for geothermal wells producing two phase flow and liquid flow(Iceland GeoSurvey, 2008)

In this project, the optimization of the wellhead pressure is not dependant on production curves and the power output is calculated as specific power production (Power produced per unit mass flow) in $kW/(\dot{m})$ where \dot{m} is in kg/s . These calculations will determine the specific power output for the cycle whilst varying the energy output of the production wells. The wells are modelled as a single well with a given enthalpy ranging from $h_0 = 1000kJ/kg$ to $h_0 = 2500kJ/kg$. The upper range of the well enthalpy is relatively high but such high enthalpies can occur in high-enthalpic high-temperature areas with steam dominated wells and even in deep drilling projects.

5 Results

5.1 Assumptions

For the optimization of the different cycles, some assumptions had to be made regarding the behavior of the geothermal reservoir and the limitations and restrictions in the power plant equipment. These assumptions are listed in the sections below.

5.1.1 Geothermal Reservoir

The following general assumptions regarding the behaviour of the geothermal system were made during the thermodynamical modelling of the different power cycles:

- The maximum wellhead pressure of the production wells was restricted to 35 bar.
- The amount of non-condensable gases in the geothermal fluid was assumed to be 1% of the total mass flow from the wells.
- The possibility of scaling due to high concentrations of minerals in the geothermal fluid was neglected.
- Enthalpy of the production wells varied from 1000 - 2500 kJ/kg.
- Production wells were modelled to produce 1 kg/s to estimate the specific power output of the different cycles.

Dead-state assumptions for exergy calculations

The conditions at dead state where exergy of the working fluid is said to be zero were taken as $T_0=5\text{ }^\circ\text{C}$ and $p_0=1\text{ bar}$.

5.1.2 Power Plant Equipment

The power plants consist of various components of complicated design and restrictions to their working conditions. The following restrictions were estimated for the different power plant components:

Isentropic efficiency of turbines, compressors and pumps

The turbines, pumps and compressors do not operate at 100% efficiency. The efficiencies for these components are given in Table 5.1 (Dickson and Fanelli, 2005).

Table 5.1: Isentropic efficiency of different power plant equipment

Equipment	Isentropic efficiency
Turbines	$\eta_t = 85\%$
Compressor	$\eta_{comp} = 85\%$
Pumps	$\eta_p = 50\%$

Overall heat transfer coefficient in heat exchangers

The overall heat transfer coefficients used to calculate the total heat exchanger area for each heat exchanger in the modelled power plants are given in Table 5.2 (Valdimarsson, 2008).

Table 5.2: Overall heat transfer coefficient \bar{U} for various heat exchangers

Fluids	$\bar{U} [W/m^2 \cdot K]$
Water-Water	2000
Steam-Water	2000
Water-Isopentane	1200
Isopentane-Isopentane	1200

Pinch assumptions in heat exchangers

Pinch assumptions restrict the maximum allowable effectiveness of the heat exchangers and ensure that the heat exchanger surface does not become excessively large.

The minimum pinch assumptions for the heat exchangers in the power cycles are given in Table 5.3

Table 5.3: Minimum pinch in various heat exchangers

Heat exchanger	Minimum pinch
Recuperator, water	5°C
Recuperator, isopentane	8°C
Boiler and preheater, water-isopentane	5°C
Condenser, water	5°C
Condenser, isopentane	5°C

Condenser pressures

The condenser pressures in the different cycles were chosen as:

- 0.1 bar for the steam condenser in the flashing units.
- 1 bar for the isopentane vapour condenser in the ORC bottoming unit.

Cooling water circuit

The following assumptions were made when modelling the cooling water circuit:

- The cooling water enters the condensers at 5°C.
- The cooling tower fan was estimated to consume 0.25 kW per unit mass flow of cooling water according to power requirements in the cooling tower fans in Hellisheiði geothermal power plant.
- Evaporation of the cooling water was neglected.

5.2 Optimization of Net Power Output

The objective function of the optimization problem for each power cycle was the specific net power output of each cycle. The specific power output is calculated for unit mass flow (1 kg/s) from the production wells and the net output was calculated by withdrawing all power requirements of the modelled equipment such as compressors, pumps and fans. In the following sections, the results for each cycle are presented and

discussed and finally, the cycles are compared with respect to their optimum power output and wellhead pressures, exergetic efficiency and the product cost of electricity generated in each cycle.

5.2.1 Single Flash Cycle

The optimized specific net power output for the single flash cycle can be seen in Figure 5.1. The specific power output ranges from 82.9 kJ/kg to 459.1 kJ/kg as the enthalpy of the production wells increase from 1000 kJ/kg to 2500 kJ/kg.

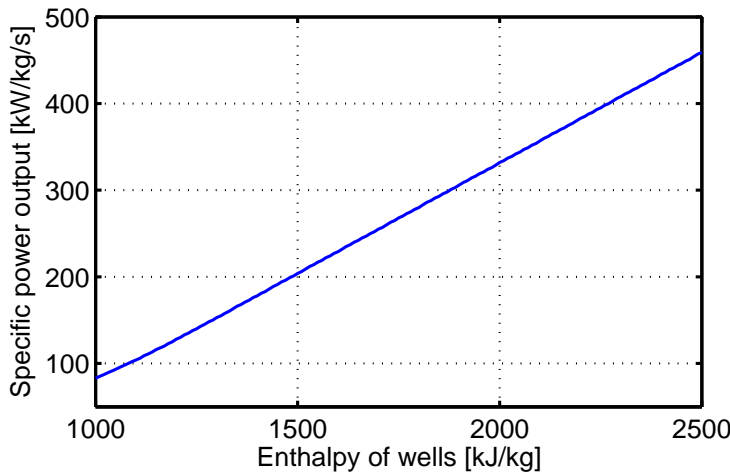


Figure 5.1: Optimized net power output from the single flash cycle

The optimum wellhead pressure is shown in figure 5.2. The optimum pressure quickly becomes restricted to 6.6 bars as the enthalpy increases due to the restriction of the steam quality at the outlet of the steam turbine as seen in figure 5.3. At approximately 1260 kJ/kg, the steam quality drops to 85% as can be seen on figure 5.3 and the pressure becomes a constant value for all the enthalpies above.

5.2.2 Double Flash Cycle

The double flash cycle has increased power output compared to the single flash cycle due to better utilization of the geothermal brine. The optimized specific net power output of the double flash cycle can be seen in Figure 5.4 and the corresponding pressure optimization is given in Figure 5.5. Here, the upper limit of the pressure was

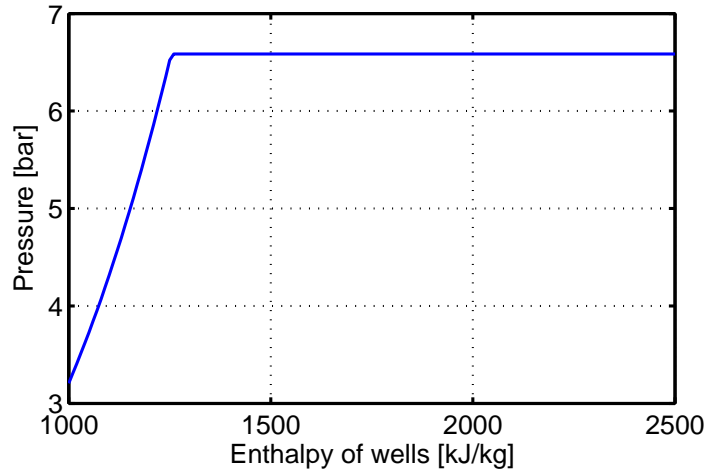


Figure 5.2: Optimized wellhead pressure for the single flash cycle

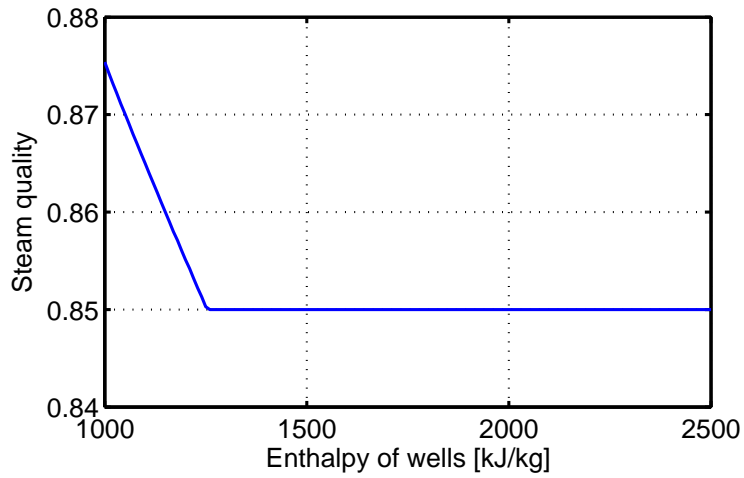


Figure 5.3: Steam quality at the turbine outlet for the single flash cycle

set to 35 bar due to assumed reservoir pressure restrictions. As the wellhead pressure becomes restricted, the pressure in the lower pressure step begins to decrease slowly to obtain the highest power output of the cycle.

The steam qualities at the turbine outlets are shown in Figure 5.6. It can be seen that the constraint for the steam quality is never active in the optimization process

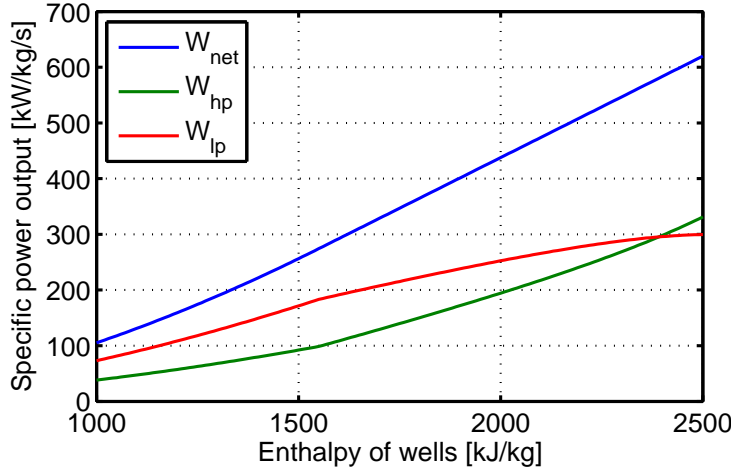


Figure 5.4: Optimized net power output from the double flash cycle

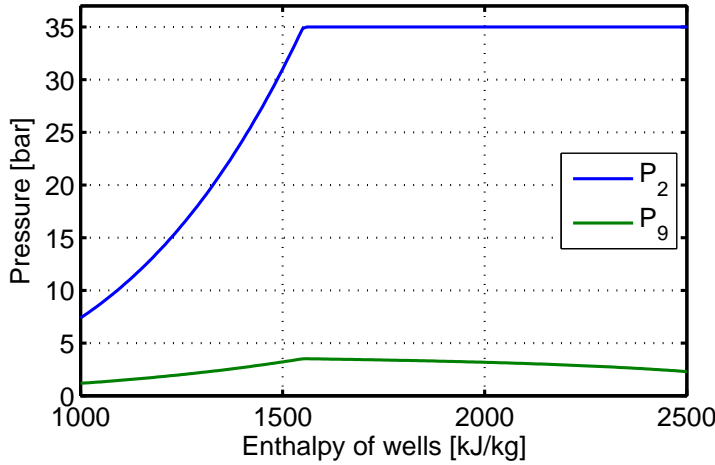


Figure 5.5: Optimized wellhead pressure and second flashing pressure for the double flash cycle

due to the pressure restrictions of the geothermal reservoir. As the wellhead pressure becomes restricted and the working pressure of the low pressure turbines begins to decrease, the steam quality at the high pressure turbine, x_5 , starts to decrease due to a larger pressure drop in the high pressure turbine, placing the point on the T-s diagram denoting the state at the turbine outlet (point 5 on the T-s diagram in Figure

3.4) further down under the saturation curve and further away from the saturated vapour line. At the same time, the pressure drop in the low temperature turbine decreases as the condensing pressure is kept constant, resulting in increasingly higher steam quality at the low pressure turbine outlet, x_{10} .

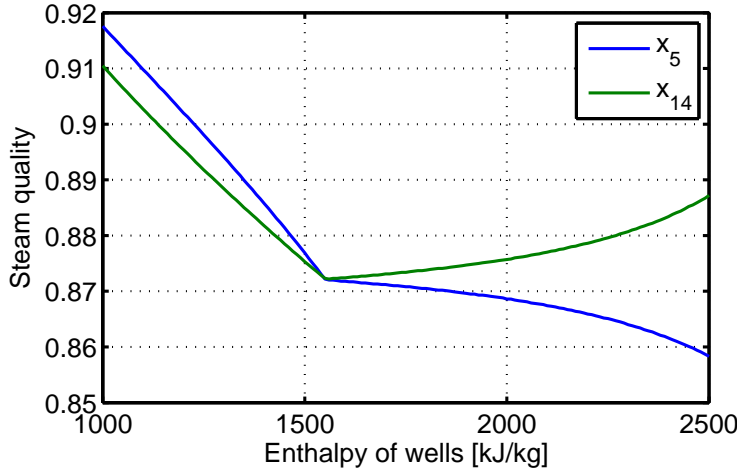


Figure 5.6: Steam quality at the turbine outlets for the two turbines in the double flash cycle

5.2.3 Single Flash Cycle with Organic Rankine Bottoming Unit

The results for the optimized combined specific net power output and the individual turbine power output of the hybrid single flash and organic Rankine cycle is shown in Figure 5.7 and the optimum working pressures in the power plant can be seen in Figure 5.8.

The net power output of the ORC bottoming unit decreases as the enthalpy increases due to the decrease in mass flow of brine from the steam separator and also because the enthalpy in the geothermal brine becomes constant when the wellhead pressure becomes constant as seen in Figure 5.8. When the mass flow of brine decreases and the enthalpy of the fluid remains constant, the heat transferred to the ORC binary unit decreases resulting in lower power output. So the highest power output from the binary unit is obtained when the production wells produce large amount of geothermal brine due to low steam quality at low enthalpies.

The power output of the single flash unit in the hybrid cycle is similar to the power

output from the single flash power plant without the bottoming unit. Thus, the bottoming unit has very little effect on the power output of the single flash cycle and only adds to the total power production of the power plant by utilizing the waste heat in the geothermal brine.

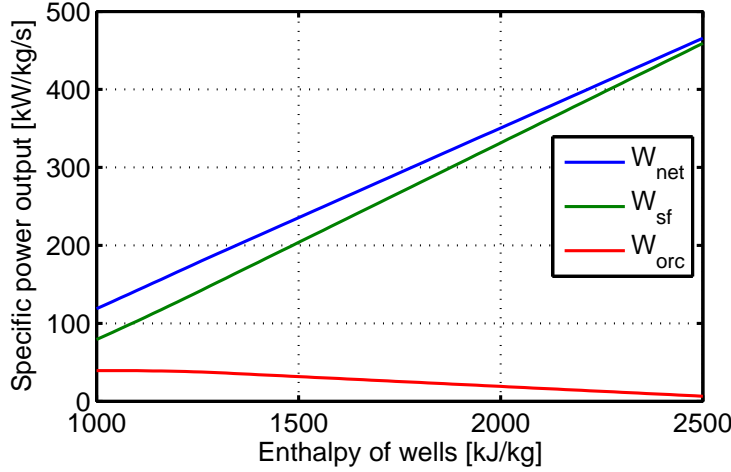


Figure 5.7: Optimized net power output from the hybrid single flash - ORC plant and the individual power output from each unit

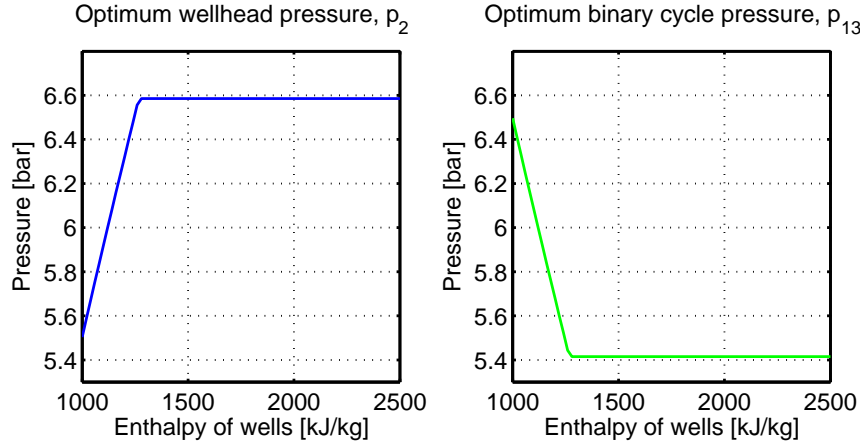


Figure 5.8: Optimized wellhead pressure of the single flash unit and the optimum pressure of the isopentane in the ORC bottoming unit

The steam quality at the steam turbine outlet is shown in Figure 5.9. The quality becomes a restriction to the process at an enthalpy input of 1280 kJ/kg, similar as

for the conventional single flash cycle. This restricts the maximum working pressure allowed in the cycle to 6.6 bar.

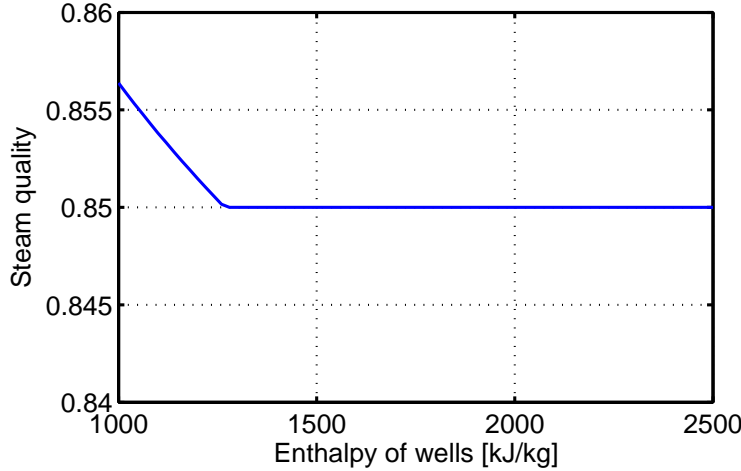


Figure 5.9: Steam quality at the steam turbine outlet in the single flash unit in the hybrid single flash-ORC power plant

5.2.4 Modified Double Flash Cycle

The specific power output of the modified double flash cycle with the added recuperator is shown in Figure 5.10 along with the power outputs from each turbine. The corresponding pressure optimization is given in figure 5.11. The upper limit of the wellhead pressure, p_2 , was set to 35 bars as previously explained for the double flash cycle. The steam quality at the turbine outlets is shown in figure 5.12.

The working pressure of the high pressure turbine increases rapidly as the enthalpy of the production wells increases, until it reaches the estimated pressure restriction of the geothermal reservoir of 35 bars. The working pressure of the low pressure turbine, denoted as p_9 , increases at first, but as the pressure in the high pressure step becomes constant, the optimum working pressure of the low pressure turbine starts to decrease until the steam quality after the low pressure turbine, x_{14} , becomes a constraint to the pressure decrease as it reaches 85% steam quality. After the enthalpy of the wells reaches about 1880 kJ/kg, both of the working pressures remain constant throughout the enthalpy change.

The modification made on the double flash cycle is based on diverting a part of or all of the geothermal brine from the steam separator to a recuperator that transfers

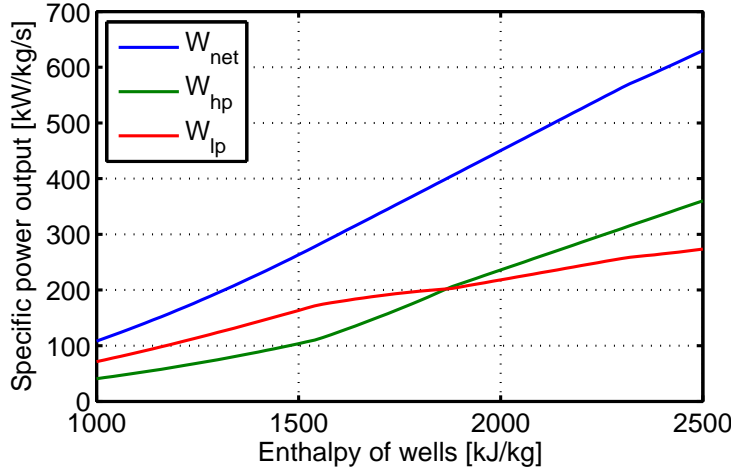


Figure 5.10: Optimized net power output from the modified double flash cycle and the individual power output from the high pressure turbine, W_{hp} , and the low pressure turbine, W_{lp}

heat to the steam after it has undergone a pressure drop in the high pressure turbine. The mass fraction, f_{reheat} , was optimized for each case of enthalpy input from the production wells and the result can be seen in Figure 5.13. For low enthalpies, the mass fraction is low and starts at about 12% for the lowest enthalpy production wells that give a large amount of water for the low pressure flashing. As the enthalpy of the production wells increases, the mass flow to the recuperator increases exponentially until all of the separator water is used for the reheating process.

The temperatures in and out of the recuperator can be seen on Figure 5.14. As a result of the changes in working pressures throughout the simulation, the temperatures of the geothermal brine from the high pressure steam separator, T_7 , and the steam from the high pressure turbine, T_{12} , change accordingly. At first, when both working pressures (p_2 and p_9) are increasing as seen on Figure 5.11, the temperatures of the brine and the steam also increase. When the wellhead pressure becomes restricted, the temperature of the geothermal brine from the high pressure steam separator becomes a constant at about 242.6 °C, but due to the corresponding decrease in the lower pressure, p_9 , the temperature of the steam also decreases. As a result, the geothermal brine is cooled further down in order to be able to fully reheat the steam into the superheated region. When the lower pressure becomes a constant due to the steam quality restriction on x_{14} , the temperature of the steam from the high pressure turbine becomes a constant of about 115.6 °C.

The temperature of the superheated steam after the reheating process in the recuper-

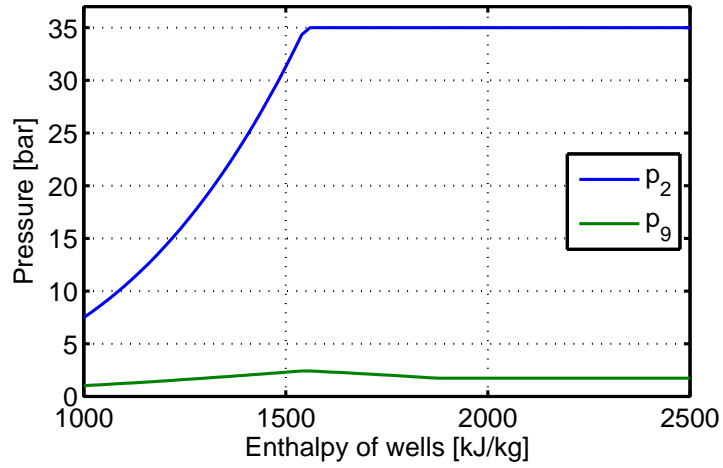


Figure 5.11: Optimized wellhead pressure and second flashing pressure for the modified double flash cycle

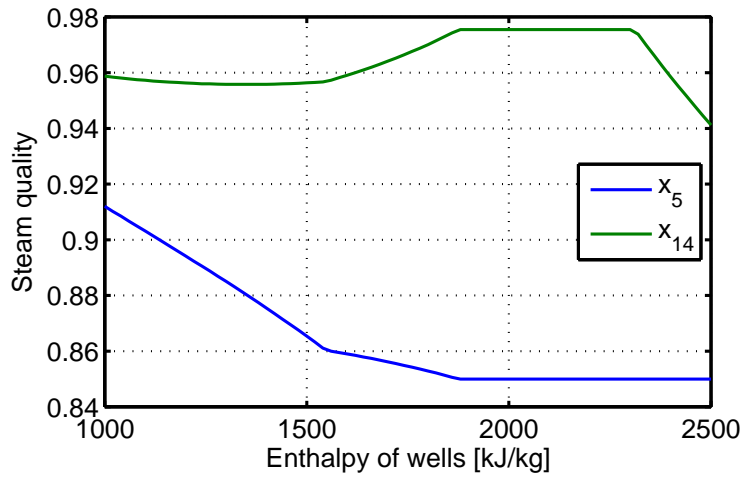


Figure 5.12: Steam quality at the turbine outlets for the two turbines in the modified double flash cycle

ator, T_{13} , suddenly begins to decrease after being held constant at enthalpies ranging from 1880 kJ/kg to 2340 kJ/kg. At such high enthalpies and at the corresponding wellhead pressure of 35 bars, the steam quality from the production wells is high and at 2340 kJ/kg the steam quality at state 2 is about 74%. This means that the

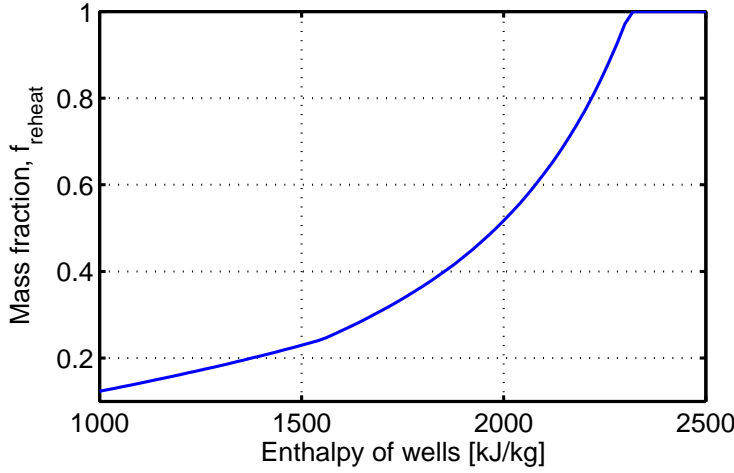


Figure 5.13: The mass fraction of geothermal brine that is led from the separator to the recuperator in the modified double flash cycle

mass flow of liquid travelling with the steam and thus, the mass flow of brine from the high pressure separator is relatively low. Due to the low mass flow of the brine from the separator, the heat needed to fully reheat the steam is not sufficient even though all of the brine is led through the recuperator. This results in the decreasing temperature of the superheated steam, but the recuperator still manages to transfer enough heat so that the steam becomes slightly superheated before entering the low pressure turbine.

5.3 Comparison of the Power Cycles

5.3.1 Net Power Output and Efficiencies of the Cycles

The optimized net specific power production from each of the powerplants discussed above is shown in Figure 5.15. The single flash cycle, where the geothermal brine is disposed of after the steam separator, has the lowest power production of all the cycles. For low enthalpy areas which produce a two phase flow with relatively low temperature range, the hybrid flash-binary power plant using an organic Rankine cycle as the bottoming unit in parallel to the single flash cycle gives the best result for the maximum power production. The hybrid single flash - ORC plant is superior to the other cycles for enthalpies lower than 1300 kJ/kg, where the fluid from the

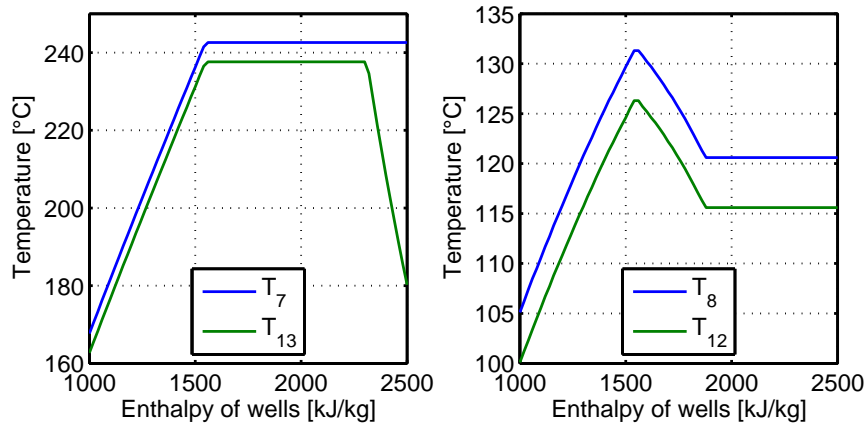


Figure 5.14: The temperature differences in both ends of the recuperator. T_7 and T_{13} are the temperatures at the inlet of the geothermal brine and outlet of the superheated steam, whereas T_8 and T_{12} are the temperatures of the return brine and the saturated steam respectively

production well consists of over 70% liquid and 30% steam at temperatures under 163 °C for the optimum wellhead pressure for the single flash cycle. For the optimum wellhead pressure of the two double flash cycles, the mass flow of steam from the production wells is only up to 20% of the total mass flow and has temperatures under 209 °C. The power output of the hybrid single flash - ORC will continue to produce more power than the single flash cycle throughout the enthalpy range but the difference between the two cycles decreases steadily as the enthalpy increases, causing them to produce almost the same amount of power for enthalpies around 2500 kJ/kg, when the steam fraction is over 80% of the produced mass flow from the wells for the given optimized wellhead pressure of 6.6 bar and there is only a small amount of brine available for the ORC unit.

At enthalpies higher than 1300 kJ/kg, the two double flash cycles start to exceed both the single flash cycle and the hybrid cycle and the difference steadily increases with the increased enthalpy from the production wells. At higher enthalpies, the double flash cycles have a clear superiority and can produce over 35% more electricity than the other cycles. The reason for the increased power output of the double flash cycles is that the single flash cycle and the hybrid cycle immediately become restricted because the steam quality at the outlet of the turbine quickly falls to 85% which puts a restriction on the maximum allowable wellhead pressure. The optimum wellhead pressures for the different cycles can be seen on Figure 5.16.

The modified double flash cycle has a small advantage over the conventional double flash cycle with respect to the net power produced. The modified double flash cycle

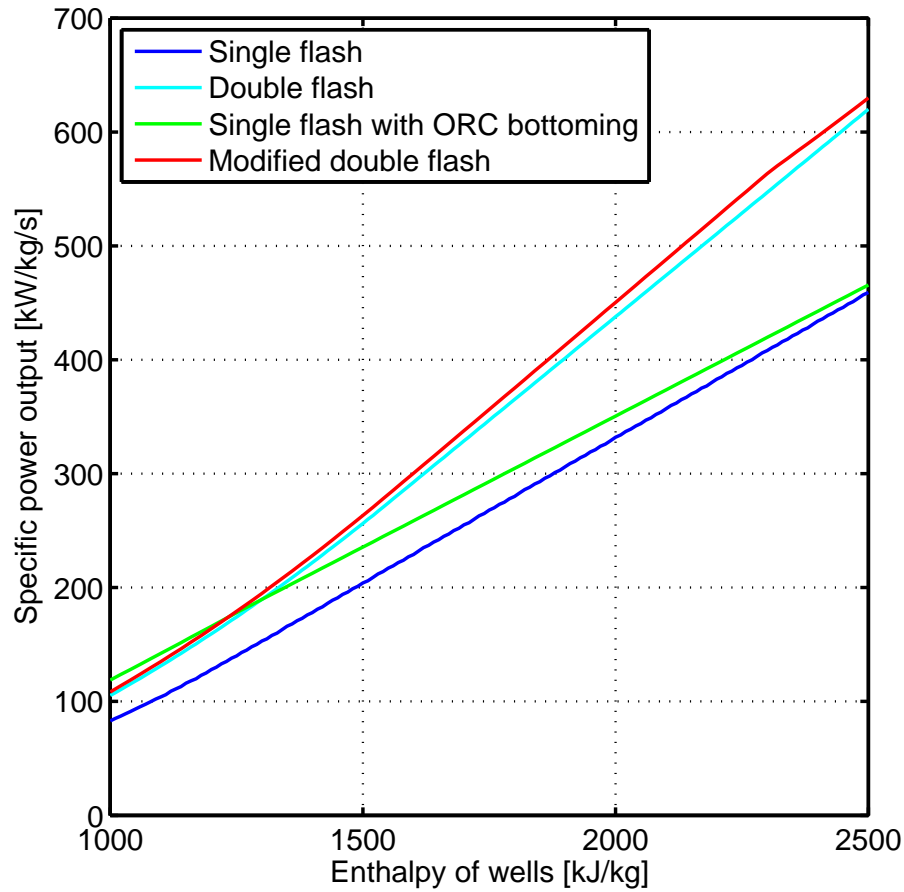


Figure 5.15: A comparison of specific net power output for the different cycles that were modelled

produces from 1% to 2.5% more power than the conventional double flash cycle.

The exergetic efficiency for the different cycles are compared in Figure 5.17. The difference in the exergetic efficiency correlates at some extent to the difference in the net specific power output for each cycle. The exergetic efficiency of the single flash cycle is the lowest one for all the different cycles, varying from 35% to 57.7%. The exergetic efficiency for the combined single flash and ORC bottoming unit varies from 47.6% to 58.5% and is superior to all the cycles for enthalpies ranging from 1000-

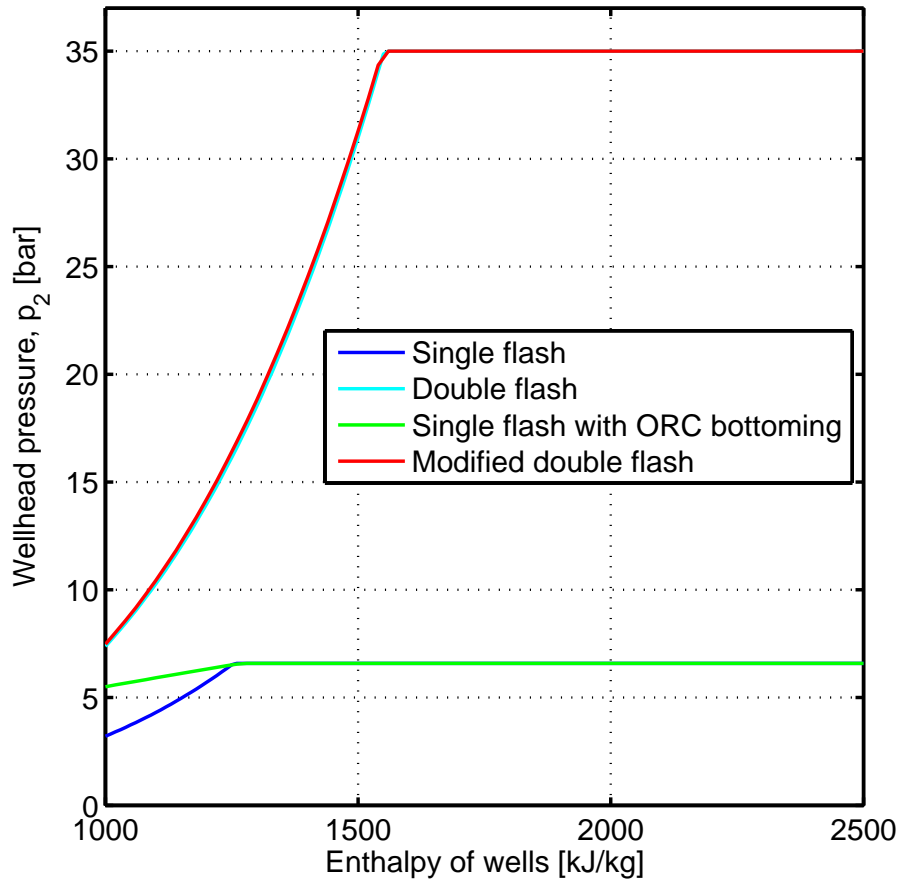


Figure 5.16: A comparison of the optimum wellhead pressure for the the different power cycles

1600 kJ/kg. The exergetic efficiencies for the two double flash cycles are similar for all enthalpies, ranging from 41.3% to 64.5% for the conventional double flash cycle and 42.5% to 65.6% for the modified double flash cycle. The modified double flash cycle is superior to all the other cycles at higher enthalpies than 1600 kJ/kg.

The discontinuous behaviour of the efficiency curves are related to previously discussed restrictions and constraints in the optimization, e.g. wellhead pressure restrictions and steam quality constraints.

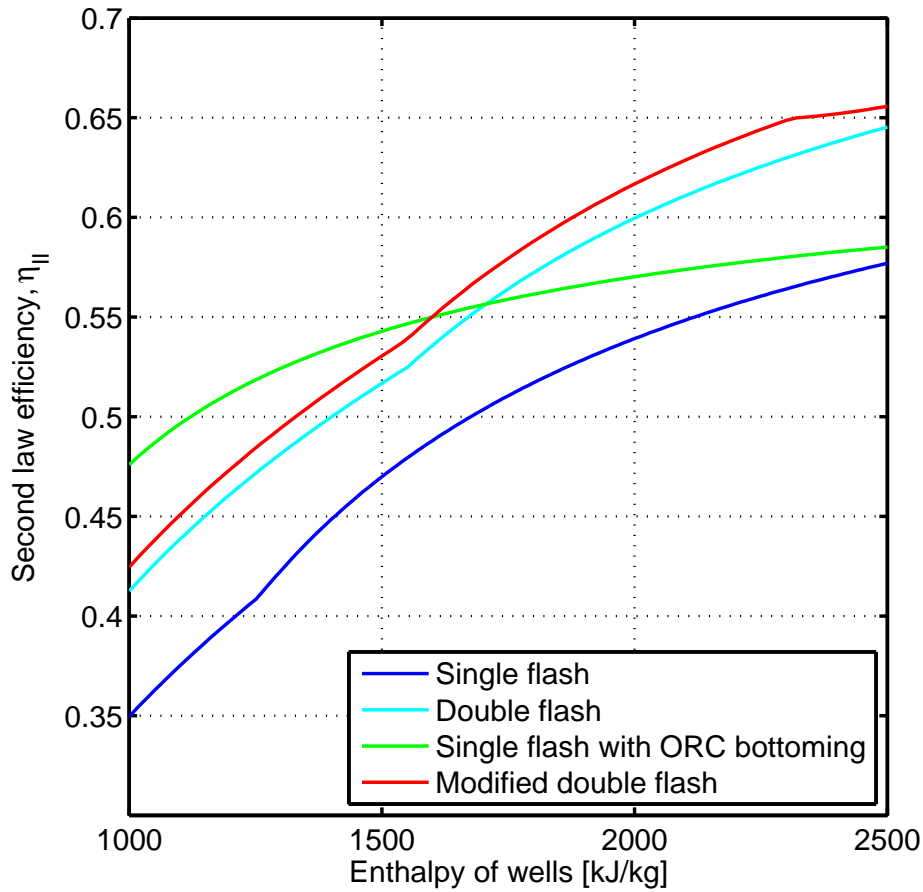


Figure 5.17: A comparison of the thermal efficiency for each cycle

5.3.2 Economical Comparison Between Cycles

A case of 100 MW_e power production

An estimate of the total production cost for the electricity generated from each cycle was calculated with methods of economics explained in Section 3.5. A case of 100 MW_e production was simulated for each cycle to be able to compare the production cost of electricity based on the cost estimation for the purchased equipment cost

and the cost of constructions, drilling of wells and the operation and maintenance cost. The equipment cost was estimated based on actual data and levelized due to confidentiality. The base for levelization was the cost for producing 100 MW_e in the single flash cycle with an enthalpy input of 1000 kJ/kg . The results of the estimated levelized production cost for each cycle can be seen in Figure 5.18 where the total production cost is plotted against the enthalpy of the geothermal fluid coming from the production wells.

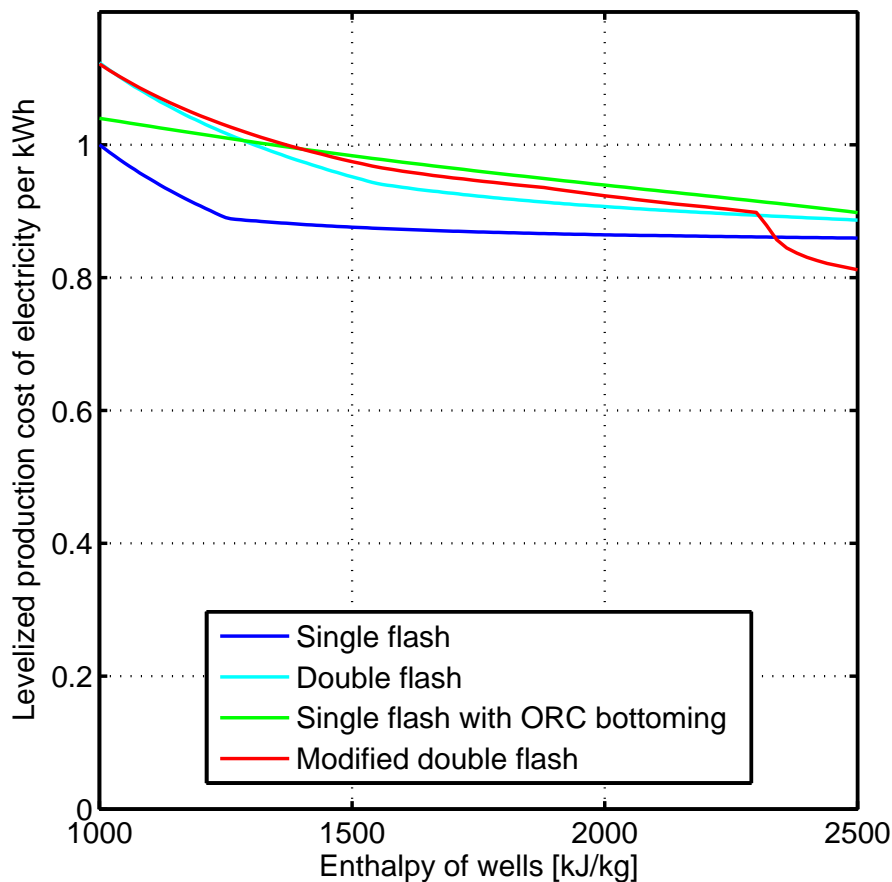


Figure 5.18: A comparison of the product cost for each cycle

The single flash power plant carries the least production cost for a net power output of 100 MW_e for almost the whole enthalpy range. This is due to the fact that the

single flash cycle has a simpler setup with fewer components compared to the other cycles and thus, carries the least purchased equipment cost.

The hybrid single flash and ORC unit has a lower production cost for low enthalpies than the two double flash cycles. At approximately 1300 kJ/kg, the production from the hybrid flash-binary plant first becomes more expensive than for the conventional double flash cycle. Thus, at lower enthalpy areas, it would be more feasible to construct a hybrid flash-binary plant than a double flash cycle, but at higher enthalpies, the double flash cycles become more economically viable.

At high enthalpy input to the power plant, the modified double flash cycle gives the minimum production cost for the 100 MW_e production. This is due to the fact that the total mass flow from the steam separator is led through the recuperator instead of the second flashing state so there is no longer need for the low pressure steam separator as for the conventional double flash cycle. This leads to a considerable decrease in the purchased equipment cost for the reheating cycle and the production cost decreases as a result of that. At lower enthalpies than approximately 2300 kJ/kg, the conventional double flash cycle is less expensive than the modified cycle due to less cost for required equipment.

Another interesting result from simulation of the 100 MW_e case is the difference in mass flow required from the production wells. The result for the required mass flow is shown in Figure 5.19. The single flash cycle always requires the greatest mass flow from the production wells which results in a need for drilling more production wells than for the other cycles, also adding to the capital cost of the power plant. The hybrid single flash and ORC power plant requires the least mass flow for lower enthalpies and the two double flash cycles require the least mass flow from the production wells for the higher enthalpies. The type of power plant chosen to be constructed at a certain geothermal area can thus be restricted to the available mass flow from the production wells. If the area does not support the mass flow required to produce these 100 MW_e from the cheapest available technology, the single flash cycle, which requires more mass flow than the more expensive cycles, then the choice will have to be to increase the production cost by using more complicated technologies with increased efficiency.

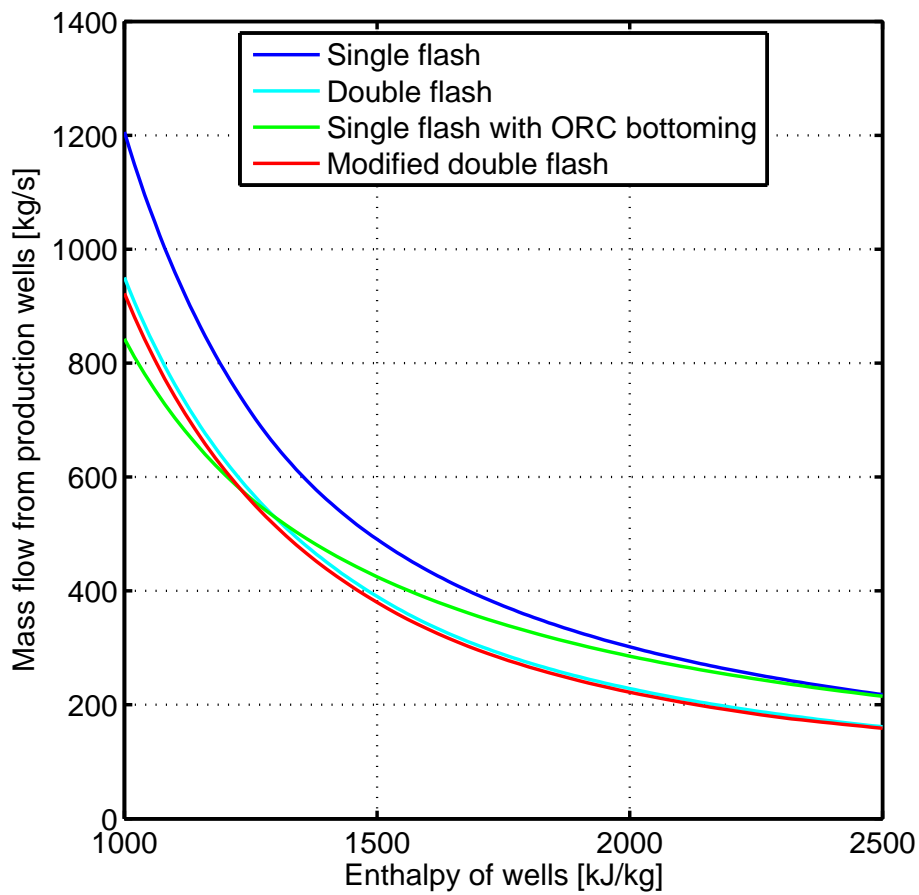


Figure 5.19: A comparison of the required mass flow from production wells to produce 100 MW_e for each cycle

6 Conclusion

The focus of this study was to investigate the possibilities of optimizing the utilization of geothermal fluid from geothermal production wells by making use of the brine that travels with the steam to the surface as it enters the geothermal power plants. The brine is separated from the high pressure steam before it enters the power plant and is often discharged back into the reservoir or to the open natural surroundings without utilizing it further. Such power plants are called single flash power plants.

Three different power cycles that utilize the geothermal brine for increased power production were modelled in order to compare the different utilization possibilities and these cycles were compared to the conventional single flash cycle. One of the cycles that were modelled in this study is a modification of a conventional double flash cycle where the possibility of superheating the geothermal steam in a recuperator after the expansion in the high pressure turbine is introduced. The results for the modified double flash cycle are promising and gave the best results for the amount of specific power output of all the different cycles. The superiority of the modified double flash cycle over the conventional double flash cycle was not measured in large numbers, as it produces about 1-2.5% more power than the conventional double flash cycle. The scaling potential of the geothermal fluid could potentially restrict the use of the recuperator as too much scaling in the recuperator could be a result of the reheating process.

By adding a bottoming binary unit parallel to the single flash cycle, the geothermal brine is utilized further and gave the best results for low enthalpy wells where the mass fraction of brine is relatively high and the temperatures corresponding to the wellhead pressures of the single flash cycle are low. Binary units are well known for their ability to utilize low temperature geothermal energy and this study confirmed that common knowledge.

The difference in production cost are associated with the difference in the total purchased equipment cost of the power plants. The two double flash cycles and the hybrid cycle require more equipment than the conventional single flash cycle which results in a higher price per produced kWh. But as these cycles also produce more power than the single flash cycle, the total revenue from selling the product increases which often justifies the increased production cost. The highest production cost for lower enthalpies was achieved in the two double flash cycles where the modified double flash cycle carried a slightly higher cost than the conventional double flash power plant. The difference between the two increased with higher enthalpies until the enthalpy reaches 2300 kJ/kg. The reheat cycle then makes it possible to discard of the low pressure steam separator as the entire mass flow of separator water is used for the re-heating process. The reheat cycle then becomes the most economically viable option due to the increase in the total purchased equipment cost. The cost estimate of the hybrid flash-binary power plant showed that it is more economically viable than the double flash cycles for enthalpies under 1300 kJ/kg. The slope of the production cost for the hybrid cycle is much lower than for the flash cycles at low enthalpies as seen in Figure 5.18, which can lead to the conclusion that at even lower enthalpies than 1000 kJ/kg, the hybrid flash-binary power plant could carry even lower production cost than the traditional single flash power plant, making it the most economically viable options for low-enthalpic areas.

The results of this study can be used as a guide to estimate which power plant technology could provide the greatest power output and the best utilization of the geothermal fluid from the wells. Usually, studies like this one are carried out with a specific geothermal site in mind. The results of this study are restricted to the assumptions that had to be made regarding chemical content and pressure limitations of the hypothetical geothermal reservoir on which the models are based on. The thermodynamical models that were constructed can easily be adopted to specific limitations of a known geothermal reservoir and the results used as an estimate of the possible power production from that specific site.

6.1 Further Studies

Although the difference between the power output of modified double flash cycle introduced in this study and the conventional double flash cycle is not great, the modified double flash cycle could deserve some additional attention as it introduces the possibility to control the amount of massflow that is led towards the second flashing stage without compromising the thermal energy by discharging the brine immediately as it comes out of the power conversion system. Often, when geothermal reservoirs have been exploited for some time, the reservoir behavior changes, for example by pressure declining, that affect the production of the power plants. Equipment such as the

turbines in the geothermal power plants are designed for some fixed design consideration and if the initial conditions change drastically after production has started, the turbines will not operate at their full capacity. The addition of the recuperator with the possibility to superheat steam could then serve as a backup system if such changes in the reservoir occur.

Another interesting result for the reheat cycle is the development of the power cycle for higher enthalpies, as the low pressure steam separator is no longer needed and it becomes more efficient to use the total mass flow of separator water for the reheating process resulting in lower production cost of the electricity generated. A restriction of the use of the recuperator could be the concentration of dissolved chemicals in the geothermal fluid that is under consideration as the reheating process could become unachievable due to too much scaling in the heat exchanger equipment. If the possibilities of the modified double flash cycle would be investigated further it could lead to some interesting results.

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